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A REPORT OF RESEARCH CONDUCTED UNDER MARAD TASK S-11
OF THE SHIP PRODUCIBILITY RESEARCH PROGRAM
TO DETERMINE THE VALUE OF STANDARD STRUCTURAL ARRANGEMENTS

STANDARD STRUCTURAL ARRANGEMENTS

Prepared by:
General Dynamics Corp.
Quincy Shipbuilding Division

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Preface

The three individual technical reports which follow were written as the result of the Ship Producibility Task S-11, "Standard Structural Arrangements". The work was performed by the Quincy Shipbuilding Division of General Dynamics under contract to Bath Iron Works Corporation.

Industry groups meeting in advance of the formulation of Task S-11 had recommended a number of structural arrangements to be investigated for standardization. These were augmented in the pre-award proposal.

The complete candidate list was forwarded for evaluation to ten major U.S. shipyards, with follow-on visits to each yard. The results of interviews were summarized for each category under the following format:

Item:

Definition:

1. What are most likely areas of use
2. How prevalent is current use?
3. Is there a qualitative rationale?
4. Is there a quantitative rationale?
5. Is there a potential for developing the rationale and for the development of standards?
6. Can cost advantages of standards be established?
7. Existing technical criteria and guidelines
8. Potential Marine Graphics Handbook entry.
9. Potential design application for evaluation of benefit.
10. Estimated study manhours.

Results were summarized in the "Progress Summary, Task S-11", July 1976, which is on file at Bath Iron Works.

An advisory group appointed by the contractor evaluated all proposed categories for greatest usefulness to the industry and for compliance with the essential objective to reduce the cost of building ships in the United States.

Nineteen categories were ranked numerically and assigned three levels of priority:

Priority 1

- 1 Structural Details
- 2 Alignment Criteria
- 3 Repair Standards
- 4 Standard Frame Spacing
- 5 Construction Openings
- 6 Straight Line Frames
- 7 Standard Welding, Details

Priority 2

- 8 Level Longitudinal
- 9 Bow Framing
- 10 Stern Framing
- 11 Tripping Brackets
- 12 Parallel Decks and Standard Deck Framing
- 13 Foundations
- 14 Erection Breaks at In-plane Structure

Priority 3

- 15 Consideration of Deflection to Minimize Distortion
 and Vibration
- 16 Rudders
- 17 Water Scoops and Sea Chests
- 18 Reinforcing and Racking Preventatives for Heavy Lifts
- 19 Bilge Keels

Items 1, 2 and 11 (structural details, alignment criteria and tripping brackets) were ultimately selected for detailed study. The consensus lower priority item on tripping brackets was considered because of special interest to the investigators.

The authors gratefully acknowledge the guidance of Mr. John Mason, Bath Iron Works Project Manager, the funding contributed by MarAd and the time and courtesy extended by all shipyards visited. Thanks are due also to the members of the Advisory Committee.

Standard Structural Arrangements,
Task S-11 of the Ship Producibility Program

Executive Summary

The intent of Task S-11 was to reduce the cost of U.S. built ships by producing a series of standard structural arrangements. Study objectives were eventually modified to the formulation of design guidelines. Twenty candidate subjects were investigated for inclusion in the study, and three were selected:

Structural Details

Misalignment Tolerance

Tripping Brackets

The report on "Potential Ship Structural Details Guidelines" defines common structural details and provides brief descriptive narrative of application and attributes. Detailed static analysis was performed for structural intersections, generally the most common and troublesome structural detail.

Illustrations are provided for details in sixteen categories. These details represent what is best in U.S. shipbuilding practice, or the development of slightly modified configurations.

The report on "Potential Ship Structure Misalignment Tolerance Guidelines" presents a review of available standards and guidelines and of published experimental and theoretical data.

Analysis of butt and cruciform joints is combined with available fatigue data to produce specific guidelines for permissible static and dynamic misalignment Of common shipbuilding Joints.

The third report in this program, "Potential Ship Structure Tripping Bracket Guidelines" was motivated by current lack of suitable guidance to the **need for tripping brackets in ship structure. The report is a designer-oriented compendium of the necessary** theoretical formulations.

It is expected that thorough and deliberate review of these reports by industry groups such as SNAME panel SP-6 and ASTM Subcommittee F-25 can develop these tentative proposals into detailed consensus guidelines or standards.

The adoption of uniform structural details will generate specific performance feedback leading to reduced repair costs. Overall ship scantlings may eventually be reduced if margins for assumed stress concentrations or defects at structural details can be eliminated.

The misalignment criteria should be extended to the analysis of greater extent of misalignment and to the systematic investigation of suitable repair. Confirming full scale fatigue testing is desirable. Consensus guidelines will save considerable time now devoted to qualitative and quantitative arguments between builder, owner and regulatory agency inspectors, while improving ship reliability anti possibly decreasing fitup costs.

Guidelines for tripping bracket location are of significant economic value in ships with deep web frames, where frequent and arbitrary use of large brackets may greatly add to shipbuilding expense. General Dynamics has already employed this analysis to justify design alteration resulting in removal of several hundred large brackets per ship.

To complete these guidelines it remains to formulate a solution for asymmetric stiffeners (angles) in bending.

All three reports on potential guidelines are expected to be of immediate use to ship structural designers.

FINAL TECHNICAL REPORT

ON

POTENTIAL SHIP STRUCTURE
MISALIGNMENT TOLERANCE
GUIDELINES

PART NO. 1 OF

STANDARD
STRUCTURAL ARRANGEMENTS
TASK S-11
OF THE
SHIP PRODUCIBILITY PROGRAM

GENERAL DYNAMICS
QUINCY SHIPBUILDING DIVISION

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Section 1

INTRODUCTION AND SUMMARY

The aim of this report is to establish a basis for a set of guidelines to be used in the evaluation of errors in alignment. The importance of alignment criteria is clear:

1. Reference 1 states "The most commonly suggested tolerance problem was misalignment, especially misalignment of inter-costal at cruciform intersections". This was noted after discussions with ship owner/operator executives and yard executives.
2. Correction and rework of structural deviations may introduce locked-in stresses, flaws or weld defects which may be more detrimental to overall strength than the original defect.
3. The cost involved in repairing, inspecting, and re-repairing where necessary, while large, is probably small compared with the cost of delays, late delivery and consequent loss of goodwill.

Due to the lack of specific guidelines for dealing with misalignment, designers and engineers have had to evaluate each case using judgment and rules of thumb, and have had to convince inspectors and surveyors that the structural integrity has not been jeopardized.

Sections 2 and 3 of this study present a summary of published standards and guidelines and of the technical literature most applicable to the alignment subject.

Sections 4 and 5 present analytical work leading to the basis for design guidelines (Section 6) for static and fatigue strength of misaligned joints, respectively. Section 6 is essentially a self contained summary of the potential design guidelines derived in this report, and could be used independently.

A limited survey of reported cases of misalignment indicates that there is no predominant pattern of occurrence or repair. Guidelines for alignment criteria and for related structural alterations should lead to significant savings of ship construction time and cost.

Section 2

COLLECTION OF EXISTING STANDARDS AND GUIDELINES

2.1 GENERAL

This section is an attempt to list standards and guidelines applicable to alignment criteria contained in the rules of the major classification societies and of other applicable rule-making technical societies. Only those guidelines are presented that define numerical limits for misalignment.

The American Bureau of Shipping (ABS) has no formal set of quality control standards but generally agrees that structural tolerance limits depend on the location of the joint, quality of material and the allowable stress level. Bureau Veritas and Lloyd's Register of Shipping likewise have no published standards for tolerances.

Nippon Kaiji Kyokai has a set of standards "Japanese Shipbuilding Quality Standards - 1975" developed by the society of Naval Architects of Japan and the University of Tokyo. These standards provide numerical limits for strength and other members. The German Shipbuilding Industry developed the "Production Standard of the German Shipbuilding Industry" which is non-binding and not enforced by Germanischer Lloyd surveyors. The final set of formal standards presented is VIS 530, "Accuracy In Hull Construction", prepared by the Swedish Shipbuilding Standards Center.

Reference 1 is a timely summary of all applicable guidelines.

The following references have been reviewed and excerpted.

Throughout the remainder of this report the following indicated abbreviations will apply:

<u>REF. NO.</u>	<u>TITLE/SOCIETY</u>	<u>ABBREVIATION</u>
1)	Swedish Shipbuilding Standards Institute	VIS-530
3	Japanese Shipbuilding Quality Standards	JSQS
4	Association of German Shipbuilding Industry	PSGS
5	Ikawajima-Harima Heavy Industries	IHI
6	American Bureau of Shipping	ABS
7	Bureau Veritas	BV
8	Det Norske Veritas	DNV
9	Germanischer Lloyd	GL
10	Lloyd's Register	LR
11	Nippon Kaiji Kyokai	NKK

2.2 ALIGNMENT STANDARDS

2.2.1 Fillet Welded Cruciform Joints

Cruciform joints with fillet welds are the classic tolerance problem. This detail will be found in the fitting of brackets, intercostal, webs, bulkheads, longitudinals, etc. Figure 2-1 details the joint with respect to the documented standards.

$$T_1 \cong T_2$$

FIGURE 2-1

ALIGNMENT OF FILLET WELDED CRUCIFORM
JOINTS

1. JSQS : For strength members:
Tolerance Limit: $A \leq 1/3 T_1$
For other members:
Standard Limit $A \leq 1/3 T_2$
Tolerance Limit $A \leq 1/2 T_2$
2. PSGS : Assembly misalignment: internal members (Stiffening)
Out of alignment by more than 1/2 the plate or
profile thickness will bc disconnected and realigned.
3. VIS-530: For strength members; maximum divergence:
 $A \leq 1/4 T_3 + 1/8 \text{ in.} \leq 1/2 T_2$
For local members; maximum divergence:
 $A \leq 1/4 T_3 + 1/8 \text{ in.}$
4. IHI : For longitudinal members within 0.6L of
and for principal transverse supporting members:
 $A \leq 1/3 T_2$
For all others the allowable limit:
 $A \leq 1/2 T_2$
5. BV: Plates are to bc properly adjusted; a slight offset
is however tolerated over part of the length of the
joint, provided it does not exceed:
-0.08 in. where $T_2 < 0.25 \text{ in.}$
-0.12 in. where $T_2 \geq 0.25 \text{ in.}$
-0.08 in. for overhead fillet welds

2.2.2 Butt Welds in Continuous Plating

1. JSQS : For strength members the tolerance limit:

$$A \leq 0.15 T \leq 0.12 \text{ in.}$$

For other members the tolerance limit:

$$A \leq 0.20 T \leq 0.12 \text{ in.}$$

2. PSGS : In butt welds, plate misalignment may be:

$$A \leq 0.15 T \leq 0.16 \text{ in.}$$

3. VIS-530: For plate members:

$$A \leq 0.15 T \leq 0.16 \text{ in.}$$

4. IIII: Skin plates (bottom shell, side shell and deck plate) and longitudinal strength members, the

allowable limit: $A \leq 0.15 T \leq 0.12 \text{ in.}$

Bulkhead plates and interior members, the

allowable limit: $A \leq 0.20 T \leq 0.12 \text{ in.}$

5. BY:: When assembling plates of the same thickness, it is to be checked that they are correctly adjusted in height, a slight offset is however tolerated where this cannot be reduced in the normal way, provided it is not greater than the greater of the values:

$$A = 0.1T + 0.04 \text{ in.}$$

0.08 in. for butts

0.12 in. for longitudinal seams

The last two values may be increased by 0.04 in. where the joints are double vee groove.

FIGURE 2-2
ALIGNMENT OF BUTT WELDED JOINTS

2.2.3 Butt Welds in Flanges And Webs

1. VIS-530: Flange breadth (see Figure 2-3).

The maximum divergence:

$$A \leq 0.04B \leq 0.31 \text{ in.}$$

Flange height (see Figure 2-4).

The maximum divergence:

$$A \leq 0.2T_1$$

FIGURE 2-3

ALIGNMENT OF LONGITUDINAL FLANGES
W/RESPECT TO BREADTH
(REFERENCE 2)

FIGURE 2-4

ALIGNMENT OF LONGITUDINAL FLANGES

W/RESPECT TO HEIGHT
(REFERENCE 2)

Section 3
LITERATURE SURVEY OF EXPERIMENTAL
AND THEORETICAL DATA

3.1 GENERAL

Published papers on the subject of misalignment are few, and by their number do not reflect the industry's interest in the problem. A non-exhaustive review of the literature focuses attention on two papers on fatigue which provide valuable numerical data to this study, and on two papers on quality assurance which could provide the framework for design guidelines and standards.

3.2 FATIGUE TESTING

- A. "Fatigue Strength Of Butt Joint With Misalignment",
(Reference 12)

This paper presents the results of a study of the effect of misalignment in butt welded joints on fatigue strength. The paper also seeks to find allowable amounts of misalignment and a method for improving joint strength.

Test specimens were made of mild steel and two types of high tensile steel with thicknesses equal to 0.25 in. and 0.63 in. and mechanical properties as shown in Table 3-1.

The sizes and shapes of the test specimens are shown in Figures 3-1 and 3-2, along with the ratio of the amount of misalignment, δ , to the thickness of the specimens, T .

Two types of tests were carried out by the investigator: a static tension test and a series of fatigue tests.

TABLE 3-1
MECHANICAL PROPERTIES
OF TEST SPECIMENS

TYPE OF STEEL	PLATE THICK (IN)	σ_Y KSI	σ_B KSI	ELONG. (%)
SS-41	0.250	39.4	63.2	24.0
	0.625	34.4	60.4	24.5
HT-50	0.250	55.5	78.5	26.0
	0.625	51.9	81.7	24.3
HT-60	0.625	79.8	96.8	12.0

GAGE LENGTH = 7.875 IN

σ_Y - YIELD STRESS

σ_B - ULTIMATE TENSILE
STRESS

δ/T	0.0	0.2	0.4	0.6
δ IN	0.0	0.05	0.10	0.15

FIGURE 3-1
DETAILS OF EXPERIMENTAL
TEST SPECIMEN

δ/T	0.0	0.2	0.4	0.6
δ IN	0.0	0.125	0.25	0.375

FIGURE 3-2
DETAILS OF EXPERIMENTAL
TEST SPECIMEN

The static tension tests were performed on the 0.63 in. specimens. Results of the static tension tests are shown in Figures 3-3, 3-4 and 3-5. The ordinate in the figures represents the tensile stress (P/A) and the abscissa shows mean strain over a gauge length of 3.95 in. including the butt welded joint. From these results it is found that the static ultimate strength of the test specimens decreases as misalignment ratio increases. Figure 3-3 indicates that mild steel specimens with misalignment ratios equal to 0.4 and less have a significant plastic deformation range, while the $\phi = 0.6$ specimen sustains about half that deformation.

Misalignment ratios of up to 0.4 appear, from a pure stress stand point, to be acceptable. **Revised σ_B is as follows:**

$$\begin{aligned}\phi &= 0.2; & \sigma_B &= 59.0 \text{ ksi} \\ \phi &= 0.4; & \sigma_B &= 57.8 \text{ ksi} \\ \phi &= 0.6; & \sigma_B &= 52.0 \text{ ksi}\end{aligned}$$

All four specimens of HT-50 exhibit a large plastic strain zone, and the HT-60 specimens do not.

Failure of the specimens generally occurred in the base metal. and not in the welded joint.

Fatigue tests were also performed on the mild steel specimens while controlling tensile deformations. Figures 3-6 and 3-7 present test results for the 0.25 in. and 0.63 in. specimens, respectively. Tests were run on five different configurations ranging from a machined piece with no weld to the 0.6T misaligned welded joint. As expected, the fatigue lives of the specimens decrease with increase in misalignment. This decrease in fatigue life is clearly related to the increased stress caused by stress concentrations at the weld toe and by the eccentricity of the

FIGURE 3-3

RESULTS OF STATIC TENSION TEST

(REFERENCE 12)

FIGURE 3-4
RESULTS OF STATIC TENSION TEST
(REFERENCE 12)

FIGURE 3-5
RESULTS OF STATIC TENSION TEST
 (REFERENCE 12)

FIGURE 3-6

S-N DIAGRAM FOR BUTT WELD • MILD STEEL: $T = 0.25$ IN
 (REFERENCE 12)

FIGURE 3-7
S-N DIAGRAM FOR BUTT WELDS: MS: T=0.625 IN
 (REFERENCE 12)

misaligned plates. Although the effect of the misaligned plates cannot easily be corrected, the severity of the stress concentration factor can be lessened by dressing the weld.

B. "An Experimental Study On Reduction Of Fatigue Strength Due To Discrepancy At Welded Joints" (Reference 13)

This study investigated the effect that misalignment has on the fatigue strength of a cruciform joint. The test specimens used are shown in Figure 3-8. All specimens were of SS-41 steel, whose mechanical properties are presented in Table 3-1. .

The specimens modeled misalignment equal to 0.0T, 0.25T, 0.50T, 1.0T and 2.0T. Each specimen was tested to failure under cyclic loading. Figure 3-9 presents the results of these tests.

Table 3-2 summarizes the effect of increasing the number of cycles as a function of degree of misalignment. The influence of misalignment is more significant in the high cycle range ($N = 10^3 \sim 10^5$) than in static tests.

3.3 QUALITY CONTROL AND ASSURANCE

Two references are summarized in this section. Each describes a separate approach to the problem of evaluating the acceptability of structural imperfections and construction errors.

A. "The Acceptability Of Weld Defects" (Reference 16)

Although this paper deals exclusively with weld defects, the suggestions made and methods described could be adapted to misalignment. What follows is a general discussion of the paper with specific reference to modifications that would encompass misalignment and structural imperfections.

The expressions "fitness for purpose", "significance of defects", "critical defect evaluation", and "engineering critical assessment" have been introduced in recent years as an attempt to show

FIGURE 3-8
DETAILS OF
EXPERIMENTAL
TEST SPECIMEN

3-13

δ CYCLES	0MM	2MM ($\frac{I}{4}$)	4MM ($\frac{I}{2}$)	8MM (T)	16MM (2T)
$\frac{1}{2}$	100	97	93	86	78
10^2	100	96	92	83	79
10^3	100	89	85	79	72
10^4	100	85	79	73	64

TABLE 3-2
REDUCTION IN EFFECTIVENESS (PERCENT) AS A
FUNCTION OF MISALIGNMENT AND NUMBER OF
CYCLES
 (REFERENCE 13)

justification for allowing structural imperfections to remain and as an incentive to high quality workmanship.

Existing standards are sketchy and arbitrarily evolved. Each classification society maintains their own set of informal guidelines, generally not released for shipyard use.

Two strong arguments can be advanced for the adoption of realistic acceptance standards:

1. Economic: The labor cost involved in repairing, reinspecting and re-repairing where necessary, while large, is probably insignificant compared with cost of delays.
2. Risk Factor: The danger that exists in repairing a harmless and readily detectable defect is that a more harmful less detectable fault will be introduced.

The aim is not to impose standards that will result in a general lowering of structural quality but rather to arrive at a set of standards that are reasonable and allow the structure to perform its intended purpose. Figure 3-10 depicts a proposal by the International Institute of Welding Commission for Welding Standards. Level B is the fitness for purpose quality fixed on the basis of engineering assessment with a suitable factor of safety. Level A is the quality level indicating good workmanship. Between Levels A and B, no repairs are necessary but the reasons for the loss of quality are investigated with a view to encouraging a desirable trend towards improvement.

The following recommendations have been made for the shipbuilding industry

1. Each basic ship design should be divided into various areas for the purposes of quality control and acceptance.

FIGURE 3-10
SCHEMATIC DIAGRAM OF QUALITY CONTROL AND ACCEPTANCE
LEVELS (REFERENCE 16)

2. Two sets of limits or standards should be developed for each area, one designated "Fitness For purpose" and the other known as the "Workmanship Standard".

B. "Assessment Of Imperfections In Ship Structural Design"
(Reference 17)

The main purpose of this paper is to describe a rational method of investigating the effect of changes in tolerance levels on structural design. Where tolerances are to be assessed it should be possible to investigate in a rational way:

1. The consequences of relaxing or tightening the tolerance.
2. The consequences of varying the tolerance depending on location.

As far as structural design is concerned, the main effect of altering tolerance limits is to either increase or decrease the load carrying capacity of the structure, thus altering the safety and reliability of the structure. The areas that will affect the risk of failure of the structure are:

1. Dimensional Control
2. Misalignment
3. Deformation
4. Welding/Cutting Distortion

The importance of risk associated with these categories depends upon the consequences of failure. Too rigid a tolerance may be prohibitive in cost.

The reliability assessment system has been developed to study and determine the effects of variations around a mean value in the parameters which describe the load, response and capability of a structure. This system may be used to assist in the appraisal of quality control standards for geometrical variables which, in addition to overall structural dimensions, includes imperfections and misalignment.

Structural reliability is defined as the probability that the structure will perform its intended functions for a specified time period when subjected to the operational loading conditions. A system designed to predict reliability would be used to determine acceptable tolerances.

The reliability assessment system is divided into four phases:

- Phase I: Data Base Input Stage
- Phase II: Capability Assessment
- Phase III: Loading Assessment
- Phase IV: Reliability Assessment

Phase I: Describes material scantlings, mechanical properties, construction data, and structural definition. It forms the permanent data base.

Phase II: Accurately identifies the design variables and determines type of distribution for each variable. In addition to these a capability distribution is established.

Phase III: Is a statistical definition of the wave induced and still water loads with necessary probability density functions.

Phase IV: This phase relates reliability to either damage or collapse.

This system could be used as a framework to help determine permissible standards (tolerances). The shortcomings noted for this system are that the locations of details are not considered with respect to the overall structural configuration likewise the consequence of failure of a detail or arrangement on the overall behavior of the structure varies with the location of the failure.

Section 4

RESULTS OF FINITE ELEMENT ANALYSES

4.1 GENERAL

The problem of misalignment has been discussed extensively, but as shown in Section 3 actual data are meager. Some shipyards have performed finite element or other analyses on specific alignment problems, when repair of such areas did not appear practical or possible. However, the results of these analyses have not been formally presented to the industry.

For purposes of this study, finite element analyses have been carried out. Three types of joints have been studied: fillet welded cruciform, full penetration welded cruciform and butt welded. In-plane loading has been investigated for varying degrees of misalignment. The results obtained from these analyses, while not complete, will with good engineering judgement facilitate the development of criteria for a variety of misalignment conditions. It is also hoped that this information may be helpful in assessing cases not covered in this study.

4.2 COMPUTER PROGRAM DESCRIPTION

All investigation and analysis has been performed utilizing the Ices Strudl-II Computer Program.

Strudl uses a finite element method where the element stiffness matrix is computed from energy considerations, after selecting a displacement or force expansion over the element, and assuming that displacements or force quantities in the

interior of the element depend on nodal quantities. The stiffness analysis is a linear, elastic, static, small displacement analysis where joint displacements are treated as unknowns. For elements the analytic procedure provides stresses or stress resultants and couples, principal stresses, strains, and principal strains, usually computed at the centroid of each element.

For additional program documentation and the description of plane strain methodology see References 18 and 19.

4.3 FINITE ELEMENT RESULTS

4.3.1 Elastic Analysis of Butt Welded Joints

The basic form of the butt welded joint used in this investigation is shown in Figure 4-1. Figures 4-2 through 4-6 present the finite element models used in the analysis. Misalignment of $6 = 0.20T$ to $= 1.00T$ in increments of $0.20T$ was investigated. Plane strain elements have been used because the plate is assumed to be long in the Z-direction with a uniformly distributed in-plane load. The deformation of the body at some distance from the ends is independent of the Z-coordinates and the displacements are functions of X and Y only. If the ends of the plate are prevented from moving in the Z-direction, then W is zero there. At the midsection of the plate, by symmetry W must also be zero. Thus the assumption that W is zero at every cross section of the plate. In such a case, the strain components:

$$\epsilon_x = \frac{\partial u}{\partial x} \quad \epsilon_y = \frac{\partial v}{\partial y} \quad \gamma_{xy} = \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}$$

CASE 2
STRESS INTENSITY

FIGURE 4-3
ALIGNMENT STUDY
CATEGORY 18 BUTTS AND SEAMS
IN PLATE
CASE 28 21 MISA SUMMEN
○ - JOINT NUMBERS
○ - ELEMENT NUMBERS

CASE 3
FINITE ELEMENT MODEL
FOR ADDITIONAL DETAILS SEE CASE 2

CASE 3
STRESS INTENSITY PLOT

FIGURE 4-3

ALIGNMENT STUDY	
CATEGORY 1: BUTTS AND SEAMS IN PLATING	
CASE 3: 0.4T MISALIGNMENT	
○	-NODE NUMBERS
●	-ELEMENT NUMBERS

CASE 4
FINITE ELEMENT MODEL
FOR ADDITIONAL DETAILS SEE CASE 2

CASE 4
STRESS INTENSITY PLOT

FIGURE 4-4

ALIGNMENT STUDY
CATEGORY 1: BUTTS AND SEAMS IN PLATING
CASE 4: 0.6 MISALIGNMENT
○ - NODE NUMBERS
⬡ - ELEMENT NUMBERS

CASE 5
FINITE ELEMENT MODEL
FOR ADDITIONAL DETAILS SEE CASE 2

CASE 5
STRESS INTENSITY PLOT

FIGURE 4-5

ALIGNMENT STUDY
CATEGORY 1: BUTTS AND SEAMS
IN PLATING
CASE 5: C.B.T. MISALIGNMENT
○ - NODE NUMBERS
□ - ELEMENT NUMBERS

CASE 6
FINITE ELEMENT MODEL
FOR ADDITIONAL DETAILS SEE CASE 2

CASE 6
STRESS INTENSITY PLOT

FIGURE 4-6
ALIGNMENT STUDY
CATEGORY 18 BUTTS AND SEAMS
IN PLATING
CASE 6: LOT MISALIGNMENT
- NODE NUMBERS
- ELEMENT NUMBERS

are functions of X and Y only and the strain components:

$$\epsilon_z = \frac{\partial w}{\partial z} \quad \gamma_{xz} = \frac{\partial w}{\partial x} + \frac{\partial u}{\partial z} \quad \gamma_{yz} = \frac{\partial w}{\partial y} + \frac{\partial v}{\partial z}$$

are equal to zero.

A uniaxial compressive load was applied with a magnitude equal to P (^{LB}/IN.) acting over a cross-sectional area equal to A (IN²/IN.1, such that in the aligned case, the axial stress (σ_x) was uniform and equal to:

$$\sigma_x = \frac{P}{A}$$

The compressive load does not detract from generality, since linearity of response assures equal absolute stress magnitudes for an applied tensile load.

Stress intensity factors, i.e. stress amplification of the mean stress for the misaligned butt welds have been plotted in Figures 4-2 through 4-6. A stress intensity factor of 1.0 is the ideal design condition.

The stress intensity plots indicate maximum stress concentrations at the toes of the welds. Gurney in Reference 20 has investigated and reported on the effect and magnitude of these stress concentrations.

The stress intensity plots for the welded butt joints have been constructed from the σ_x (axial) stresses since these are the dominant stress components.

Table 4-1 presents a synopsis of the stress intensity plots with corresponding maxima for each case of misalignment.

MISALIGNMENT δ : T-THICKNESS	MAXIMUM STRESS INTENSITY FACTOR	$\frac{\sigma_{ALL.}}{\sigma_{ACTUAL}}$
0.2T = 0.2 in.	1.67	0.60
0.4T = 0.4 in.	2.35	0.43
0.6T = 0.6 in.	2.90	0.34
0.8T = 0.8 in.	3.50	0.29
1.0T = 1.0 in.	3.90	0.26

TABLE 4-1
FINITE ELEMENT ANALYSIS RESULTS
FOR WELDED BUTT JOINTS

The stress intensity factor, as stated earlier, is the factor by which the peak stress exceeds the mean or P/A stress across the section. The third column in Table 4-1, $\sigma_{ALL}/\sigma_{ACTUAL}$, is the inverse of column 2, namely the ratio of mean section stress to maximum acceptable stress. The elongation for the total length of the model, i.e., 9.6 inch, For $\xi=0.0$ is 0.0064 inch.

Figure 4-7 represents a modified plot of data in Table 4-1. The abscissa indicates percent misalignment with respect to plate thickness; the ordinate represents the ratio of average stress to allowable stress, where the allowable stress is established by the designer. Three possible values of allowable stress are assumed: $0.6\sigma_Y$, incipient yield, and total yielding.

The curves are drawn against individual ordinates to provide a visual measure of permissible load amplitudes. Zone 1 is limited by a peak stress of $0.6\sigma_Y$ Zone 2 is limited by σ_{YIELD} , i.e. yield at some point in the structure, and Zone 3 by $\sigma_{ULTIMATE}$, where the latter is defined not as tensile failure but as the level at which the entire cross-section reaches yield.

The applicability of this data is essentially limited to statically loaded members, where the attainment of partial yielding is generally accepted. Consequently a design guideline would most likely use the curve for $\sigma_{ALL} = \sigma_{YIELD}$. For example, For- $\sigma_{ALL} = \sigma_{YIELD}$ the average static stress with $\xi = 0.5$ could be approximately $0.45\sigma_Y$, i.e. 15 KSI for a steel with $\sigma_Y = 33$ KSI.

FIGURE 4-7: WELDED BUTT JOINTS / MISALIGNMENT VERSUS STRESS RATIO

4.3.2 Plastic Analysis of Butt Welded Joints

Generally, localized yielding will not be detrimental to the structural integrity of the ship. Therefore some structural elements may be stressed above the elastic limit. This section discusses how the finite element method has been used to determine the elasto-plastic response of structural configurations considered in this study.

The elastic response of a structure for any given load condition can be checked against the capability of the structure. By this process the adequacy of the design can be evaluated. The designer often performs a sophisticated elastic analysis of the structure and is then forced to make a series of simplifying assumptions when determining the capability, thus negating the value of the analysis.

The finite element method has been used in this investigation of the elasto-plastic behavior of plate structures. The approach is a linear step-wise procedure. The method assumes the structure behaves linearly and that the total stiffness matrix can be formed by considering the geometry and material properties associated with the individual elements at that particular step. The analysis is accomplished in the following manner:

1. An initial load is chosen within the elastic range.
2. The Stiffness matrix for the structure is formed based on the element material properties.
3. Load is increased until at least one element nodal stress reaches σ_{YIELD} .

4. Elements that indicate yielding are replaced with an artificial force system, which is equilibrated to the resistive body forces at yield.
5. Combined elastic Properties and the artificial load system are used to continue the analysis.
6. The process is repeated until a transverse section of the structure experiences total yielding.

(For additional details see Reference 21.)

The case of 1.0T misalignment is the only condition investigated in this analysis, and the data have been used to extrapolate from the incipient yield curve to provide results for all intermediate cases of misalignment.

The extrapolation is made by reducing the ratio of $\sigma_{Ult}/\sigma_{Yield}$ at $\delta = 1.00$ by the value of δ , thus accounting for the fact that at $\delta = 0$ initial and total yielding occur simultaneously.

Figure 4-8 represents the process of progressive yielding of elements and Figure 4-7 shows the extrapolated curve for $\sigma_{ALL} = \sigma_{ULTIMATE}$. Appendix A discusses the calculations necessary for replacing the yielded element with the resistive force system. Results of this analysis reveal that the total yield load is forty percent greater than the incipient yield load at $\delta = 1.0$.

4.3.3 Elastic Analysis of Fillet Welded Cruciform Joints

The basic form of the joint and of the finite element model used in this investigation of the fillet welded cruciform joints is shown in Figures 4-9 to 4-13. Misalignment from $\delta = 0.0T$ to $\delta = 1.0T$ in increments of $\delta = 0.20$ was investigated. Plane strain elements with a thickness equal to 1 inch for the misaligned members and 1/2 inch and 1 inch for

DETAIL A

DETAIL D

DETAIL B

DETAIL E

DETAIL C

FIGURE 4-8
WELDED BUTT JOINTS
MISALIGNMENT = 1/8"
RESULTS OF INCREMENTAL
PLASTIC ANALYSIS

CASE 18: 0.2T MISALIGNMENT
FULL PENETRATION WELD
THROUGH PLATE THICKNESS
= 0.5T

CASE 21: 0.2T MISALIGNMENT
FILLET WELD
THROUGH PLATE THICKNESS
= 0.5T

CASE 31: 0.2T MISALIGNMENT
FULL PENETRATION WELD
THROUGH PLATE THICKNESS
= 1.0T

CASE 41: 0.2T MISALIGNMENT
FILLET WELD
THROUGH PLATE THICKNESS
= 1.0T

FIGURE 4-9

ALIGNMENT STUDY

CATEGORY 2: CRUCIFORM
WELDS

FINITE ELEMENT MODELS

○ - NODE NUMBERS

○ - ELEMENT NUMBERS

CASE 5: 0.4T MISALIGNMENT
FULL PENETRATION WELD
THROUGH PLATE THICKNESS
= 0.4T

CASE 6: 0.4T MISALIGNMENT
FILLET WELD
ROUGH PLATE THICKNESS
= 0.4T

FOR ADDITIONAL DETAILS
SEE CASE 5

CASE 7: 0.4T MISALIGNMENT
FULL PENETRATION WELD
THROUGH PLATE THICKNESS
= 0.4T

CASE 8: 0.4T MISALIGNMENT
FILLET WELD
ROUGH PLATE THICKNESS
= 0.4T

FOR ADDITIONAL DETAILS
SEE CASE 6

FIGURE 4-10
MISALIGNMENT STUDY
CATEGORY 2: CRUCIFORM
WELDS
FINITE ELEMENT MODELS

- - NODE NUMBERS
- - ELEMENT NUMBERS

CASE 10: 0.6T MISALIGNMENT
 FULL PENETRATION WELD
 THROUGH PLATE THICKNESS
 -0.5T

FOR ADDITIONAL
 DETAILS SEE CASE 9

CASE 11: 0.6T MISALIGNMENT
 FULL PENETRATION WELD
 THROUGH PLATE THICKNESS
 -0.5T

FOR ADDITIONAL
 DETAILS SEE CASE 11

FIGURE 4-11
 CASE 11 STUDY
 CATEGORY 2: CRACKS
 FINITE ELEMENT MODELS
 ○ - NODE NUMBERS
 ○ - ELEMENT NUMBERS

CASE 17: LOT MISALIGNMENT
FULL PENETRATION WELD
THROUGH PLATE THICKNESS
-0.5T

CASE 18: LOT MISALIGNMENT
FILLET WELD
THROUGH PLATE THICKNESS
-0.5T

CASE 19: LOT MISALIGNMENT
FULL PENETRATION WELD
THROUGH PLATE THICKNESS
-1.0T

CASE 20: LOT MISALIGNMENT
FILLET WELD
THROUGH PLATE THICKNESS
-1.0T

FIGURE 4-13
ALIGNMENT STUDY
CATEGORY 2: CRUCIFORM
WELDS
FINITE ELEMENT MODELS
○ - NODE NUMBERS
○ - ELEMENT NUMBERS

the continuous through plate were used. A root gap of approximately 1/16 inch has been included in the model.

A uniaxial compressive load was applied. Stress intensity factors for the misaligned cruciform joints have been plotted in Figures 4-14 to 4-23.

The stress intensity factors indicate large stress concentrations at the toes of the fillets and at the corners of the root gap. This is consistent with work done by Gurney (Reference 20), and Sotah (Reference 22).

The stress intensity plots for the welded cruciform joints have been constructed from calculated Von Mises stress where:

$$\sigma_{VON} = \sqrt{\sigma_x^2 + \sigma_y^2 - \sigma_x \sigma_y + 3\tau_{xy}^2} = \sigma_{ACTUAL} \quad .$$

(See Figure 4-1 for explanation of terms.) .

Tables 4-2 and 4-3 present the stress intensity factors with corresponding maxima for each case of misalignment. The maximum stress intensity factor occurs at the toe of the fillet weld.

Figures 4-24 and 4-25 present a series of curves based on Tables 4-2 and 4-3. The abscissa indicates percent misalignment with respect to plate thickness; the ordinate represents the ratio of average stress to allowable stress. Figure 4-24 depicts three curves and four designated zones for the fillet welded cruciform joints with a through plate thickness of 0.5T. Figure 4-25 presents similar data for a through plate thickness of 1.0T. The $T_{ULTIMATE}$ curve has been approximated by use of results reported in the following section.

FIGURE 4-1A
ALIGNMENT STUDY
CATEGORY 2: CRUCIFORM
JOINTS
STRESS INTENSITY PLOTS
0.2T MISALIGNMENT
2.5T THRU PLT THICKNESS

FIGURE 14-15
ALIGNMENT STUDY
CATEGORY 2: CRUCIFORM
JOINTS
STRESS INTENSITY PLOTS
0.2T MISALIGNMENT
1/8T THRU PLATE THICKNESS

FIGURE A-19
ALIGNMENT STUDY
CATEGORY 2: CRUCIFORM
JOINTS
STRESS INTENSITY PLOTS
0.4T MISALIGNMENT
LOT THRU PLATE THICKNESS

FIGURE 4-18
ALIGNMENT STUDY
CATEGORY 2: CRUCIFORM
JOINTS
STRESS INTENSITY PLOTS
0.6T MISALIGNMENT
0.5T THRU PLATE THICKNESS

Case

CASE 12

FIGURE 4-19
ALIGNMENT STUDY
CATEGORY 2: CRUCIFORM
JOINTS
STRESS INTENSITY PLOTS
0.6T MISALIGNMENT
1.0T THRU PLATE THICKNESS

FIGURE 8-22
ALIGNMENT STUDY
CATEGORY 2: CRUCIFORM
JOINTS
STRESS INTENSITY PLOTS
WOT MISALIGNMENT
0.5T THRU PLATE THICKNESS

FIG. 4-33
ALIGNMENT STUDY
CATEGORY: 1.0
STAFF: 1.0
UNIT: 1.0
SCALE: 1.0

MISALIGNMENT δ : T - THICKNESS	MAXIMUM STRESS INTENSITY FACTOR	$\frac{\sigma_{ALL.}}{\sigma_{ACTUAL}}$
0.2T = 0.2 in.	1.67	0.60
0.4T = 0.4 in.	2.05	0.49
0.6T = 0.6 in.	2.73	0.37
0.8T = 0.8 in.	3.24	0.31
1.0T = 1.0 in.	4.06	0.25

TABLE 4-2
FINITE ELEMENT ANALYSIS RESULTS
FOR FILLET WELDED CRUCIFORM JOINTS
W/ THROUGH PLATE THICKNESS = 0.50T

MISALIGNMENT δ : T-THICKNESS	MAXIMUM STRESS INTENSITY FACTOR	$\frac{\sigma_{ALL.}}{\sigma_{ACTUAL}}$
0.2T = 0.2 in.	1.67	0.60
0.4T = 0.4 in.	2.29	0.44
0.6T = 0.6 in.	2.81	0.36
0.8T = 0.8 in.	3.41	0.29
1.0T = 1.0 in.	4.07	0.25

TABLE 4-3
FINITE ELEMENT ANALYSIS RESULTS
FOR FILLET WELDED CRUCIFORM JOINTS
W/ THROUGH PLATE THICKNESS = 1.0 T

FIGURE 4-24: FILET WELDED CRUCIFORM JOINTS MISALIGNMENT VERSUS
STRESS RATIO W/THROUGH PLATE THICKNESS = 0.50T

FIGURE 4-25: FILLET WELDED CRUCIFORM JOINTS MISALIGNMENT VERSUS
STRESS RATIO W/ THROUGH PLATE THICKNESS = 1.0T

4.3.4 Plastic Analysis of Fillet Welded Cruciform Joints

Plastic analysis for the cruciform joint has been performed only for the case of 0.6T misalignment. Results of this analysis are recorded in Appendix A and have been used to develop the curve (σ_{ULTIMATE}) shown on Figures 4-24 and 4-25. Figure 4-26 illustrates the progressive yielding as calculated for the finite element model.

4.3.5 Elastic Analysis of Full Penetration Cruciform Joints

The basic form of the joint used in this investigation of the full penetration cruciform joints is shown in Figures 4-7 to 4-11, which describe the finite element models used in this analysis. Misalignment of $\delta = 0.0T$ to $\delta = 1.0T$ in increments of $\delta = 0.20$ were investigated. Plane strain elements with a thickness equal to 1 inch for the misaligned members and 1/2 and 1 inch for the continuous through plate were used in the analysis. A root gap of 0 inch has been included in the model.

A uniaxial compressive load was applied as previously described in Section 4.3.3 stress intensity factors for the misaligned cruciform joints have been plotted in Figures 4-12 to 4-22. These stress intensity plots do not exhibit the large internal stress concentration that were evident for the fillet welds with root gap. The maximum stress intensity factor occurs at the toe of the reinforcing fillet. The plots have been developed using the dominant σ_x stresses.

Tables 4-4 and 4-5 present the stress intensity factors with corresponding maxima for each case of misalignment.

FIGURE 4-26
FILLET WELDED CRUCIFORM
JOINT
MISALIGNMENT = $0.6T$
RESULTS OF INCREMENTAL
PLASTIC ANALYSIS

MISALIGNMENT δ : T - THICKNESS	MAXIMUM STRESS INTENSITY FACTOR	$\frac{\sigma_{ALL.}}{\sigma_{ACTUAL}}$
0.2T = 0.2 in.	1.34	0.74
0.4T = 0.4 in.	1.78	0.56
0.6T = 0.6 in.	2.51	0.40
0.8T = 0.8 in.	3.21	0.31
1.0T = 1.0 in.	4.21	0.24

TABLE 4-4
FINITE ELEMENT ANALYSIS RESULTS
FULL PENETRATION CRUCIFORM JOINTS
W/THROUGH PLATE THICKNESS = 0.50 T

MISALIGNMENT δ : T-THICKNESS	MAXIMUM STRESS INTENSITY FACTOR	$\frac{\sigma_{ALL.}}{\sigma_{ACTUAL}}$
0.2T = 0.2 IN	1.47	0.68
0.4T = 0.4 IN	2.10	0.48
0.6T = 0.6 IN	2.67	0.37
0.8T = 0.8 IN	3.32	0.30
1.0T = 1.0 IN	4.02	0.25

TABLE 4-5

FINITE ELEMENT ANALYSIS RESULTS

FULL PENETRATION CRUCIFORM JOINTS

W/THROUGH PLATE THICKNESS=1.0T

Figures 4-27 and 4-28 represent plots of the above data. Each depicts three curves and four designated zones for the full penetration cruciform joints with a through plate thickness of $0.5T$ and $1.0T$, respectively. No elasto-plastic analysis was performed on these joints: Instead, the information presented for the fillet welded cruciform joint has been used to construct a set of extrapolated **JULTIMATE** curves.

FIGURE 4-27: FULL PENETRATION CRUCIFORM JOINTS MISALIGNMENT VERSUS
STRESS RATIO W/THROUGH PLATE THICKNESS = 0.50T

FIGURE 4-28: FULL PENETRATION CRUCIFORM JOINTS MISALIGNMENT VERSUS STRESS RATIO σ/σ_{YIELD} THROUGH PLATE THICKNESS = 1.0T

Section 5 FATIGUE STRENGTH OF MISALIGNED JOINTS

5.1 GENERAL

The most critical ship structural elements are generally subjected to cyclic or fatigue type of loading. For such elements the guidance to be obtained from the results of the preceding static finite element analysis is not sufficient.

Reference 12 and 13, discussed in detail in Section 3, provide experimental fatigue data related to misaligned butt welded and cruciform joints. Although these data by themselves provide little guidance to the designer, they form an adequate basis for-extrapolated and generalized data to be derived and discussed in the following paragraph.

5.2 METHODOLOGY

The available experimental data are for a stress ratio $R = 0$ (R is the algebraic ratio of minimum to maximum stress). Although it is becoming generally recognized that stress range is more relevant to the true fatigue performance of welded elements than is stress ratio (Reference 23), these experimental values have nonetheless been extrapolated to a stress ratio $R = -1$ (full stress reversal). That stress ratio is more typical of ship structural behavior, more conservative for design and as will be seen, simplifies the response analysis.

Extrapolation is performed by using a factor corresponding to design fatigue stress values for mild steel taken from Table 10.4 of Reference 24. The experimental data are limited to tests up to 10^5 cycles, whereas a twenty year ship life corresponds to about 10^8 wave encounters or cycles. Extrapolation to the higher range is made by using ratios of higher cycle fatigue strength, i.e., 6×10^5 and 2×10^6 to strength at 10^5 cycles, again taken from Reference 24. The final extrapolation to 10^8 cycles is made graphically.

The expressions used to form the above ratios are:

<u>No. of Cycles</u>	<u>Allowable Stress</u>
2,000, 000	$F_A = \frac{19000}{(1 - 0.73R)}$
600,000	$F_A = \frac{24000}{(1 - 0.60R)}$
100,000	$F_A = \frac{28000}{(1 - 0.75R)}$

Where:

F_A = Allowable unit fatigue stress, psi

R = Algebraic ratio of minimum to maximum stress.

The fatigue curves for butt and cruciform joints, discussed and shown in Sections 5.3 and 5.4, are then used to calculate (by an iterative process) the maximum permissible stress range for a stress histogram of a ship structural element over 10^8 cycles.

A sample histogram is shown in Figure 5-1. Note that with number of cycles plotted on a logarithmic abscissa, straight line elements join maximum and zero stress levels. This is an approximation justified by a generally similar measured pattern of probability distribution for linearized ship responses to a long term sea spectrum.

Maximum values of stress used in constructing the histogram can be approximated by the calculated response to bending moment, shear or pressure corresponding to peak quasistatic design wave height, such as $L/20$ or $1.0\sqrt{L}$.

The Palmgren-Miner cumulative fatigue damage theory is used in conjunction with a finite subdivision of the histogram.

FIGURE 5-1
SIMPLIFIED
STRESS HISTOGRAM

The expression

$$\sum \frac{n_i}{N_i} \leq 0.5$$

Where n_i = number of cycles at stress σ_i
 N_i = number of cycles to failure at σ_i

implies a two-fold margin on service life.

By using $R = -1$ the stress ratio is equal for every block in the histogram, which simplifies the calculation and greatly reduces the quantity of required experimental data.

As an example, the final iterative cycle to establish the extreme permissible stress range corresponding to the S-N diagram in Figure 5-5 is outlined below:

Step 1. The maximum stress is divided into eight equal increments and these are plotted on the ordinate.

Stress increment = 1 ksi

Values plotted: (8, 7, 6, 5, 4, 3, 2, 1)

Step 2. Extend lines parallel to abscissa until they cross the respective S-N curve.

Points of intersection are to be used as N values.

Step 3. Calculate cumulative damage factor:

$$\sum_{i=1}^8 \frac{n_i}{N_i} = \frac{2}{26000} + \frac{30}{74000} + \frac{500}{600000} + \frac{3000}{2000000} + \frac{30000}{6000000} + \frac{200000}{10000000} + \frac{2000000}{25000000} + \frac{20000000}{70000000} = 0.52$$

$$\sum \frac{n_i}{N_i} \leq 0.50$$

5.3 FATIGUE DATA FOR SEAMS AND BUTTS

Figures 5-2 to 5-5 are the S-N diagrams for butt welded plates with misalignment varying from 0.0 to 0.6T. The $R = 0$ curves are shown only as reference.

Calculating maximum permissible stress range for $R = -1$ by the method of Paragraph 5.2 leads to the summary in Table 5-1. The tabulated values can be very useful in the evaluation of functional adequacy of misaligned joints subject to cyclic loading.

FIGURE 5-2
S-N DIAGRAM FOR
BUTT WELDS
MILD STEEL: $\delta = 0.0$

FIGURE 5-3
S-N DIAGRAM FOR
BUTT WELDS
MILD STEEL: $\frac{\delta}{T} = 0.2$

FIGURE 5-4
S-N DIAGRAM FOR
BUTT WELDS
MILD STEEL: $\frac{\sigma}{T} = 0.4$

FIGURE 5- 5
S-N DIAGRAM FOR
BUTT WELDS
MILD STEEL: $\frac{\sigma}{T} = 0.5$

TABLE 5-1

MISALIGNED BUTT WELDS	
δ/T	MAXIMUM STRESS RANGE ^(R=-1)
0.0	37200 PSI
0.2	29200 PSI
0.4	26000 PSI
0.6	15600 PSI

MAXIMUM CALCULATED STRESS
RANGES FOR MISALIGNED BUTT
WELDS EXPERIENCING CYCLIC LOADING

5.4 FATIGUE DATA FOR CRUCIFORM JOINTS

Figures 5-6 to 5-9 are the S-N diagrams for cruciform welded plates with misalignment ratio δ varying from 0.0 to 1.0. Again the curves for $R = 0$ are shown only as reference.

Calculated maximum permissible stress ranges for $R = -1$ are listed in Table 5-2.

FIGURE 5-6
S-N DIAGRAM FOR
CRUCIFORM JOINTS
MILD STEEL: $\frac{\sigma_2}{\sigma_1} = 0.0$

FIGURE E-7
S-N DIAGRAM FOR
CRUCIFORM JOINTS
MILD STEEL $\bar{r}_1 = 0.25$

FIGURE 5-8
S-N DIAGRAM FOR
CRUCIFORM JOINTS
MILD STEEL: $\frac{\delta}{T_1} = 0.50$

FIGURE 5-9
S-N DIAGRAM FOR
CRUCIFORM JOINTS
MILD STEEL: $\frac{\sigma}{T_1} = 0.0$

TABLE 5-2

MISALIGNED CRUCIFORM JOINTS	
δ/T_1	MAXIMUM STRESS RANGE ^(R=-1)
0.0	38400 PSI
0.25	32800 PSI
0.50	28000 PSI
1.0	24000 PSI

MAXIMUM CALCULATED STRESS
RANGES FOR MISALIGNED CRUCIFORM
JOINTS EXPERIENCING CYCLIC LOADING

Section 6

DATA FOR MISALIGNMENT ACCEPTABILITY GUIDELINES

6.1 GENERAL

This section summarizes the results of static and fatigue analysis of butt welded and cruciform joints discussed in Sections 4 and 5. The relevant plots and tables from those sections are reproduced here for easy reference. The limitations inherent in their potential use can best be understood by reading earlier sections.

6.2 STATIC LOADING GUIDELINES

6.2.1 Seams and Butts In Plating

Although the title of this section implies seams and butts in continuous plating, it is not the intent of this section to exclude butts in web plates, flanges, etc.

Figure 6-1 depicts basic butt misalignment conditions as a function of type of loading.

The loadings presented in Figure 6-1 can be described as follows:

Loading 1 Describes a tensile or compressive load which this and other studies have found to create the most severe stress condition resulting from the eccentricity of the reactive or resistive forces, and the stress concentrations developed at the toe of the weld. Results indicate that initial yielding occurs in this area.

Loading 2 Indicates a pressure loading or some form of bending load. The effect of this loading on the joint is limited to the stress concentration occurring at the toe of.

1.) TENSION / COMPRESSION

2.) PRESSURE / BENDING

3.) SHEAR

FIGURE 6-1
CASE I: MISALIGNMENT
DETAILS FOR
SEAMS AND BUTTS
IN PLATING

the weld. As long as an adequate section modulus is maintained through the joint, its strength will depend on the magnitude of the stress concentration factor and not on the degree of misalignment. The stress concentration may be reduced by tapering the weld.

Loading 3 Depicts shear loading on the joint. For this case, the degree of misalignment is not a factor as long as the cross-sectional area in the weld is equivalent to that of the plate.

Figure 6-2 presents a set of curves that could be used to determine the misalignment limit for a butt joint. Four zones are shown which can be used to determine the ratio of average stress to allowable stress. Average stress is that at a distance from the joint and allowable stress is a limiting value based on judgment and experience.

Zone 4 is definitely unacceptable since it implies large plastic deformation and potential failure. Similarly Zone 1 is too conservative: An allowable stress of $0.6\sigma_y$ is acceptable as an average value, with the tacit understanding that some local increase in acceptable.

Acceptability criteria would be expected to center about the incipient yield line that separates Zones 2 and 3. In a potential application, static strength should be checked whether or not fatigue loading is a consideration.

Suitability for purpose must be considered: Primary hull structure such as deck or bottom plating should be designed and built to stricter alignment tolerances than deckhouse fronts or tween decks, for example.

6.2.2 Cruciform Joints In Plating

Figure 6-3 depicts the basic misaligned loading conditions corresponding to cruciform joints. As discussed in Section 6.2.1, load type No. 1 is the most severe and the only one specifically considered in this study.

Figure 6-4 presents curves similar to those in Figure 6-2. These curves can provide misalignment guidelines for fillet welded and full penetration welded joints with ratios of intercostal to through plate thickness of 1.0 and 2.0.

Potential application of data for the four zones in Figure 6-4 is guided by the same criteria of suitability for purpose discussed in Section 6.2.1.

6.2.3 Modified Cruciform Joints In Continuous Plating

A modified cruciform joint has one or both of its intercostal members entering the joint at an angle other than 90 degrees (Figure 6-5).

Figure 6-6 describes the perfect alignment case, where all plate centerlines are coincident.

Since the force F can be broken down into components, it is evident that the modified cruciform joint can then be related back to the cruciform joint described in Section 6.2.2.

6.2.4 General Categories

The potential for misalignment exists for any welded joint, including chocks, stanchions, bulkhead and deck stiffeners, etc. Evaluation of a misaligned joint may permit assignment to either of the basic categories treated earlier.

Figures 6-7 and 6-8 depict a few examples of joints with potential misalignment which are common in ship structure.

If a misaligned member is a component of a larger structural system it may deserve special consideration. For example, a single misaligned stiffener in a transverse bulkhead will cause a load redistribution to adjacent members which implies a less severe condition than if several stiffeners were misaligned.

6.3 CYCLIC LOADING GUIDELINES

Ship bottom, side shell and deck structural elements are subject to cyclic loading as a result of pressure, shear and bending moment variation resulting from passage through waves.

Limited experimental data on fatigue strength of misaligned joints have been empirically extended to provide a basis for acceptability guidelines.

6.3.1 Butts and Seams In Plating

Figure 6-9 presents extrapolated S-N curves for a stress ratio * $R = -1$ for butt welds with misalignment ratio $\frac{\delta}{t}$ of 0.0, 0.2, 0.4 and 0.6.

The load spectrum for a ship element is generally made up of a few cycles at large stress amplitude with ever increasing cycles at progressively lower stress amplitudes. By using the Palmgren-Miner cumulative fatigue damage criterion with a factor of safety of about 2.0, it is possible to establish the maximum permissible stress range (difference between maximum and minimum stress in a load cycle) for a given degree of misalignment. Those data are presented in Table 6-1.

* stress Ratio $R =$ algebraic ratio of minimum to maximum stress in a load cycle.

MISALIGNMENT OF STANCHIONS

MISALIGNMENT OF DECK GIRDERS
WITH SHELL FRAMES

MISALIGNMENT OF BULKHEAD
GIRDERS AND BACK-UP
STRUCTURE

FIGURE 6-8
MISALIGNMENT
DETAILS OF TYPICAL
SHIPBOARD
STRUCTURE

ET-9

FIGURE 6-9: S-N DIAGRAMS FOR MISALIGNED BUTT JOINTS: $R=-1.0$

TABLE 6-1

MISALIGNED BUTT WELDS	
δ/T	MAXIMUM STRESS RANGE ^(R=-1)
0.0	37200 PSI
0.2	29200 PSI
0.4	26000 PSI
0.6	15600 PSI

MAXIMUM CALCULATED STRESS
RANGES FOR MISALIGNED BUTT
WELDS EXPERIENCING CYCLIC LOADING

The maximum stress range can be approximated during design evaluation by calculating maximum and minimum stresses due to pressure, shear or bending moment based on any established quasistatic extreme wave height criterium, such as $L/20$ or $\frac{1}{10}\sqrt{L}$. Cyclic distributions other than those due to long term exposure to waves should be calculated by direct reference to the S-N diagrams of Figure 6-9, using the methodology discussed in Section 5.

6.3.2 Cruciform Joints

Figure 6-10 presents extrapolated S-N curves for a stress ratio $R = -1$ for cruciform joints with ~~5~~ misalignment ratios of 0.00, 0.25, 0.50 and 1.00.

By the method discussed in Section 6.3.1, maximum calculated stress ranges for cruciform joints subject to long term wave load distributions are shown in Table 6-2.

91-9

FIGURE 6-10: S-N DIAGRAMS FOR MISALIGNED CRUCIFORM JOINTS: $R = -1.0$

TABLE 6-2

MISALIGNED CRUCIFORM JOINTS	
δ/T_1	MAXIMUM STRESS RANGE ^(R=-1)
0.0	38400 PSI
0.25	32800 PSI
0.50	28000 PSI
1.0	24000 PSI

MAXIMUM CALCULATED STRESS
RANGES FOR MISALIGNED CRUCIFORM
JOINTS EXPERIENCING CYCLIC LOADING

Saction 7
SURVEY OF STRUCTURAL ALTERATIONS TO
CORRECT MISALIGNMENT

7.1 GENERAL

During ship construction structural alterations are sometimes required to correct construction errors and to accommodate design modifications made during the manufacturing process. These may include addition or deletion of temporary and permanent openings, reduction in scantlings, misalignment, unfairness, etc. These alterations and corrections must be made in a manner that meets the physical and operational objectives of the specification and the requirements of the regulatory agencies. It is in the best interests of the shipbuilder and the owner that these alterations be accomplished in a timely manner, using the most economical combination of material, labor and facilities, to produce a product that meets the above requirements.

The following sections provide the designer with a collection of corrective methods for misalignment errors that have exceeded permissible limits. No recommendations accompany this review.

7.2 MISALIGNMENT OF BUTTED MEMBERS

Misalignment of butted members may range from very large (200-ton) units to single flat bar stiffeners. When plating misalignment is to be corrected, the most often used method is to force the two plates into acceptable alignment by the use of jacks, wedges and weights and then weld the joint. To reduce the effect of stress concentrations, the weld is tapered or "battered" so that the toe angle is small. Misalignment of butts or seams in plating is generally allowed only over a small portion of the total joint length.

7.2.1 Flanges

Figure 7-1 depicts a misalignment condition between member flanges. Five methods for correcting this misalignment are presented. Methods 1, 2 and 3 involve the removal of a section of the web plate and possibly of the flange and the rewelding of a tailored section which provides a continuous load path for the flanges. Methods 4 and 5 show the inclusion of a flange reinforcing member which requires less work but conversely will introduce higher stress concentrations.

7.2.2 Web Plates

Figure 7-2 indicates three methods for correcting misalignment of butted web plates. These methods may be applied to tees, angles and flat bars. When misalignment between web plates occurs, the most common correction method is to cut the web plate free of the supporting member and force the webs into alignment. Method 2 presents an example of the correction with a recommended disengagement length of $50T$ where T = web plate thickness. When the misalignment is too great to force the webs into alignment it may be necessary to replace a portion of the member. Method 1 depicts the introduction of a skewed filler piece while Method 3 shows the addition of a web splice and a flange doubler plate. When the misaligned members carry large lateral loads and are skewed, knuckled or bent, it is recommended that a support member be installed as close to the knuckle as possible, see Figure 7-3 for details.

7.3 MISALIGNMENT OF CRUCIFORM JOINTS

Cruciform joints are the most susceptible to misalignment. This form of misalignment would appear to be a direct result of inaccuracies in fit-up and the inability of the shipfitter to properly locate back-up structure. This problem may range in severity from the single misaligned bulkhead stiffener to the complete misalignment of the machinery casing with its back-up structure.

MISALIGNMENT EXAMPLE

METHOD-1:
REMOVE SECTION OF
GIRDER, REPLACE WITH
TAILORED SECTION.

METHOD-2:
COPE OUT A SECTION
OF ONE OF THE MEMBERS,
REPLACE WITH A
TAILORED SECTION.

METHOD-3:
REMOVE A WEDGE-
SHAPED PIECE OF WEB
BEND FLANGE DOWN
REWELD.

METHOD-4:
REMOVE SECTION OF
FLANGE, REPLACE WITH
INSERT PLATE.

METHOD-5:
ADD NEW CHOCK TO
FLANGES.

FIGURE 7-1
STRUCTURAL ALTERATIONS
FOR
MISALIGNMENT OF
BUTTED FLANGES

7.3.1 Continuous Plating

Misaligned cruciform joints are normally corrected by increasing the leg of the fillet weld or by adding doubler plates. Figure 7-4 presents three methods by which misalignment may be corrected. Method 1 shows a weld increase, however this method is limited to small values of δ & since the larger the weld size the greater the opportunity for a poor weld and for excessive deformation.

Methods 2 and 3 employ doublers or sole plates to improve the error in alignment. An evaluation of the adequacy of these methods is not attempted.

7.3.2 Stiffeners, Beams and Stanchions

Figure 7-5 shows common methods for correcting alignment errors in the above structural elements.

7.4 MISALIGNMENT OF MODIFIED CRUCIFORM JOINTS

Figure 7-6 depicts a misaligned modified cruciform joint with a set of common corrective methods. Methods 1 through 3 have been discussed in Section 7.3.1, Methods 4 through 6 are similar with some degree of modification. This type of detail is used for foundations, miscellaneous bulkheads and where structure is added on and common reference points do not exist.

METHOD 1:
INCREASE LEG OF FILLET
WELD BY UP TO 20% OF
LEG SIZE.

METHOD 2:
ADDITION OF DOUBLER
PLATE OR FLAT BAR.

METHOD 3:
ADDITION OF DOUBLER
PLATE OR SOLE PLATE.

FIGURE 7-4
STRUCTURAL ALTERATIONS
MISALIGNMENT OF
CRUCIFORM
JOINTS
CONTINUOUS PLATING

Section 8
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APPENDIX A

CALCULATIONS FOR ELASTO-PLASTIC ANALYSIS OF MISALIGNED JOINTS

This Appendix Contains Computations Relative to
Section 4 - Results of Finite Element Analysis

APPENDIX A

PART A:

THIS APPENDIX CONTAINS THE COMPUTATIONS NECESSARY TO PERFORM THE PLASTIC ANALYSIS FOR THE BUTT WELDED JOINT WITH A 1.0T MISALIGNMENT. SEE FIGURE 4-6 FOR CORRESPONDING DETAILS:

LOAD CASE 1:

$$P = 4000 \text{ LB}$$

ELEMENT 74 & 79:

S_{xx}	S_{yy}	S_{xy}
149: -21441 PSI	-6034	8470
90: -8924 PSI	-6694	7400
96: -32754 PSI	-4768	4430
97: -18365 PSI	+1409	3794
102: -31714 PSI	-3049	640
103: -17854 PSI	+2751	-3049

$$\sigma_T = \sqrt{S_{xx}^2 + S_{yy}^2 - S_{xx} S_{yy} + 3S_{xy}^2}$$

149: -24124 PSI

90: 15132 PSI

96: 31600 PSI

97: 19234 PSI

102: 30325 PSI

103: 20083 PSI

NODE 96:

$$\sigma_T = 31600 \text{ PSI} < \sigma_{YIELD}$$

$P = 4000 \text{ LB}$ - ELASTIC RANGE:

YIELD LOAD:

$$P_Y = 4000 \times \sigma_T / 31600 = 4177 \text{ LB}$$

YIELDING OF NODE 96:

NODE 102:

YIELD LOAD: $P_Y = 4353 \text{ LB}$

ELEMENT 79 MUST THUS BE MODIFIED TO REFLECT ARTIFICIAL MATERIAL PROPERTIES.

ELEMENT 79 YIELDS SEE DETAIL A, FIGURE 4-8 ,
 $P = 4200 \text{ LB}$

ELEMENT 24,28,33:

S_{XX} :	S_{YY}	S_{XY}
28:-16080 PSI	2070 PSI	482 PSI
29:-27288 PSI	-2505 PSI	530 PSI
34:-13782 PSI	2063 PSI	3184 PSI
35:-28000 PSI	3739 PSI	3881 PSI

σ_T :

17229 PSI
 26141 PSI
 15907 PSI
 30787 PSI

YIELD LOAD: NODE 35:

$$P_Y = 4200 \times \sigma_Y / 30787 = 4500 \text{ LB}$$

YIELD LOAD: NODE 29: $P_Y = 5300 \text{ LB}$

ELEMENT 28 & 33: NODE 35 YIELDS AT $P = 4500 \text{ LB}$
SEE DETAIL B, FIGURE 4 - 8.

ELEMENT 28, 33: NODE 35

ARTIFICIAL MATERIAL PROPERTIES:

ELEMENTS: 74, 75, 79, 80, 84, 85:

LOAD CASE 3:

$P = 4600^{LB} + \text{INCREMENTAL LOADING}$

$P = 5300^{LB} / \text{NODE 29 YIELDS.}$

	SXX: PSI	SYY: PSI	SXY: PSI	σ_T : PSI
90	-18056	-1930	5663	19776
91	-13600	902	1707	14380
97	-30604	-5032	2568	28780
98	-20222	-3727	-11	18680
103	-28584	-1491	1251	27592
109	-27620	-2037	-4803	27928
149	-20721	-4003	7870	23415

PYIELD:

NODE 97: 6077^{LB}

" 103: 6337^{LB}

" 109: 6263^{LB}

REMOVE NODE 29:

SEE DETAIL D, FIGURE 4-8 .

LOAD CASE 4 :

$P = 5300^{LB} + \text{INCREMENTAL LOADING}$

AT $P = 6000^{LB}$

	S_{XX} : PSI	S_{YY} : PSI	S_{XY} : PSI	σ_T : PSI
97	-36623	-7158	2886	33990
98	-24188	-4696	76	22216
103	-34729	-2367	1333	33687
104	-18303	+ 53	1550	18500
109	-33382	-3399	- 4520	32767
110	-18478	- 231	4592	20010

YIELD LOADS:

$$P_{Y-97} = 5825^{LB}$$

$$P_{Y-103} = 5877^{LB}$$

$$P_{Y-109} = 6042^{LB}$$

SEE DETAIL E, FIGURE 4 - 8 .

LOAD CASE 5:

$P = 5800^{LB} + \text{INCREMENTAL LOADING}$

FAILURE OF SECTION AT 5875^{LB} .

SUBSTITUTE ARTIFICIAL LOADINGS TO MODEL
YIELDED ELEMENTS.

SEE DETAIL FIGURE 4-26 FOR LOAD CASE 2
RESPONSE.

NODE 72:

NODE: 84,90 :

NODE: 90,96,102 :

NODE: 102,108:

LOAD CASE 3:

$P=6350\text{ LB}$

PART B:

THIS SECTION CONTAINS THE COMPUTATIONS NECESSARY TO PERFORM THE PLASTIC ANALYSIS FOR THE FILLET WELDED CRUCIFORM JOINT WITH A 0.6T MISALIGNMENT.

SEE FIGURE 4-11 FOR CORRESPONDING DETAILS.

LOAD CASE I:

$$P = 5430 \text{ LB}$$

ELEMENTS: 54, 59, 64, 69, 74, 79, 84:

	Sxx	Syy	Sxy	σ
72	-28625	-3172	8028	30528
78	-34655	-3700	170	32962
84	-29637	-1938	118	28718
90	-29115	-1767	-73	28273
96	-28710	-1746	-71	27878
102	-27976	-1927	926	27111
108	-27522	-2211	4026	27388

YIELD LOAD: $P_Y = \frac{33000}{\sigma} \times 5430$

72	$P_Y = 5870 \text{ LB}$
78	$P_Y = 5436 \text{ LB}$
84	$P_Y = 6240 \text{ LB}$
90	$P_Y = 6338 \text{ LB}$
96	$P_Y = 6428 \text{ LB}$
102	$P_Y = 6611 \text{ LB}$
108	$P_Y = 6543 \text{ LB}$

NODE 78:

SEE DETAIL , FIGURE 4-26 FOR INITIAL YIELDING.

LOAD CASE 2:

$P = 5800 \text{ LB}$

ELEMENTS: 54, 59, 64, 69, 74, 79, 84:

	S_{xx}	S_{yy}	S_{xy}	σ
72	-30283	-3476	8586	32327
84	-31030	-1798	187	30173
90	-31013	-1858	71	30127
96	-30662	-1868	74	29844
102	-29862	-2060	495	28900
108	-29374	-2357	4295	29232

YIELD LOAD: $P_Y = 33000 \times 5800 / \sigma$

72	$P_Y = 5920 \text{ LB}$
84	$P_Y = 6343 \text{ LB}$
90	$P_Y = 6353 \text{ LB}$
96	$P_Y = 6413 \text{ LB}$
102	$P_Y = 6623 \text{ LB}$
108	$P_Y = 6548 \text{ LB}$

PART 2

FINAL TECHNICAL REPORT
ON
POTENTIAL SHIP STRUCTURE
TRIPPING BRACKET
GUIDELINES
PART NO. 2 OF
STANDARD
STRUCTURAL ARRANGEMENTS
TASK S-11
OF THE
SHIP PRODUCIBILITY PROGRAM

GENERAL DYNAMICS
QUINCY SHIPBUILDING DIVISION

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Section 1

INTRODUCTION AND SUMMARY

Differentiated from the unrestrained lateral-torsional buckling of beams, the lateral-torsional buckling failure mode of plate-stiffeners is by laying-over **or** tripping about an enforced axis of rotation (the stiffener-to-plate line of attachment). Like Euler buckling, tripping is a possible primary mode of failure which must be prevented. There are three basic design procedures to prevent tripping failure:

1. Use sections having sufficient torsional rigidity for unbraced span length.
2. Use intermediate lateral supports to reduce unbraced span lengths.
3. Use tripping brackets to reduce unbraced span lengths.

The objective of this study was **to** provide guidelines to establish the need for tripping brackets, their spacing and their configuration. Motivation for the study was a consensus among shipyard designers that this guidance was lacking, and that the costs incurred by arbitrary placement of large tripping brackets are excessive.

The guidelines are presented in summary form in Section 2, with application examples in Section 3. Tripping bracket configuration is briefly discussed in Section 4. Although the design guidelines are self-contained, maximum benefit and understanding will be obtained by reading the outline of analytical techniques employed in the study, which are presented in Section 5.

Review of the literature leads to the conclusion that researchers have preferred to reformulate existing solutions rather than build on the background of others. The compilation

in this study should help focus attention on those areas Of the tripping phenomenon most needing attention: a) behavior of asymmetrical stiffeners under lateral load (bending), and b) experimental verification of all formulations. Not only is the tripping problem for asymmetrical sections in bending unsolved, but there appears to be no accurate method for calculating the stress distribution in such a member other than finite element analysis. The latter is unacceptable for routine design work.

Section 2

TRIPPING INSTABILITY DESIGN GUIDELINES

2.1 AXIAL COMPRESSION

2.1.1 Symmetric Sections

Flat Bars

$$F_{Ta} = 0.38 E \left(\frac{t}{d} \right)_w^2 \quad (2-1)$$

Tees

F_{Ta} is the larger value of Equation (2-2) or (2-3)

$$F_{Ta} = \frac{0.35 A_f E}{I_{po}} \sqrt{\frac{b_f^2 d_w^2}{A_f (S_p/t_p^3 + 2 d_w/t_w^3)}} \quad (2-2)$$

$$F_{Ta} = \frac{0.80 A_f E}{I_{po}} \left(\frac{b_f d_w}{l} \right)^2 \quad (2-3)$$

2.1.2 Asymmetric Sections

Angles

F_{Ta} is the larger value of Equation (2-4) or (2-5)

$$F_{Ta} = \frac{0.50 A_f E}{I_{po}} \sqrt{\frac{b_f^2 d_w^2}{A_f (S_p/t_p^3 + 2 d_w/t_w^3)}} \quad (2-4)$$

$$F_{Ta} = \frac{2.5 A_f E}{I_{po}} \left(\frac{b_f d_w}{l} \right)^2 \quad (2-5)$$

2.2 LATERAL BENDING

2.2.1 Symmetric Sections

Flat Bars

$$F_{Tb} = \frac{0.13 E t_w}{S_x} \quad (2-6)$$

Tees

I_{lb} is the larger value of Equation (2-7) or (2-8)

$$F_{Tb} = \frac{0.17 A_f E}{d_w S_x} \sqrt{\frac{b_f^2 d_w^2}{A_f (S_p/t_p^3 + 2 d_w/t_w^3)}} \quad (2-7)$$

$$F_{Tb} = \frac{0.40 A_f E}{d_w S_x} \left(\frac{b_f d_w}{l} \right)^2 \quad (2-8)$$

2.3 FLANGES

$$F_{TF} = 0.60 E \sqrt{\left(\frac{b}{t} \right)_f \cdot \left(\frac{t}{d_w} \right)^3} \quad (2-9)$$

2.4 TRIPPING IN THE INELASTIC RANGE ($F_T > \frac{1}{2} F_Y$)

$$F_{Ti} = F_Y \left(1 - \frac{F_Y}{4 F_{Te}} \right) \quad (2-10)$$

2.5 COMBINED AXIAL COMPRESSION AND LATERAL BENDING

Symmetric Sections

$$\frac{f_a}{F_{Ta}} + \frac{f_b}{F_{Tb}} \leq 1.0 \quad (2-11)$$

2.6 TRIPPING BRACKET STIFFNESS AND STRENGTH

2.6.1 Required Stiffness

$$K_{REQ'D} = \frac{C_i F_{Ti} A}{l} \quad (2-12)$$

2.6.2 Required Strength

$$P_{BRKT} = \frac{C_i F_{Ti} A}{250} \quad (2-13)$$

Number of Spans	2	3	4	8	≥ 10
C_i	4.00	6.00	6.82	7.80	8.00

2.7 DESIGN FACTOR OF SAFETY

The tripping stresses calculated by using the design guidelines have a factor of safety of 1, i.e, they are the theoretical critical values. Rather than apply an arbitrary factor of safety, it is recommended to equate tripping stress to 1.00 to 1.15 times the proportional limit. Below the proportional limit the full strength of the section is not being utilized, while above the proportional limit the strength drops off rapidly.

2.8 NOMENCLATURE OF DESIGN GUIDELINES

The following definitions of symbols and terms are to be understood, in the absence of other specifications, where they appear in the design guideline.

A	Cross sectional area of stiffener
A_f	Cross sectional area of stiffener flange
$(b/t)_f$	Stiffener flange width to thickness ratio
b_f	Flange width
C_1	Span dependent constant
dw	Stiffener depth
E	Young's modulus
f_o	Axial stress
f_b	Bending stress
F_{1Q}	Axial tripping stress
F_{Tb}	Bending tripping stress
F_{Te}	Elastic tripping stress
F_{Ti}	Inelastic tripping stress
F_{TF}	Flange tripping stress
I_{Po}	Stiffener polar moment of inertia about toe
J	Length between lateral supports
Sp	Stiffener spacing
S_x	Section modulus of plate-stiffener combination to flange

t_p - Plate thickness

t_w Web thickness

$(t/d)_w$ Stiffener web thickness to depth ratio

Section 3
APPLICATION EXAMPLES OF DESIGN GUIDELINES

EXAMPLE 1

Determine the axial compressive tripping stress, F_{Ta} , for the flat bar shown in Figure 3-1. The unbraced span length is 172 inches and the material EH-36 steel, $F_y = 51$ ksi.

Figure 3-1

Solution

$$F_{Ta} = 0.38E \left(\frac{t}{d} \right)_w^2 \quad (2-1)$$

$$F_{Ta} = 0.38(29.5 \times 10^3) \left(\frac{2}{32} \right)^2 = 43.8 \text{ ksi}$$

$F_{Ta} > \frac{1}{2} F_y \therefore$ Correct for inelasticity

$$F_{Ti} = F_y \left(1 - \frac{F_y}{4F_{Ta}} \right) \quad (2-10)$$

$$F_{Ti} = 51 \left(1 - \frac{51}{4(43.8)} \right) = 36.2 \text{ ksi (Answer)}$$

EXAMPLE 2

Determine the axial compression tripping stress, F_{Ta} for the tee shown in Figure 3-2. The unbraced span length is 48 in. and the material mild steel, $F_y = 33$ ksi

Figure 3-2

Solution

$$F_{Ta} = \frac{0.3SA_f E}{I_{po}} \sqrt{\frac{b_f^2 d_w^2}{A_f (S_p^2 I_p^3 + 2d_w^2 I_w^3)}} \quad (2-2)$$

$$F_{Ta} = \frac{0.3 \times 4 \times (1.68) \times (29.5 \times 10^3)}{79.3} \sqrt{\frac{(3)^2 (5.87)^2}{1.68 (29^3 + 2(5.87^3 / 0.283)}} = 78.8 \text{ ksi}$$

$$F_{Ta} = \frac{0.80 A_f E}{I_{po}} \left(\frac{b_f d_w}{l} \right)^2 \quad (2-3)$$

$$F_{Ta} = \frac{0.80 (1.68) (29.5 \times 10^3)}{79.3} \left(\frac{3 (5.87)}{48} \right)^2 = 67.3 \text{ ksi}$$

$$F_{Ta} (78.8 \text{ ksi}) > \frac{1}{2} F_y \quad \therefore$$

$$F_{Ti} = F_y \left(1 - \frac{F_y}{4 F_{Ta}} \right) \quad (2-10)$$

$$F_{Ti} = 33 \left(1 - \frac{33}{4 (78.8)} \right) = 29.5 \text{ ksi} \quad (\text{Answer})$$

(During Testing (1) this grillage failed by stiffener tripping at 27.3 ksi.)

EXAMPLE 3

Determine the Axial Compression tripping stress, F_{Ta} , for the tee shown in Figure 3-3. The unbraced span length is 48 in. and the material mild steel, $F_y = 33 \text{ ksi}$.

Figure 3-3

(1) Smith, "Compressive Strength of Welded Steel Ship Grillages", RINA, 1975

Solution

$$F_{Ta} = \frac{0.80 A_f E}{I_{po}} \left(\frac{b_f d_w}{l} \right)^2 \quad (2-3)$$

$$F_{Ta} = \frac{0.80(1.13)(29.5 \times 10^3)}{292} \left(\frac{3(11.94)}{48} \right)^2 = 50.8 \text{ ksi}$$

Since Equation (2-3) is always conservative and $F_{Ta} > F_y$, Equation (2-2) need not be used.

$$F_{Ta}(50.8) > \frac{1}{2} F_y \quad \therefore \quad \text{Correct for inelasticity}$$

$$F_{Ti} = F_y \left(1 - \frac{F_y}{4F_{Ta}} \right) \quad (2-10)$$

$$F_{Ti} = 33 \left(1 - \frac{33}{4(50.8)} \right) = 27.6 \text{ ksi} \quad (\text{Answer})$$

(During testing ¹this grillage failed by stiffener tripping at 31.3 ksi.)

¹ Smith, "Compressive Strength of Welded Steel Ship Grillages", RINA, 1975

Section 4

TRIPPING BRACKET DETAILS

Figure 4-1 illustrates some common and uncommon tripping brackets used by the shipbuilding industry. A more comprehensive review is to be found in Reference ¹.

The single best guideline to tripping bracket design is that their effectiveness is no better than that of the structure to which it is attached. Tripping brackets are best when anchored against intersecting structural shapes and are least efficient when welded to an unstiffened plate field.

Figure 4-1 Tripping Bracket Details

Figure 4-1 Tripping Bracket Details (Cont'd)

Section 5
OUTLINE OF TECHNIQUES USED IN DEVELOPMENT OF GUIDELINES

5.1 INTRODUCTION AND GENERAL COMMENTS

This section is presented in the belief that the Designer/Engineer will benefit more from the design guidelines if he understands the methods used to derive them.

Section 5.2 is based on theoretical background presented in numerous elementary texts on instability. It is not felt necessary to repeat this information here. The assumption that the stiffener ends are simply supported, prevented from twisting, and free to warp is realistic for most ship structure.

Furthermore, for structure such as transverse web frames which are elastically restrained at their base it is possible that the buckle wave length is less than the span length. Thus the interior portion definitely behaves as if it were simply supported.

To avoid using a value of rotational restraint which would cause excessive bending of the stiffener web, K_0 is modified to include the bending stiffness of the stiffener web. Although this approach is not theoretically correct, it is felt to be better than placing an arbitrary upper limit on K_0 (plate.)

A plot of equation (5-1) is shown in Figure 5-1. As can be seen for span lengths less than l_{cr} , Equation (5-11) can underestimate F_{Ta} because it assumes the number of buckled half waves is a continuous instead of discrete function. In this event Equation (5-1) with $K_0=0$ may give a better estimate. In any case, Equation (5-13) is always the minimum value of F_{Ta} . (Assuming $K_0=0$ is equivalent to assuming the toe of the stiffener is pinned.)

Figure 5-1

The comments made for Section 5.2 similarly apply to Section 5.3. The potential energy function is presented in case a more accurate analysis is wanted when the moment is nonuniform, i.e., the transverse lateral load is not zero or the end moments are unequal. These conditions (variable coefficients) can be solved using the Raleigh-Ritz method. A method is not presented to calculate F_{Tb} for asymmetric shapes. Their use as inner bottom longitudinal and shell longitudinal in ballast tanks indicates the coupling effect is probably very slight.

A stiffener will not trip unless its flange is in compression. The unbraced compression length is considered to be the length of flange between inflection points.

Lateral buckling of flanges (Section 5.4) can not strictly be considered tripping. However, it is felt that when using the rotation restraint available at the stiffener toe this is a prudent check which should be made for deep sections having heavy flanges.

Although material presented in Section 5.5 is based on work done entirely on beams and columns, Equation (5-35) has been widely used on other structures, from spacecraft to ships. It is cautioned that no initial imperfection is explicitly accounted for by Equation (5-35).

Section 5.7 is based on the Euler buckling of columns. Its use should be very conservative when applied to bracing for stiffener tripping. The term F_{Ta} represents the total force acting on the stiffener cross section (not including plate). The stiffness required is measured at the flange.

5.2 TRIPPING DUE TO AXIAL COMPRESSION

5.2.1 Symmetric Stiffeners

The basic equation for tripping of a symmetric stiffener about an enforced axis of rotation is (Reference 2, p 140): .

$$F_{Ta} = \frac{E}{I_{po}} \left[\frac{J}{2.6} + \frac{\pi^2 M^2}{l^2} T_o + \frac{l^2}{\pi^2 m^2} \frac{K_o}{E} \right] \quad (5-1)$$

Where

- F_{Ta} = Axial compression tripping stress
- E = Young's modulus
- I_{po} = Stiffener polar moment of inertia about toe
- J = Torsional constant
- M = number of buckled half-waves
- T_o = warping constant
- Q = unbraced span length
- k_o = rotational restraint

The derivation of Equation (5-1) assumes the stiffener ends are simply supported and prevented from twisting, but the flange is free to warp. Local deformations of the stiffener are not considered, see Figure 5-2.

Figure 5-2

In Equation (5-1) the axial tripping stress F_{Ta} is a function of the number of buckled half-waves m . F_{Ta} is minimized if m is minimized, or:

$$\frac{dF_{Ta}}{dm} = 0 \quad (5-2)$$

Applying Equation (5-2) to Equation (5-1), the critical wave number is:

$$m = \sqrt[4]{\frac{K\phi l^4}{\pi^4 E T_o}} \quad (5-3)$$

Substituting Equation (5-3) into Equation (5-1) gives:

$$F_{Ta} = \frac{E}{I_{po}} \left[\frac{J}{2s^6} + \sqrt{\frac{T_o K\phi}{E}} \right] \quad (5-4)$$

Equation (5-4) is valid only if l is treated as a continuous function. This assumption is accurate enough for $m \geq 3$. If $m < 3$, Equation (5-1) should be used to determine F_{Ta} where m is the lower adjoining integer of Equation (5-3). The minimum value of m is 1.

The terms in Equation (5-1), except for the rotational restraint, are proportion of the stiffener cross-section, See Figure 5-3. The only terms which require further explanation are T_o and $K\phi$.

Figure 5-3

The warping constant T_0 is a property of the stiffener cross-section and of the location of the center of rotation relative to the shear center. It is analogous to the ordinary flexural and torsional constants, but has different dimensions. The equation for T_0 is:

$$T_0 = Cw + \alpha^2 I_y \quad (5-5)$$

Cw for various shapes can be determined from equations found in Reference 3. Since it is much smaller than the second term in Equation (5-5) for shipbuilding shapes, omitting it causes no appreciable error.

During tripping the rotation of the stiffener is resisted by the bending stiffness of the plate. If the plate is considered to be made up of beam strips, this restraint can be determined by the method shown in Figure 5-4. As can be seen from this Figure, the minimum restraint is for antisymmetric tripping.

Figure 5-4

The above analysis gives an exaggerated value of K_{ϕ} since bending of the web will occur during tripping. This can be corrected if the bending stiffness of the stiffener web is included in the calculation of K_{ϕ} .

Figure 5-5

Referring to Figure 5-5, the equation for the rotational, restraint including the web stiffness is:

$$\frac{1}{K_\phi} = \frac{1}{K_P} + \frac{1}{K_W} \quad (5-6)$$

or in terms of the plate and stiffener web:

$$K_\phi = \frac{E}{3(1-\nu^2)} \left[\frac{1}{\frac{b_P}{t_P^3} + \frac{2d_W}{t_W^3}} \right] \quad (5-7)$$

For flat bars the term $\sqrt{1-\nu^2} K_\phi$ is much smaller than $0.38J$. Thus, Equation (5-4) reduces to:

$$F_{T_a} = \frac{0.38 EJ}{I_{Po}} \quad (5.8)$$

or substituting the expressions for J and I_{Po}

$$F_{T_a} = 0.38E \left(\frac{t}{d} \right)_W^2 \quad (5-9)$$

Equation (5-9) is Equation (2-1) of the design guideline. It is interesting to note that Equation (5-9) is the local plate buckling equation for an infinitely long $(\frac{l}{d} > 5)$ uniformly compressed plate with one long edge simply sup

For tee's the term $\sqrt{\frac{T_0 K_0}{E}}$ is much larger than 0.38J. Thus Equation (5-4) reduces to:

$$F_{Ta} = \frac{2E}{I_{po}} \sqrt{\frac{T_0 K_0}{E}} \quad (5-10)$$

or substituting the expressions for T_0 and K_0 :

$$F_{Ta} = \frac{0.35 A_f E}{I_{po}} \sqrt{\frac{b_f^2 d_w^2}{A_f (S_p/3 + 2d_w/3)}} \quad (5-11)$$

Equation (5-11) is Equation (2-2) of the Design Guideline.

When m from Equation (5-3) is a noninteger less than 3, it is possible that Equation (5-11) significantly underestimates F_{Ta} . In this event a better estimate of F_{Ta} may sometimes be made using Equation (5-1) with $m=1$ and $K_0=0$. The term 0.38J which is small compared to $\frac{\pi^2 I_0}{l^2}$ may be neglected. Thus Equation (5-1) reduces to:

$$F_{Ta} = \frac{\pi^2 E I_0}{I_{po} l^2} \quad (5-12)$$

or substituting the expression for I_0 :

$$F_{Ta} = \frac{0.8 A_f E}{I_{po}} \left(\frac{b_f d_w}{l} \right)^2 \quad (5-13)$$

Equation (5-13) is Equation (2-3) of the design guideline.

5.2.2 Asymmetric Stiffeners

Because asymmetric stiffeners such as angles lack an axis of symmetry perpendicular to the plate, flexure and torsion are coupled and therefore, the axial buckling and tripping stresses are coupled. The coupled axial buckling-tripping stress, which

is always smaller than either the individual axial buckling or tripping stress, can be calculated from the following equation (Reference 4):

$$F_{Ca} = \frac{F_E + F_{Ta}}{2(1 - \xi^2)} \left[1 - \sqrt{\frac{1 - 4F_E F_{Ta}(1 - \xi^2)}{(F_E + F_{Ta})^2}} \right] \quad (5-14)$$

Where F_{Ca} = Coupled axial buckling-tripping stress

F_E = Euler plate stiffener buckling stress

F_{Ta} = Axial compression tripping stress [Equation (5-1)]

ξ = Coupling parameter = $\frac{X_o - X_s}{r_o}$

X_o = Coordinate of plate stiffener shear center from stiffener web

X_s = Coordinate of plate stiffener centroid from stiffener web

r_o = Polar radius of gyration = $\sqrt{\frac{I_{po}}{A_{p-s}}}$

I_{po} = Polar moment of inertia of stiffener about toe

A_{p-s} = Plate stiffener cross-sectional area

Provided Euler buckling of the plate stiffener does not govern, i.e. $F_E > F_{Ca}$ can conservatively be taken as 75 percent of F_{Ta} . Equation (2-4) and (2-5) of the design guideline are thus based on 75 percent of Equation (5-10) and (5-12) respectively, with the appropriate expressions substituted for r_o and K_o . It should be noted that for the same size flange, an angle resists tripping due to axial compression better than a tee.

5.3 TRIPPING DUE TO LATERAL LOADING

5.3.1 Symmetric Stiffeners

The potential energy function for lateral buckling of a stiffener about an enforced axis of rotation is (Reference 2, p.164):

$$\Phi = \frac{1}{2} \int_0^l (E I_o \ddot{\phi}^2 + G J \dot{\phi}^2 + 2 a M \ddot{\phi} \phi - a w_y \phi + K_o \phi^2) dz \quad (5-15)$$

The following differential equation is derived from Equation (5-15) by applying the calculus of variations:

$$0 = E\Gamma_0 \phi^{IV} - (GJ - 2\alpha M) \phi'' - (\alpha w_y - K\phi) \phi \quad (5-16)$$

Where

- E = Young's modulus
- Γ_0 = Warping constant
- G = Shear modulus
- J = Torsional constant
- α = Distance from shear center to center of rotation
- M = Simple **beam moment** due to externally applied forces
- α = Distance transverse load is applied above shear center
- w_y = Transverse distributed load
- $K\phi$ = Rotational restraint

If the stiffener is subjected to a uniform moment only, i.e., the lateral load is zero, the solution to Equation (5-16) is:

$$\phi = A_1 \sin \frac{n\pi z}{l} \quad (5-17)$$

Substituting Equation (5-17) into Equation (5-16) and simplifying gives:

$$M_{CR} = \frac{1}{2\alpha} \left[E\Gamma_0 \frac{n^2 \pi^2}{l^2} + GJ + K\phi \frac{l^2}{n^2 \pi^2} \right] \quad (5-18)$$

Equation (5-18) can be expressed in terms of stress if it is assumed:

$$F_{TB} = \frac{M_{CR}}{S_{FLG}} \quad (5-19)$$

Where S_{FLG} = section modulus to the flange of the stiffener plate combination.

Using the same minimization procedure outlined in Section 5.2.1, the equation for the critical wave number is

$$m = \sqrt[4]{\frac{K_{\phi} l^4}{E I_G \pi^4}}$$

substituting Equation (5-20) into Equation (5-19) gives:

$$F_{Tb} = \frac{E}{\alpha S_{FLG}} \left[\frac{J}{S_{\phi}^2} + \sqrt{\frac{I_G K_{\phi}}{E}} \right] \quad (5-21)$$

Similar to Equation (5-4), Equation (5-21) is valid for $m \geq 3$. If $m < 3$, then the lower adjoining integer from Equation (5-20) should be used in Equation (5-19) to determine F_{Tb} .

Although Equation (5-21) was developed for uniform moment, it can conservatively be used for other loading conditions.

For flat bars the term $\sqrt{\frac{I_G K_{\phi}}{E}}$ is much smaller than $0.19J$. Thus Equation (5.3) reduces to:

$$F_{Tb} = \frac{EJ}{S_{\phi}^2 \alpha S_{FLG}} \quad (5-22)$$

or substituting the expression for J and noting $\alpha = \frac{dw}{2}$

$$F_{Tb} = \frac{0.13 E t_w^3}{S_{FLG}} \quad (5-23)$$

Equation 5.3 is Equation 2.1 of the design guideline.

For lee's the term $\sqrt{\frac{\Gamma_o K \phi}{E}}$ is much larger than 0.19J
Thus Equation 5-21 reduces to:

$$F_{Tb} = \frac{E}{\alpha S_{FLG}} \sqrt{\frac{\Gamma_o K \phi}{E}} \quad (5-24)$$

or substituting the expression for Γ_o and $K \phi$ and noting $\alpha = d_w$

$$F_{Tb} = \frac{0.17 A_f E}{d_w S_{FLG}} \sqrt{\frac{b_c^2 - d_w^2}{A_f \left(\frac{S_{p,3}}{t_p} + 2 \frac{d_w}{t_w} \right)}} \quad (5-25)$$

Equation (5-25) is Equation (2-7) of the Design Guideline.

When m from Equation (5-20) is a noninteger less than 3, it is possible that Equation (5-25) significantly underestimates F_{Tb} . In that event, a better estimate of F_{Tb} may sometimes be made by using Equation (5-18) with $m=1$ and $K \phi = 0$. The term 0.19J which is small compared to $\frac{\pi^2 \Gamma_o}{2 \alpha S_{FLG} l^2}$ may be neglected. Thus Equation (5-18) reduces to:

$$F_{Tb} = \frac{\pi^2 E \Gamma_o}{2 \alpha S_{FLG} l^2} \quad (5-26)$$

or substituting the expression for Γ_o and again noting $\alpha = d_w$

$$F_{Tb} = \frac{0.40 A_f E}{d_w S_{FLG}} \sqrt{\frac{b_c^2 - d_w^2}{l^2}} \quad (5-27)$$

Equation (5-27) is Equation (2-8) of the design guideline.

5.3.2 Asymmetric Stiffeners

Tripping of asymmetric stiffeners due to lateral load is analogous to axial buckling of an eccentrically loaded column. The asymmetric stiffener begins to twist (trip) as soon as lateral load is applied due to the eccentricity of the shear center to the web, See Figure 5-6. At first the twisting

increases slowly, but as f_b approaches F_{cb} (F_{cb} = the coupled lateral bending tripping stress) the twisting increases more and more rapidly until at $f_b = F_{cb}$ the twist increases without bound. The larger the eccentricity of the shear center, the sooner infinite twisting is approached see Figure 5-7.

Figure 5-6

Figure 5-7

An analytical method has not yet been developed to calculate F_{cb} . Reference 5 suggests using the Bryan plate buckling stress of the flange as a stability criteria. Reference 6 states that for the loading condition shown in Figure 5-8 the local buckling failure mode is dominant over the tripping failure mode.

Figure 5-8

This report does not present a method to determine the tripping stability of asymmetric shapes under lateral load. It is recommended that pending further work in this area, the existing classification societies' rules be followed in this area.

Note that members requiring the application of large and costly tripping brackets, such as deep web frames, are generally symmetric sections for which guidelines have been formulated.

5.4 LATERAL BUCKLING OF FLANGES

Stiffeners having deep webs and heavy flanges may trip due to Euler buckling of the flange about its strong axis, See Figure 5-9. This behavior is similar to that of a beam on an elastic foundation where the flange is the beam and the web is the elastic foundation. The buckling equation is (Reference 7, p. 140):

$$F_T = \frac{m^2 \pi^2 E I_y}{A_f l^2} + \frac{K_w l^2}{m^2 \pi^2 A_f} \quad (5-28)$$

Figure 5-9)

where	m = Number of buckled half waves
	E = Young's modulus
	I _y = Flange moment of inertia about web
	A _f = Flange area
	l = Span
	K _w = Elastic spring constant

Using the same minimization procedure outlined in Section 5.2.1, the equation for the critical wave number is:

$$m = \sqrt{\frac{K_w I_y}{\pi^4 E I_y}} \quad (5-29)$$

Substituting Equation (5-29) into Equation (5-28) gives

$$F_T = 2E \sqrt{\frac{K_w I_y}{E A_f^2}} \quad (5-30)$$

If the web is assumed to be a guided cantilever, the elastic spring constant is

$$K_w = \frac{E t_w^3}{0.91 d_w^3} \quad (5-31)$$

Substituting Equation (5-31) into Equation (5-30) and simplifying gives

$$F_T = 0.6 E \sqrt{(b/t)_f \cdot (t/d)_w^3} \quad (5-32)$$

Equation (5-32) is Equation (2-9) of the Design Guidelines

5.5 INELASTIC RANGE

5.5.1 Axial Compression

The equations in Section 5.2 can be used to cover inelastic range tripping by substituting the tangent modulus for Young's Modulus when the calculated tripping stress is above the proportional limit. This method requires trial and error to find the calculated tripping stress which corresponds to the assumed tripping stress used to select the tangent modulus.

To accomplish this procedure for steel columns in the inelastic range without iteration, Bleich proposed the following parabolic equation:

$$F_e' = F_y - (F_y - F_p) \frac{F_p}{F_e} \quad (5-33)$$

Where F_e' = Inelastic range column buckling stress
 F_y = Material yield point
 F_p = Material proportional limit
 F_e = Euler column buckling stress

Column buckling tests have since shown the stress-strain curve departs from linearity below the proportional limit due to residual stresses. This led the Column Research Council (CRC) to replace F_p in Equation (5-33) by

$$F_p' = F_y - F_R \quad (5-34)$$

Where F_p' = Shape proportional limit
 F_R = Residual stress

Although the maximum residual compressive stress in the flange of hot-rolled steel shapes is approximately $0.3 F_y$, consideration of its effect on both weak and strong axis buckling caused the CRC to select F_R equal to $0.5 F_y$ thus

$$F_e' = F_y \left(1 - \frac{F_y}{4 F_e} \right) \quad (5-35)$$

In reference 8 Det Norske Veritas uses Equation (5-35) to calculate the inelastic range stability of ship plate stiffeners for both Euler buckling and tripping. Similarly, Equation (5-35) has been adopted by this report. It is cautioned, however, that if the residual stresses in the

stiffener flange are more adverse than the residual stresses shown in Figure 5-10 for a hot-rolled shape, Equation (5-35) will be unconservative.

Figure 5-10

5.5.2 Lateral Bending

The stress due to lateral bending varies across the stiffener cross-section. Thus if the tripping stress is in the inelastic range, the tangent modulus will also vary across the stiffener cross-section. This makes the analysis too complicated for a rational solution. Fortunately, Equation (5-35) agrees with test results for the lateral buckling of beams (Reference 9). Therefore, Equation (5-35) has also been adopted for inelastic tripping due to lateral bending by this report.

5.6 COMBINED AXIAL COMPRESSION AND LATERAL BENDING

5.6.1 Symmetric Stiffeners

The interaction between axial and lateral tripping for symmetric stiffeners is a very complex phenomenon. A sufficient, although not necessary, condition of stability is (Reference 10, p 739):

$$\frac{f_a}{F_{Ta}} + \frac{f_b}{F_{Tb}} \leq 1.0 \quad (5-36)$$

Equation (5-36) is the Design Guideline interaction equation for symmetric stiffeners.

5.6.2 Asymmetric Stiffeners

Pending development of F_{cb} no interaction equation is proposed for asymmetric stiffeners.

5.7 BRACING

5.7.1 Required Stiffness

The following method for calculating the minimum stiffness required for tripping brackets to be fully effective was first suggested by Winter for the lateral bracing of columns. A summary of Winter's work on lateral bracing requirements is contained in Reference 11.

Figure 5-11

span shown in Figure 5-n(a). If the stiffness is initially zero, the column will buckle in one half-wave when the load reaches $\frac{EI}{G(2l)^2}$, see Figure 5-n(b). As the lateral bracing stiffness is increased, the load required to buckle the column also increases until, **at a certain stiffness, the load reaches**

$\frac{\pi^2 EI}{(l^2)}$ and the column buckles in two half-waves, Figure 5-11(C). Increasing the lateral bracing stiffness beyond this value causes no further increase in critical load, Figure 5-n(d). The minimum lateral bracing stiffness necessary for the column to reach its maximum strength for the unbraced span length can be calculated by assuming an infinitesimally small lateral displacement is superimposed upon the antisymmetric buckled shape of Figure 5-11(c), see Figure 5-11(e). Summing moments about midspan (Point "B") for the upper half of the column (B C) gives

$$0 = \frac{F_x}{2} \cdot l - P_{CR} \cdot d \quad (5-37)$$

Noting stiffness is force per unit displacement, Equation (5-37) can be rearranged to give

$$K'_{ld} = \frac{2P_{CR}}{l} \quad (5-38)$$

Equation (5-38) applies only for two spans. It can be made more general by introducing a span dependent coefficient. Thus

$$K'_{ld} = \alpha \cdot \frac{P_{CR}}{l} \quad (5-39)$$

A curve for α is presented in Reference 11 is shown in Figure 5-12.

Figure 5-12

Because of imperfections, real columns begin deflecting at the onset of loading. The effect this has on required bracing stiffness can be determined by again considering the two span column shown in Figure 5-11(a). Now instead of being straight, assume the column has the initial imperfection form shown in Figure 5-13. The equilibrium equation at incipient buckling becomes

$$0 = \frac{F_x l}{2} - P_{CR}(d + d_0) \quad (5-40)$$

or

Figure 5-13

$$K_{ACT} = K_{id} \left(1 + \frac{d_0}{d}\right) \quad (5-41)$$

a more useful form of Equation (5-41) is

$$d = \frac{d_0}{\left(\frac{K_{ACT}}{K_{id}} - 1\right)} \quad (5-42)$$

Equation (5-42) shows that the actual bracing stiffness must be greater than the ideal bracing stiffness. If K_{ACT} equals K_{id} the deflection becomes infinite which is equivalent to buckling. Winter suggests K_{ACT} be twice K_{id} to keep the deflections prior to buckling acceptable.

The Design Guideline uses Equation (5-39) to determine the required bracing stiffness to prevent tripping. The values of α have been doubled in the Design Guideline to account for imperfections.

5.7.2 Required Strength

The force exerted on the tripping bracket at buckling is

$K_{ACT} \cdot d$ thus from Equation (5-42)

$$F_{REQ'D} = d \cdot \left(\frac{K_{ID}}{1 - \frac{K_{ID}}{K_{ACT}}} \right) \quad (5-43)$$

and for K_{ACT} equal to $2K_{ID}$

$$F_{REQ'D} = d \cdot \left(\frac{2\alpha P_{CR}}{1} \right) \quad (5-44)$$

Assuming a maximum tolerance for d is $\frac{1}{250}$, Equation (5-44) becomes

$$F_{REQ'D} = \frac{2\alpha P_{CR}}{250} \quad (5-45)$$

Equation (5-45) is used in the Design Guideline to determine . tripping bracket strength. The factor of 2 is contained in the design chart for α .

Section 6

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FINAL TECHNICAL REPORT
ON
POTENTIAL SHIP STRUCTURE
STRUCTURAL DETAILS
GUIDELINES

PART NO. 3 OF

STANDARD
STRUCTURAL ARRANGEMENTS
TASK S-11
OF THE
SHIP PRODUCIBILITY PROGRAM

GENERAL DYNAMICS
QUINCY SHIPBUILDING DIVISION

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Section 1
INTRODUCTION AND SUMMARY

In the last few years there has been a belated recognition of the importance of structural details in the design and performance of ship structure. At one end is an extensive collection of commonly used details (Reference 1) , while at the other there is a massive and very useful survey of in-service performance of details (Reference 2) . Several technical papers, for example References 3 and 4, discuss experimental and analytical results for specific detail configurations, and Committee 3a of the International Ship Structures Congress was created in time for the 1976 session with specific responsibility in this area.

The purpose of the present study is part of an effort to formulate standards for ship construction in an attempt to reduce building costs. Structural details, perhaps more than any other component of hull structure, lend themselves to standardization because attempts in that direction are already made by many shipyards on an individual and on a ship by ship basis.

Cost reduction due to eventual standardization of details is less related to reduced cost of construction than to enhanced ship performance, such as reduced scantlings of load carrying members and decreased requirements for maintenance in repair.

The industry now lacks the background data for an unequivocal selection of optimum details, but could use what information is currently available in an attempt to reduce diversity.

Section 2 is a general critical discussion of structural details that forms the basis for the tentative collection of standard details in Section 4.

Section 3 presents data on finite element analysis of clearance cuts with various collar and support arrangements that are useful in providing guidance for the selection of perhaps the most frequently used structural detail.

Reducing the incredible variety of any type of detail will lead to clearer performance evaluation of specific design details than is possible with today's proliferation. The feedback from such preliminary standardization would then be directly applicable to focusing analytical and experimental investigations, to providing numerical design guidelines, and to promulgating more definitive standards.

Section 2
REVIEW OF STRUCTURAL DETAILS

2.1 GENERAL

The definition of requirements and the cooperation of the United States shipbuilding industry was sought through visits to most major United States shipyards (Reference 5). The importance of attempts to standardize structural details . was well recognized, and many shipyards generously assisted by providing plans or booklets of their structural details.

All details were cut-out, numbered and attached, by category, to "wallpaper". The "wallpaper" was discussed in sessions attended by no less than two structural engineers and two structural designers. The objective was to single out good and bad individual features or complete details. Criteria for the discussions were:

1. Strength (potential for failure, history of damage)
2. Installation (ease of fitup)
3. Complexity (cost)
- 4 . Accessibility (welding, painting, maintenance)

Grouping of details was by commonly used categories. These categories and the number of samples available for review are listed below:

<u>Category</u>	<u>Number</u>
Cutouts and Collars	105
Miscellaneous Cutouts	96
Patches	7
Stanchion End Connections	57
Face Plates	18
Stiffener End Connections	64
Chocks	42

<u>Category</u>	<u>Number</u>
Panel Stiffeners	50
Beam Brackets	79

For discussion purposes these details have been grouped into the following more general categories:

Structural intersections (cutouts and collars)
Miscellaneous cutouts (snipes, limber holes,
penetrations, patches, oil stops and water
stops)
Brackets and chocks
Stiffener endings and transitions
Panel stiffeners
Stanchion end connections

In the following paragraphs the discussion is limited to broad general comments and to remarks on specific good and bad features of actual details encountered in the survey. The application of this discussion is made in the selection and design of those details to be found in Section 4.

2.2 STRUCTURAL INTERSECTIONS

The commonly used terminology for types of structural intersections is cutouts and collars, which would be better described as free and collared (or lugged) structural intersections. No such redefinition is attempted here for this or any other category of detail.

The necessary variety in this type of detail is great because of the shape variations in penetrating members (tees, angles, flat bars, bulb flats, etc.) and because of the degree of support required by the penetrating member.

Cutouts are provided to permit stiffeners or girders to pass through deeper structural members such as webs, bulkheads and decks. They may also serve the secondary purpose of providing drainage or venting openings in non-tight members. Where necessary, these cutouts are partially or fully collared to increase strength or restore tightness.

The simple clearance cut shown in Figure 2-1 is typical of where the penetrated member, in this case a bulkhead, is not a support point for the stiffeners.

Figure 2-1

At the other extreme is Figure 2-2, representing a clearance cut with a heavily loaded stiffener. Shear load is taken out by lugs or collars to both the heel and the bosom of the stiffener, and by a panel stiffener lapped over the web of the stiffener.

Figure 2-2

A modification of this detail is shown in Figure 2-3. Here the substitution of a lapped collar to the heel of the stiffener simplifies fitup of parts during assembly or erection.

Figure 2-3

All the most commonly used poor features in clearance cuts are illustrated by Figure 2-4.

Figure 2-4

- a. If the stiffener exceeds the depth of the opening that opening must be trimmed, probably by hand, possibly introducing notches (stress risers) .
- b. The sharp radii in the cutout have produced service cracks.
- c. The small size of the snipe at the base of the penetrated member creates difficult access for welding and may introduce corrosion and maintenance problems.

- d. Welding past the web of the stiffener may introduce undercut at the edge of the stiffener, resulting in a notch effect at a highly stressed location.

Desirable features of cutouts are illustrated by Figure 2-5.

Figure 2-5

- a. Alignment or dimensional errors can be corrected with relative ease by straight line trimming of the tabs.
- b. Large radius of the snipe and flange cutouts provides better access for welding.
- c. Notches produced by welding of the tabs to the stiffener are more nearly in line with the direction of applied stress than normal to it, therefore reducing the possibility of crack initiation and propagation.
- d. Short tangency extensions to the snipe cut permit modest trimming without introducing acute angles or disturbing accessibility.

Additional comments relative to collars (tabs) and snipes will be found in the paragraphs on stiffener endings and miscellaneous cutouts. It should be noted that more complex yet favorable cutout geometries create little difficulty for the modern shipyard equipped with numerically controlled flame cutting. In fact, the complex cutout is somewhat of a guarantee that the correct tool will be used and that casual manual burning will be avoided.

2.3 MISCELLANEOUS CUTOUTS

2.3.1 Penetrations

The one-sided pipe coaming in Figure 2-6 is commonly used in the machinery space. The inability to seal weld the interior permits trapping moisture and can lead to corrosion. Although a penetrating coaming is preferred, it is recognized that the advanced planning it requires is not always practical.

Figure 2-6

2.3.2 Temporary Openings and Patches

It is good shipyard practice to minimize temporary openings. Provision for access should be provided in the design stage by optimum use of lightening holes, with special attention to the position of structural units during subassembly.

Round openings are preferred, with oval openings sized to OSHA requirements used when required for personnel access.

Lapped patches over temporary openings should be used only in areas of low stress. Their use generally requires approval by the resident inspector. Flush patches with an integral backing bar (Figure 2-7) must be used when only one side is accessible for welding. The unwelded backing bar edge is susceptible to corrosion.

Figure 2-7

2.3.3 Miscellaneous openings

There is great diversity in the configuration of snipes drain holes, etc. , and it is difficult to rank one detail over another without specific consideration of purpose and location.

Round holes are generally preferred over half-round and oval holes, i.e. Figure 2-8a is favored over Figure 2-8b. Small radius cutouts as illustrated by Figure 2-9 should be avoided: coating and maintenance is difficult, and they obviously produce higher stress concentration. To minimize the effects of stress concentration, small radius cuts are better drilled than burned. Figure 2-10 shows an opening close to a structural intersection that should generally be avoided.

Figure 2-8a

Figure 2-8b

Figure 2-9

Figure 2-10

The elliptical cutout in Figure 2-11 provides greater drainage area, a lower "drain" and less stress concentration than that in Figure 2-12.

Figure 2-11

Figure 2-12

Straight snipes as shown in Figure 2-13 are very common but less desirable than rounded cuts. Welding at the edges, particularly if a weld wrap is required, does not lead to a very sound weld.

Figure 2-13.

Figure 2-14 shows the extension of a snipe to accommodate an adjacent seam or butt. Partial collars (Figure 2-15) or full collars (Figure 2-16) may be necessary to restore strength in heavily loaded areas.

Figure 2-14

Figure 2-15

Figure 2-16

2.4 BRACKETS AND CHOCKS

2.4.1 Beam Brackets

Beam brackets, not knee brackets, are generally treated as structural details. These brackets serve to transfer end moments and shear into adjacent structure, and to reduce the unsupported span of a beam.

The standardization of beam brackets is nearly impossible because of the diversity their function calls for. An extreme case is mentioned in Reference 2, where 81 geometrical forms of beam brackets were observed on a single ship.

For coplanar angle or flat bar stiffeners, a lapped bracket is preferred to a butted bracket. Brackets with sniped flange are used where the bracket edge is long and where loads are relatively high. When higher loading requires a continuous bracket (Figure 2-17), tangency and flange chocks should be fitted to the intersecting structural members. Without these chocks the knife-edge arrangement can result in cracks.

Figure 2-17

2.4.2 Chocks

Chocks are used as reinforcement or backup in areas of concentrated force; as intermediate load carriers in areas of structural discontinuity and as a means of stabilizing flanges in structural elements. Tangency chocks are generally the same thickness as the web of the girder to which they are attached, and flange chocks are of similar thickness to that of the flange to which they are welded.

Sniping the corner of a chock is a good design feature which allows the weld to be easily wrapped. If the angle of the chock edge is steep enough (e.g. 60 degrees) sniped corners are not necessary. The snipe should be about 1/2 inch.

Some snipes of smaller dimensions (Figures 2-18a and 2-18b) do not provide enough material for a good weld wrap and tend to "burn up".

Small interior snipes as shown in Figure 2-19 simplify fit-up of the chock, but make welding more difficult and create a difficult area to access for cleaning, painting and maintenance. Interior snipes should provide enough of a cut-away to reduce these difficulties or be eliminated altogether.

Figure 2-18a

Figure 2-18b

Figure 2-19

2.5 STIFFENER ENDINGS AND TRANSITIONS

Three degrees of stiffener end fixity are recognized by the American Bureau of Shipping in establishing section modulus requirements. Efficient brackets provide fixed end support, clips provide intermediate support, and no end attachment corresponds to a pinned connection.

Fixed end connections are typically attempted with angles or tees as stiffeners, and free ended stiffeners are often flat bars.

When web and flange of a stiffener are butted against adjacent structure to create a fixed end condition, it is common to use flange chocks, as shown in Figure 2-20, to reduce stress concentration.

Figure 2-20

Lapped end connections and clips are credited with similar end fixity. The lapped connection provides greater lateral stability, but requires stiffeners on opposite sides of the molded trace. Typical overlaps are in the range of 2-2 1/2 inches, and are illustrated in Figure 2-21. The snipe detail shown in Figure 2-21b allows better weld access and continuity than that in Figure 2-21a.

Figure 2-21a

Figure 2-21b

Clips are generally made at least as thick as the web of the stiffener to which they are attached. Increasing the clip scantlings allows it to carry the full stiffener-induced load.

A clipped connection should allow a gap between stiffener end and adjacent structure large enough to permit good welding and painting access.

Stiffener endings assumed to be free ended may have much of their section cut away (Figure 2-22) or have no end attachment (Figures 2-23, 2-24 and 2-25). A free stiffener end cut square is more difficult to weld than one that is sniped and introduces larger end stress concentrations. The gap for an unattached stiffener end should be large enough to allow access for welding. Figure 2-25 illustrates a better arrangement than Figure 2-24, although the snipe in the former is considered too complex for *commercial* practice. The corner of Figure 2-25 should be sniped to a depth greater than 1/8 inch for satisfactory welding. One-half inch is a reasonable minimum.

Figure 2-26 illustrates a producibility problem when it is required to wrap the weld against the angle bosom.

Figure 2-22

Figure 2-23

Figure 2-24

Figure 2-25

Figure 2-26

Clip ended stiffeners as shown in Figure 2-27 are often used without backup structure, giving rise to potential "punch through". The clip on the angle stiffener in Figure 2-27a overlaps the flange, and because of greater resulting eccentricity reduces the reaction force on the plating. On occasion the hardspot is relieved by fitting a sole plate as shown in Figure 2-28.

Figure 2-27a

Figure 2-27b

Figure 2-28

Transition pieces can also fall into the category of face plates. Their purpose is to provide gradual transition between structural members of different width (Figure 2-29) and depth, or to run out stiffeners, say longitudinal at the ends of the ship (Figure 2-30).

Figure 2-30

2.6 PANEL STIFFENERS

Panel stiffeners are used as local or intermediate deck structural reinforcement, as stiffeners intended to prevent buckling of deep webs, and as support to members penetrating these webs. Panel stiffeners are generally rolled angles or flat bars, whose section properties are specified by the Classification Societies for any given application.

The distinguishing features of types of panel stiffeners are their method of end connection, generally discussed in Section 2-5. Sniped corners, particularly for flat bar panel stiffeners as shown in Figure 2-31, allow better material utilization and better access for welding. The desirability of about a 1/2 inch land for better wrapping was discussed earlier.

Angle or tee panel stiffeners not connected to intersecting structure are commonly cut square as shown in Figure 2-32. In the presence of a lateral load this arrangement results in high stresses at the end connection. If lateral load is present, the panel stiffeners can be lapped as shown in Figure 2-33b, where the cutaway at the left clears the weld of the intersecting member. That weld will interfere with the detail as shown in Figure 2-33a.

Figure 2-31

Figure 2-32

Figure 2-33a

Figure 2-33b

2.7 STANCHION END CONNECTIONS

Stanchion end connections provide for transition between generally dissimilar structural members.

Although stanchions are normally viewed as compressive members (columns), they may also be loaded in tension, particularly at the ship's bow. This results from section deformation due to large lateral pressures. Therefore the stanchion end connection should have the cross sectional area and section properties required by its function. Generally those properties should be equivalent to those of the stanchion itself.

It is simpler to design end connections for an H or I-beam stanchion than for a pipe stanchion, although the required orientation of the former in a particular application may make alignment with deck girders difficult.

It is common practice to provide a sole piece for pipe stanchions whose wall thickness exceeds the thickness of the plate to which they are attached. A particularly effective, although admittedly complex, connection for pipe stanchions is shown in Figure 2-34. It has high tensile capability **and is very compact.**

Figure 2-34

The I-beam connection in Figure 2-35 provides good alignment with two intersecting members, and is therefore better than that shown in Figure 2-36 which requires additional chocks. The load capacity of supporting girders must be considered. Clearance cuts in close proximity to stanchion end connections are often provided with a flush full collar to increase strength. Figure 2-37 illustrates a connection that is difficult to fabricate. The detail in Figure 2-38 could be improved if a 90 degree rotation of the stanchion were permissible.

Figure 2-35

Figure 2-36

Figure 2-37

Figure 2-38

Section 3

ANALYSIS OF STRUCTURAL INTERSECTIONS

3.1 GENERAL

Specific attention was focused on structural intersections because for any given ship they generally constitute the largest single group of details and because many structural failures have been attributed to stress concentrations due to unfavorable geometry.

The analysis was undertaken in two parts: one to investigate the effect of cutout geometry and type of loading on stress distribution, and the second to establish the effect of type of stiffener connection on stress level and joint efficiency.

Only a static analysis has been attempted, consequently the results are primarily useful for comparing alternative configurations, rather than for definitive evaluation of the load carrying capacity of the detail.

The finite element analysis was performed with the ICES "STRU DL II" computer program, specifically the IUG version V3MO of June 1976.

3.2 CLEARANCE CUTS

3.2.1 Clearance Cuts - Elliptic

The analytical model employed represents a beam with a 29 inch x 1/2 inch web and 12-1/2 inch x 1 inch flange with a length of 10 feet. The girder is attached to a plate with an equivalent section area of 30 square inches. Three cutouts are modeled: coarse ones at the ends and one with a fine mesh at the center, which is the one analyzed. The details of the model are shown in Figures 3-1, 3-2, 3-3 and 3-4. Elliptical and flat-topped clearance cuts were studied under three

FIXED END

3-2

FINITE ELEMENT ANALYSIS MODEL
ELLIPTICAL CUTOUT

Figure 3-1

FOR MODEL
OF THIS AREA
SEE FIG. 3-4

FINITE ELEMENT ANALYSIS MODEL ELLIPTICAL CUTOUT

Figure 3-2

ELLIPTICAL CUTOUT

Figure 3-3

DETAILED MODEL OF CUTOUT AREA

ELLIPTICAL CUTOUT

Figure 3-4

loading conditions:

- Case 1 - Axial
- Case 2 - Bending
- Case 3 - Shear

The analysis of the flat-topped cutout was simplified by applying the displacements of the elliptic fine mesh segment boundary to the modified fine mesh segment.

3.2.2 Elliptic-Results

For each loading case the stress amplification in way of the clearance cut centerline was calculated and plotted. For example Figure 3-5 indicates a maximum amplification factor of 2.53 at the cutout. This means that the maximum axial stress is 2.53 times the average stress calculated for the

LOAD CASE NO. 1: AXIAL LOAD

Figure 3-5

intact section. Overall and local plots of stress magnification were prepared (Figures 3-6 and 3-7) , where the values shown are the ratios of von Mises equivalent stress of the cutout to those of the intact section. Von Mises equivalent stresses were used because they provide a generally accepted yielding criterion that takes all plane stress components into account.

Load distribution and section stress magnification factors. for the case of pure bending are shown in Figure 3-8. Note that essentially no section amplification occurs due to bending.

The stress amplification plots in Figures 3-9 and 3-10 represent ratios with respect to the maximum intact section stress, i.e. at the flange of the girder.

Pure section shear (Figure 3-11) is developed by applying a shear and a moment at the free end that induce canceling moments at the central cutout (Section A-A). Stress amplification due to shear is 1.95, or about a similar order of magnitude as that due to axial load.

Stress magnification factor plots in Figures 3-12 and 3-13 indicate that the highest concentration occurs at the upper corners of the cutout and is in the order of 4.4. A plot of stress magnification factor along the periphery of the cutout loaded in shear is presented in Figure 3-14.

Since axial loads on girders are generally small, it can be concluded that section shear is the most important factor to be considered in the design of cutouts. Future fatigue studies could be guided by this conclusion.

|LOAD CASE NO. 11| = AXIAL LOAD STRESS MAGNIFICATION FACTOR

| Figure 3-6

LOAD CASE NO.1: AXIAL LOAD STRESS MAGNIFICATION FACTOR

DETAIL @ CUTOUT

Figure 3-7

LOAD CASE NO. 2: BENDING

(SHEAR = 0)

Figure 3-8

LOAD CASE NO. 2: BENDING STRESS MAGNIFICATION FACTOR

Figure 3-9

LOAD CASE NO. 2 : BENDING STRESS MAGNIFICATION FACTOR

DETAIL 9 CUTOUT

Figure 3-10

LOAD CASE NO. 3: SHEAR
(MOMENT = 0 @ CUT-OUT)

Figure 3-11

LOAD CASE NO. 3: SHEAR STRESS MAGNIFICATION FACTOR

Figure 3-12

LOAD CASE NO. 3: SHEAR STRESS MAGNIFICATION FACTOR
DETAIL 9 CUTOUT

Figure 3-13

SHEAR STRESS CONCENTRATION FACTOR
ELLIPTICAL CUTOUT

Figure 3-14

3.2.3 Clearance Cuts - Straight

The analytical model configuration is described in Figures 3-15 to 3-18. Load cases considered were identical to those discussed in Section 3.2.

3.2.4 Straight - Results

The stress magnification factor due to axial load (Figure 3-19) is less than for the elliptic cutout at the center-line, but results in a slightly larger maximum factor occurring at an upper corner.

A plot of magnification factors based on von Mises stresses is presented in Figure 3-20, pointing to similar magnitudes but steeper stress gradients than were calculated for the elliptic cut. The bending case is shown in Figure 3-21 and 3-22, leading to the same conclusion stated in Section 3.3: that bending has negligible effect on cutout stresses.

Stress magnification factors for shear, Figures 3-23, 3-24 and 3-25, are the same as for the elliptical cutout, with slightly smaller gradients.

The choice of the elliptic cutout as a tentative standard detail in Section 4 is admittedly somewhat arbitrary, since no measurable difference is suggested by the analysis. As stated in Section 2, it is expected that manual cutting and trimming (and the workmanship inherent thereto) will be avoided by specifying a cutout shape that is preferably produced by automated processes.

FIXED END

FINITE ELEMENT ANALYSIS MODEL
STRAIGHT CUTOUT

Figure 3-15

FOR MODEL
OF THIS AREA
SEE FIG. 3-18

FINITE ELEMENT ANALYSIS MODEL STRAIGHT CUTOUT

Figure 3-16

STRAIGHT CUTOUT

Figure 3-17

DETAILED MODEL OF CUTOUT AREA

STRAIGHT CUTOUT

Figure 3-18

LOAD CASE NO. I: AXIAL LOAD

Figure 3-19

LOAD CASE NO. 1: AXIAL LOAD STRESS MAGNIFICATION FACTOR
DETAIL @ CUTOUT

Figure 3-20

LOAD CASE NO. 2: BENDING
(SHEAR = 0)

STRESS @ "A-A"
(WITH CUT-OUT)

CALCULATED STRESS DISTRIBUTION

Figure 3-21

LOAD CASE NO. 2 : BENDING STRESS MAGNIFICATION FACTOR
DETAIL ⑨ CUTOUT

LOAD CASE NO.3: SHEAR (MOMENT = 0 @ CUT-OUT)

Figure 3-23

LOAD CASE NO.3: SHEAR STRESS MAGNIFICATION FACTOR
DETAIL @ CUTOUT

Figure 3-24

Figure 3-25

3.3 COLLARED INTERSECTIONS

The model employed for finite element analysis consists of a beam-plate combination as described in Section 3.2. The beam is 90 inches long to its center of symmetry, and fixed at the end. The load on intersecting longitudinals was applied at either end of the longitudinals. Four conditions of support are considered for the stiffeners: a tab to the heel with and without panel stiffener support, and tabs to heel and bosom with and without panel stiffener support.

3.3.1 Single Tab

The analytical model is shown in Figures 3-26 and 3-27. Only the left cutout and the intersecting stiffener are modeled in detail (Figures 3-28 and 3-29).

For this support case only, analysis was performed for a load at one stiffener (Figure 3-30). The equivalent von Mises stresses were ratioed to similar stresses calculated for a model of an intact (i.e. reeved) member, and the resulting amplification factors are plotted in Figure 3-31 along the cutout boundary. Note stress amplification factors in the range of 7-8 versus the ideal or reeved case.

The more general and realistic case of all (five) stiffeners equally loaded is shown in Figure 3-32, and resulting stress amplification factors are plotted in Figure 3-33.

As an interesting aside, an error was made in establishing the coordinates of two adjacent elements along the periphery of the radius-sniped corner at the web of the longitudinal. That error is shown in Figure 3-34. The resultant amplification of stress at those elements was 1.8 times higher than for the smooth contour, which was established when this error was later corrected. The sensitivity to small changes in cut outline can be considered a strong indication of the need for clean automatic burning, though it is possible that the effect was exaggerated by the inherently peculiar behaviour of triangular elements. Further investigation of this sensitivity to cut outline would seem to be called for.

FINITE ELEMENT ANALYSIS MODEL
ELLIPTICAL CUTOUT_w/SINGLE TAB

Figure 3-26

FOR MODEL OF THIS
AREA SEE FIG. 3-29

FINITE ELEMENT ANALYSIS MODEL
ELLIPTICAL CUTOUT w/ SINGLE TAB

Figure 3-27

ELLIPTICAL CUTOUT w/ SINGLE TAB

Figure 3-28

ANGLE

· DETAILED MODEL OF CUTOUT AREA
ELLIPTICAL CUTOUT w/ SINGLE TAB

Figure 3-29

LOAD CASE 1: LOAD ON OUTSIDE LONGITUDINALS

Figure 3-30

STRESS CONCENTRATION FACTOR
FOR LOAD CASE I

Figure 3-31

LOAD CASE 2 : ALL LONGITUDINALS LOADED

Figure 3-32

STRESS CONCENTRATION FACTOR
FOR LOAD CASE 2

Figure 3-33

Figure 3-34

3.3.2 Double Tab

Figures 3-35 to 3-39 describe the model used to analyze the effect of adding a tab to the bosom of the stiffener. The stress amplification plot in Figure 3-40 indicates that the bosom collar reduces stress amplification at base snipe and clearance cut by only about 16 percent. This can be explained in terms of the much greater flexibility of the longer tab. For symmetrical stiffeners such as tees or flat bars the effect of double tabs should be significantly greater.

3.3.3 Single Tab With Panel Stiffener

Figures 3-41 to 3-45 describe the analytical model used to define the load carrying contribution of a panel stiffener attached to the heel of the penetrating member. The plot of stress magnification ratios along the cutout shown in Figure 3-46 suggests only a slightly greater stress reduction (to 18 percent) due to connection to the panel stiffener than was obtained by addition of the bosom tab.

Figure 3-35

FOR MODEL OF THIS
AREA SEE FIG.3-38

FINITE ELEMENT ANALYSIS MODEL
ELLIPTICAL CUTOUT w/DOUBLE TABS

Figure 3-36

ELLIPTICAL CUTOUT w/ DOUBLE TABS

Figure 3-37

DETAILED MODEL OF CUTOUT AREA
ELLIPTICAL CUTOUT w/DOUBLEc- TABS

Figure 3-38

LOAD FOR ELLIPTICAL CUTOUT w/DOUBLE TABS

Figure 3-39

STRESS CONCENTRATION FACTOR
ELLIPTICAL CUTOUT w/DOUBLE TABS

Figure 3-40

FINITE ELEMENT ANALYSIS MODEL
ELLIPTICAL CUTOUT w/ SINGLE TAB AND PANEL STIFFENER

Figure 3-41

FINITE ELEMENT ANALYSIS MODEL
ELLIPTICAL CUTOUT w/SINGLE TAB AND PANEL STIFFENER

Figure 3-42

ELLIPTICAL CUTOUT w/ SINGLE TAB AND PANEL STIFFENER

DETAILED MODEL OF CUTOUT AREA
ELLEPTICAL CUTOUT w/ SINGLE TAB AND PANEL STIFFENER

Figure 3-44

LOAD FOR ELLIPTICAL CUTOUT w/ SINGLE TAB
AND PANEL STIFFENER

Figure 3-45

STRESS CONCENTRATION FACTOR
ELLIPTICAL CUTOUT W/ SINGLE TAB AND PANEL STIFFENER

Figure 3-46

3.3.4 Double Tab With Panel Stiffener

The model for this extreme support case is described in Figures 3-47 to 3-51, and stress magnification factors are plotted in Figure 3-52. The conclusion is that the addition of the bosom tab has no measurable benefit over and above attachment of the panel stiffener.

3.3.5 Analysis Summary

The analysis in this section provided the somewhat surprising results that addition of a second tab and/or panel stiffener has little effect on the stresses of an angle stiffener supported by a tab to the heel. Equally surprising is that the additional support provided by a panel stiffener is not measurably greater than that provided by a tab to the angle bosom.

Figure 3-47

FINITE ELEMENT ANALYSIS MODEL
ELLIPTICAL CUTOUT W/DOUBLE TABS AND PANEL STIFFENER

Figure 3-48 ||

Figure 3-49

**DETAILED MODEL OF CUTOUT AREA
ELLIPTICAL CUTOUT WITH DOUBLE
TABS AND PANEL STIFFENER**

Figure 3-50

LOAD FOR ELLIPTICAL CUTOUT w/ DOUBLE TABS AND PANEL STIFFENER

Figure 3-51

STRESS CONCENTRATION FACTOR
ELLIPTICAL CUTOUT W/DOUBLE TABS AND PANEL STIFFENERS

Figure 3-52

Section 4
TENTATIVE COLLECTION OF STANDARD SHIP
STRUCTURAL DETAILS

To assist in the future development of standards, this section presents a collection of ship structural details that represent either a direct adoption of what seems the best now in use in the U. S. shipbuilding industry, or the development of slightly modified configurations based on comments in Section 2.

Details are presented in the following categories:

	DETAIL TYPE	FIGURE NO.	PAGE NO.
1.	Clearance Cutouts	4-1 to 4-4	4-3 & 4-4
2.	Clearance Cutouts With Nontight Lapped Collars	4-5 to 4-11	4-5 to 4-8
3.	Clearance Cutouts With Web Attachments	4-12 to 4-14	4-9 & 4-10
4.	Clearance Cutouts With Flange Attachments	4-15	4-11
5.	Clearance Cutouts With Web & Flange Attachments	4-16 & 4-17	4-12
6.	Clearance Cutouts With Web Attachments & Nontight Lapped Collars	4-18 to 4-20	4-13 & 4-14
7.	Clearance Cutouts with Flange Attachments & Nontight Lapped Collars	4-21	4-15
8.	Clearance Cutouts With Web & Flange Attachments & Nontight Lapped Collars	4-22 to 4-23	4-16
9.	Clearance Cutouts With Tight Lapped Collars	4-24 to 4-27	4-17 & 4-18
10.	Clearance Cutouts With Tight Flush Collars	4-28 to 4-31	4-19 & 4-20
11.	Miscellaneous Cutouts	4-32 to 4-36	4-21 to 4-23
12.	Nontight Penetrations	4-37	4-24
13.	Chocks and Brackets	4-38 to 4-45	4-25 to 4-28
14.	Stiffener Endings	4-46 to 4-52	4-29 to 4-31
15.	Panel Stiffeners	4-53 to 4-58	4-32 to 4-35
16.	Stanchion Endings	4-59 & 4-60	4-36

The key to selection is adequate strength and good producibility and maintainability. As an example, the rounded corners of the collars in Figures 4-5 and 4-6 permit continuous welding with adequate all around access. Similarly the tight lapped collars in Figures 4-24 and 4-25 are intended for easy welding access to the base of the collar. This philosophy has been carried throughout the remainder of the applicable details in this section and is most apparent in the details presenting stiffener endings and panel stiffeners.

The miscellaneous cutouts in Figures 4-34 and 4-35 have found application in Europe. They are only suggested to U.S. shipbuilders as an excellent method of reducing the stress at the base of any cutout terminating normal to ship plating. These details have the obvious drawback of being susceptible to damage prior to installation.

Only two beam brackets have been illustrated, Figures 4-44 and 4-45, but they demonstrate the essential characteristics that the many possible variations should possess.

Similarly the great variety of possible stanchion end connections causes the listing here to include only two that are generally illustrative of good practice, see Figures 4-59 and 4-60.

It is the intent of this section to serve as a basis of individual or committee discussion and review, thereby possibly leading to consensus standards. Reference 2 and Section 2 of this report will be of assistance in that review.

FABRICATED ANGLE

ROLLED ANGLE

Figure 4-1

4-3

BULB FLAT

Figure 4-2

CLEARANCE CUTOUTS

TEE

Figure 4-3

SLAB/FLAT BAR

Figure 4-4

CLEARANCE CUTOUTS

Use where web must be dropped
over stiffener with adequate clearance.

FABRICATED ANGLE

ROLLED ANGLE
COLLAR AT HEEL OF ANGLE
Figure 4-5

Use tab to bosom only if tab
to heel is very difficult.

FABRICATED ANGLE

ROLLED ANGLE
COLLAR AT BOSOM OF ANGLE
Figure 4-6

CLEARANCE CUTOUTS WITH
NONTIGHT LAPPED COLLARS

BULB FLAT
COLLAR AT HEEL

Figure 4-7

BULB FLAT
COLLAR AT BOSOM

Figure 4-8

CLEARANCE CUTOUTS WITH
NONTIGHT LAPPED COLLARS

4-7

TEE |
WEB ATTACHMENT ONLY |

Figure 4-9 |

TEE |
WEB AND FLANGE ATTACHMENT

Figure 4-10 |

CLEARANCE CUTOUTS WITH
NONTIGHT LAPPED COLLARS

SLAB/FLAT BAR

Figure 4-11

CLEARANCE CUTOUTS WITH
NONTIGHT LAPPED COLLARS

ROLLED ANGLE

Figure 4-12

BULB FLAT

Figure 4-13

CLEARANCE CUTOUTS WITH
WEB ATTACHMENT

SLAB
Figure 4-14

CLEARANCE CUTOUTS WITH
WEB ATTACHMENT

Figure 4-15

CLEARANCE CUTOUTS WITH FLANGE ATTACHMENT

FABRICATED ANGLE

Figure 4-16

ROLLED ANGLE

Figure 4-17

CLEARANCE CUTOUTS WITH
WEB & FLANGE ATTACHMENT

4-13

ANGLE

Figure 4-18

BULB FLAT

Figure 4-19

CLEARANCE CUTOUTS
WITH WEB ATTACHMENT &
NONTIGHT LAPPED COLLARS

Figure 4-20

CLEARANCE CUTOUTS
WITH WEB ATTACHMENT &
NONTIGHT LAPPED COLLARS

4-15

TEE

Figure 4-21

CLEARANCE CUTOUTS
WITH WEB & FLANGE ATTACHMENT
& NONTIGHT LAPPED COLLARS

ROLLED ANGLE

Figure 4-24

FABRICATED ANGLE

Figure 4-25

CLEARANCE CUTOUTS
WITH TIGHT LAPPED COLLARS

CLEARANCE CUTOUTS WITH TIGHT FLUSH COLLARS

BULB FLAT

Figure 4-30

TEE

Figure 4-31

CLEARANCE CUTOUTS
WITH TIGHT FLUSH COLLARS

Figure 4-32

Figure 4-33

MISCELLANEOUS CUTOUTS
CLEARANCE CUTS AT PLATE BUTT

Figure 4-34

Figure 4-35

MISCELLANEOUS CUTOUTS
ALTERNATE DETAILS

TYPICAL DRAIN AND LIMBER HOLE
IN DEEP STRUCTURE

Figure 4-36

Figure 4-37

MISCELLANEOUS CUTOUTS
NONTIGHT PENETRATIONS

Figure 4-38

Figure 4-39

Figure 4-40

Figure 4-41

Use primarily to avoid knife
edge crossings at foundations

Figure 4-42

Bolting chock adjacent to
bolt hole and not in line
with backup structure

Figure 4-43

4-28

Figure 4-44

Figure 4-45

CHOCKS & BRACKETS

Stiffener endings for non-tight opening which permits access from both sides

Figure 4-46

Stiffener endings for tight intersection which permits welding from one side only

Figure 4-47

Figure 4-48

Figure 4-49

Longer tab improves end
restraint and can be used
with minimal backup

STIFFENER ENDINGS

For applications with
low lateral load

Figure 4-50

Figure 4-51

Figure 4-52

STIFFENER ENDINGS

Figure 4-53

Arrangement for L or Tee
panel stiffener in presence
of large lateral load

Figure 4-54

PANEL STIFFENERS

Limit application to panels
with low lateral load

Figure 4-55

Figure 4-56

Good arrangement for L stiffener with lateral load. Common plane for stiffeners, reduces error. In horizontal application avoids turning stiffener leg up.

Figure 4-57

Figure 4-58

4-36

Figure 4-59

Figure 4-60

STANCHION ENDINGS

Section 5
REFERENCES

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