

SSC-405

**FATIGUE RESISTANT DETAIL
DESIGN GUIDE FOR SHIP
STRUCTURES**



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1999

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FATIGUE RESISTANT DETAIL DESIGN GUIDE FOR SHIP STRUCTURES

This Guide is a result of a 1995 Ship Structure Committee Symposium in Washington, DC, at which it was recognized that practicing naval architects required help in applying fatigue design research and development results. This Guide consolidates the state-of-the-art in ship structural detail fatigue design.

The information in the Guide is presented in four sections, starting with an overview of the fatigue design problem and related issues. The second includes a catalogue of typical structural details for commercial and combatant ship types, with suggested structural improvements for fatigue life extension. This, along with a brief discussion of fabrication issues, represents the "Level 1" approach to ship structure fatigue design. The third section of the report presents analytical procedures for fatigue based design of structural details. Load, stress and fatigue analysis procedures are presented at two levels of complexity, "Level 2" and "Level 3". The simplified Level 2 approach makes use of Classification Society rules and analytical methods, whereas, the spectral Level 3 approach explicitly considers vessel operational profiles and wave climates in statistical load estimation and makes use of FE models for stress analysis. Both the Level 2 and 3 approaches are based on a "hot spot" stress approach using Miner's rule in either a discrete or continuous form. The final section of the Guide provides worked fatigue design examples to demonstrate the concepts presented in the previous sections.

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NOMENCLATURE

ABS	American Bureau of Shipping
AP	Aft perpendicular
A_i	Stress coefficient
a	Crack depth: or Length of double bottom panel; or Scale parameter of the basic S-N curve, or Acceleration (m/s^2)
a_l	Combined longitudinal acceleration (m/s^2) [<i>Appendix B</i>]
a_o	Acceleration constant [<i>Appendix B</i>]
a_{pt}	Tangential pitch acceleration (m/s^2) [<i>Appendix B</i>]
a_{px}	Longitudinal component of pitch acceleration (m/s^2) [<i>Appendix B</i>]
a_{pz}	Vertical component of pitch acceleration (m/s^2) [<i>Appendix B</i>]
a_{ry}	Horizontal component of roll acceleration (m/s^2) [<i>Appendix B</i>]
a_{rz}	Vertical component of roll acceleration (m/s^2) [<i>Appendix B</i>]
a_t	Combined transverse acceleration (m/s^2) [<i>Appendix B</i>]
a_v	Combined vertical acceleration (m/s^2) [<i>Appendix B</i>]
a_x	Surge acceleration (m/s^2) [<i>Appendix B</i>]
a_y	Acceleration due to sway and yaw (m/s^2) [<i>Appendix B</i>]
a_z	Heave acceleration (m/s^2) [<i>Appendix A</i>]
a_1 a_2	Fatigue design curve parameter
B	Moulded breadth of ship (m or ft); plate thickness [<i>Appendix B</i>]
BV	Bureau Veritas
b	Transverse width of double bottom panel
CG	Center of gravity
CMS	Committee on Marine Structures
C_B	Block coefficient [<i>Appendix B</i>]
C_w	Wave coefficient [<i>Appendix B</i>]
C_{wp}	Waterplane area coefficient
D	Cumulative fatigue damage
d_i	Fatigue damage inflicted by each stress cycle
D_m	Moulded Depth of Ship (m)
DNV	Det Norske Veritas
E	Young's Modulus [GPA]
FE	Finite Element
FP	Forward perpendicular
FTL	Fleet Technology Limited

$F(H_{s1})$ Cumulative probability of lower limit wave height
 $F(H_{s2})$ Cumulative probability of upper limit wave height
 $F(\text{Sea State})$ Probability of occurrence of a given sea state

$F_{load}(H_s, V, \theta, L.C.)$	Characteristic (RMS) load response of the vessel for each operational condition (defined by wave height, speed, heading, load condition)
$F_{\Delta\sigma}(\Delta\sigma_{hot\ spot})$	Rayleigh short-term stress range cumulative distribution function for the i^{th} operational condition
f	Freeboard at the transverse section considered (m or ft.) [Appendix B]
f_f	Correction factor to convert from one probability level to another probability level
f_{ef}	Encounter frequency correction factor
$f_{mc}(H_s)$	Composite distribution of significant wave heights
$f_{mc}(H_s; T_z)_{composite}$	Composite distribution of wave heights and zero crossing periods (composite scatter diagram)
$f_{mi}(H_s; T_z)$	Probability distribution of wave heights and zero crossing periods in Marsden Zone
$f_{mcomb}(H_s; T_z)$	Probability of wave height/zero crossing period in the Combined Marsden Zone
f_{total}	Total probability (three-dimensional probability) or percent of time for each operational condition
$f_V(V H_s)$	Conditional probability of speed, V , given a wave height, H_s (or sea state) – all periods included
$f_V(V (H_s; T_z))$	Conditional probability of speed, V , given a wave height combination or sea state, $(H_s; T_z)$
$f_{\theta}(\theta H_s)$	Conditional probability of heading, θ , for given wave height, H_s , - all periods included
$f_{\theta}(\theta (H_s; T_z))$	Conditional probability of heading, θ , given a sea state, $(H_s; T_z)$
$\overline{f_z}$	zero crossing rate (H_z)
$\overline{f_z}$	average zero crossing rate (H_z)
g	Gravitational constant
G_x	General form for the Limit State equation in a First Order reliability analysis
GL	Germanischer Lloyd
GM	Metacentric height [Appendix B]
HSE	UK Health and Safety Executive
H_s	Significant wave height (m or ft.)
h	Shape parameter for Weibull Distribution (general)
h_n	Weibull shape factor for n^{th} loading condition
h_0	Basic long-term Weibull shape parameter, modified for location of Point of Interest
h_s	Vertical distance from point considered to surface inside a tank (m) [Appendix B]
IACS	International Association of Classification Societies
IIW	International Institute for Welding
ISSC	International Ship and Offshore Structures Congress
I	Moment of Inertia (general)
I_a, I_b	Moment of inertia about the transverse neutral axis, including the effective width of plating, of long (a) and short (b) stiffeners

I_h	Hull cross section moment of inertia about the vertical neutral axis
I_{pi}	Moment of inertia of effective width of plate – $i = a$ or b for long or short direction (grillage)
I_v	Hull cross section moment of inertia about the transverse neutral axis
i_a, i_b	Blended stiffness per unit (girder and plate) about transverse (longitudinal) neutral axis of double bottom
K	Stress concentration factor (general)
K_b	Stress Concentration Factor Dependent on aspect ratio, ρ , and panel's boundary conditions
K_G	Global stress concentration factor to account for gross structural geometry (e.g., hatch openings, shear lag) affecting the local nominal stress field
K_g	Stress concentration factor due to the gross geometry of the detail
K_n	Stress concentration factor due to non-symmetric stiffeners
K_{te}	Stress concentration due to eccentricity
$K_{t\alpha}$	Stress concentration due to angular mismatch
K_w	Notch stress concentration; local weld configuration stress concentration
K_0	Stress transfer function relating vessel load response to detail hot spot stress
k	Wave number (from dispersion relation (ω^2/g), or number of stress blocks
k'	roll correction factor for bilge keels [Appendix B]
k_f	Side pressure panel factor [Appendix B]
k_r	Roll radius of gyration (m) [Appendix B]
k_{wm}	Moment distribution factor
k_{ws}	Shear distribution factor
k_s	Side pressure form factor [Appendix B]
L	Rule length of ship (m or ft) [Appendix B]
L_{BP}	Length between perpendiculars (m or ft.)
L_{OA}	Length overall (m or ft.)
L.C.	Load condition (e.g., ballast, full load, etc.) [Appendix B]
LNG	Liquified Natural Gas
LPG	Liquified Petroleum Gas
LR	Lloyd's Register
ℓ_e	Effective span of stiffener or longitudinal (m or ft.)
l_s	Distance (span) between bulkhead and transverse frame (m or ft.)
M	Bending moment (MNm or LTft.)
M_d	Vertical bending moment range (MNm or LTft.)
M_{dh}	Design wave-induced hogging moment amplitude (MNm or LTft.) including the effects of whipping [Appendix B]
M_{ds}	Design wave-induced sagging moment amplitude (MNm or LTft.) including the effects of whipping

M_h	Horizontal bending moment amplitude at the location of interest (MNm or LTft.)
M_{hog}	Hogging moment (MNm or LTft.)
M_{sw}	Stillwater bending moment (MNm or LTft.) [<i>Appendix B</i>]
M_{sag}	Sagging Moment (MNm or LTft.)
M_v	Vertical (sagging or hogging) bending moment amplitude at the location under consideration (MNm or LTft.)
M_e	Maximum allowable misalignment measured from the centerlines of intersecting plates (mm or in.)
MEDS	Marine Environmental Data Services
m	Location parameter of the Weibull distribution
m_1, m_2	Fatigue design curve slope parameters
m_0	Spectral zeroth moment employed in spectral analysis
m_δ	Moment factor due to relative deflection between transverse supports
N	Total number of cycles (wave encounters or stress reversals) experienced by the structure within a time, t ; used to express Fatigue Life.
N_i	Average number of loading cycles to failure under constant amplitude loading at the i^{th} stress range
N_t	Total number of loading cycles to failure
N_{load}	Total number of load conditions
N'	Total number of Marsden Zones along the route
N_s	Number of cross ties in cargo or ballast tank
NOAA	National Oceanic and Atmospheric Administration
n_i	Number of stress cycles in stress block “ i ”
n_o	Number of wave encounters corresponding to the service level of probability for load/stress
P_f	Probability of fatigue-reduced failure
POI	Point of Interest
p	Effective lateral pressure
p_d	Dynamic pressure amplitude (kPa)
p_{dp}	Combined pressure dominated by pitch motion in head/quartering seas [<i>Appendix B</i>]
p_{dr}	Combined pressure dominated by roll motion in beam/quartering seas [<i>Appendix B</i>]
p_e	External pressure amplitude (half pressure range) related to the draft of the load condition considered (kPa) [<i>Appendix B</i>]
p_{ext}	External pressure (kPa)
p_i	Fraction of time at the i th operational condition
p_{int}	Internal pressure (kPa) [<i>Appendix B</i>]
p_1	Pressure due to vertical acceleration [<i>Appendix B</i>]
p_2	Pressure due to transverse acceleration [<i>Appendix B</i>]
p_3	Pressure due to longitudinal acceleration

p_{ip}	Dynamic pressure term (kPa) for pitch-induced pressure
p_{st}	Static pressure
p_n	Fraction of design life in the n^{th} load condition
$\text{prob}(H_s)$	Marginal probability of wave heights
$\text{prob}(H_s; T_z)$	Marginal probability of the sea state
$\text{prob}(V \text{ and } (H_s; T_z))$	Joint probability of speed and sea state
$\text{prob}(\theta \text{ and } (H_s; T_z))$	Joint probability of heading and sea state
$\text{prob}(\text{emergence})$	Probability of emergence of POI for each stationary condition
$Q(\Delta\sigma_{\text{hotspot}})$	Weibull long-term cumulative probability distribution functions for two parameter Weibull distribution (h and q parameters)
$Q'(\Delta\sigma_{\text{hotspot}})_{\text{long}}$	Weibull long term probability function pertinent to hot spot stress
$Q'(\Delta\sigma_{\text{hotspot}})_{\text{short}}$	Probability distribution function pertinent to hot spot stress and a single stationary condition (short term)
q	Scale parameter for Weibull distribution
RAO	Response amplitude operator
RINA	Royal Institute of Naval Architects
RMS	Root mean square
R_{pp}	Distance from (pitch) axis of rotation to local centre of mass or volume (m) <i>[Appendix B]</i>
R_{pX}	Longitudinal distance from the pitch axis of rotation to the local centre of mass or volume (m) <i>[Appendix B]</i>
R_{pZ}	Vertical distance from the pitch axis of rotation to the local centre of mass or volume (m) <i>[Appendix B]</i>
R_R	Distance from the axis of roll rotation to the local centre of mass or volume (m) <i>[Appendix B]</i>
R_{RY}	Transverse distance from the roll axis of rotation to the local centre of mass or volume (m or ft.) <i>[Appendix B]</i>
R_{RZ}	Vertical distance from the roll axis of rotation to the local centre of mass or volume (m or ft.) <i>[Appendix B]</i>
$R(\omega_e)$	Response function
r_a	Distance from point considered to the transverse neutral axis of panel (m or ft.)
r_b	Distance from point considered to the longitudinal neutral axis of panel (m or ft.)
r_i	Ratio of the i^{th} zero crossing rate, to the average zero crossing rate for all operational conditions
r_{ij}	Relative number of stress cycles in short-term condition i, j against the total number of cycles in the vessel life
r_δ, r_π	Moment factors for interpolation to crack location along stiffener length
r_p	Reduction of pressure amplitude in the wave zone <i>[Appendix B]</i>
SCF	Stress Concentration Factor
SSC	Ship Structure Committee

SWL	Summer load waterline
S_{θ}	Wave spectral density (m^2*s)
$S_{\eta}(\omega_e)$	Modified wave height spectrum (m^2*s)
s	Stiffener spacing (mm or in.)
s_a	Transverse spacing between girders or longitudinals running in the longitudinal direction (m or ft.)
s_b	Longitudinal spacing between girders or web frames in the transverse direction (m or ft.)

T_{act}	Stillwater draft at the considered load condition (m) [<i>Appendix B</i>]
T	Draft of ship at load condition (m); also Period (s)
\bar{T}	Long-term average stress period (s)
T_P	Peak wave period (s)
T_{PP}	Period of pitch (s) [<i>Appendix B</i>]
T_R	Period of roll (s) [<i>Appendix B</i>]
T_S	Significant wave period (s)
T_z	Zero crossing period (s)
t	Time (s); also Thickness
t_b	Bracket thickness (mm or in.)
t_{corr}	Corrosion thickness allowed
t_d	Design life of ship expressed in seconds (s)
t_f	Stiffener flange thickness (mm or in.)
t_n	Net plate thickness (mm or in.)
t_r	Reference thickness (mm or in.)
t_s	Continuous plate thickness (mm or in.)
t_w	Web thickness (mm or in.)
TAPS	Trans-Alaska Pipeline Service
V	Vessel speed (knots) [<i>Appendix B</i>]
VLCC	Very Large Crude Carrier
V_H	Horizontal shear (kN) [<i>Appendix B</i>]
V_{dh}	Vertical shear load at midship due to hogging (kN) [<i>Appendix B</i>]
V_{ds}	Vertical shear load at midship due to sagging (kN) [<i>Appendix B</i>]
x	Length measurement variable (general), or Longitudinal distance from the AP to the section considered (m or ft.)
x_c	Distance from end of stiffener to crack location (mm or in.)
x_s	Longitudinal distance from centre of free surface of liquid in tank to pressure point considered (m or ft.) [<i>Appendix B</i>]
y	length measurement in variable in transverse direction (general), or Transverse distance from the centre line to the point of interest (m or ft.) [<i>Appendix B</i>]
y_s	Transverse distance from centre of free surface of liquid in tank to point of interest (m or ft.) [<i>Appendix B</i>]
\bar{y}	Transverse distance from centreline for roll calculation [<i>Appendix B</i>]

Z	Section modulus
Z_{deck}	of deck
Z_s, Z_t	of stiffener or longitudinal (mm^3 or in.^3)
z	Vertical distance on the hull:
z_{na}	from the neutral axis to the point of interest (m or ft.)
z_{bl}	from the baseline to the point of interest (m or ft.)
z_{wl}	from the still waterline (m or ft.) (ship upright) [Appendix B]
z_1, z_2	Instantaneous immersions of POI due to ship motion (m or ft.)
α	Maximum roll angle, single amplitude (rad) [Appendix B]
β	Reliability index derived from limit state equation $G(x)$
$\Gamma()$	Gamma function
$\Gamma(;)$	Complementary incomplete gamma function
$\gamma(;)$	Incomplete gamma function
$\Delta\sigma_c$	Corrected design stress range
$\Delta\sigma_{\text{hotspot}}$	Reference hot spot stress range (MPa)
$\Delta\sigma_{\text{nom}}$	Reference nominal stress range (MPa)
$\Delta\sigma_{\text{notch}}$	Notch stress range (MPa) [Appendix A]
$\Delta\sigma_o$	Design stress range for the n^{th} loading condition (MPa)
$\Delta\sigma_{\text{slope}}$	Stress range at which change in slope occurs (MPa)
$\Delta\sigma_{\text{S-N}}$	Design stress range allowed by the relevant S-N curve (MPa)
δ	Deformation of nearest frame relative to transverse bulkhead (mm or in.)
ε	Stress correlation coefficient
Φ	Standard normal distribution function
ϕ	Pitch angle, single amplitude (rad)
η	Fatigue usage factor, or Torsion factor in grillage
θ	Ship's heading relative to wave direction (degrees, $0^\circ = \text{head seas}$)
θ'	spreading angle for wave spectrum (rad)
λ	Wave length (m or ft.)
μ_G	Mean of the limit state equation $G(x)$
μ_i	Proportion of time spent in the i^{th} area (Marsden zone)
ν	Poisson ratio

ρ	panel aspect ratio
ρ_{ij}	Correlation coefficient for variables i and j in limit state analysis
ρ_{sw}	Density of seawater (1.025 t/m ³)
σ	Stress (MPa)
σ_b	Local bending stress (MPa)
σ_{eq}	Equivalent stress (MPa)
σ_G	Standard deviation of limit state equation G(x)
σ_m	Membrane stress
σ_p	Peak stress (MPa)
σ_r	Residual stress (MPa)
σ_y	Yield strength (MPa)
σ_t	Total stress at crack location (MPa)
σ_{tp}	Peak total stress at crack location (MPa)
σ_{nom}	Nominal stress
σ_{lower}	Stress range bin lower value (MPa)
σ_{mid}	Stress range bin mid value (MPa)
σ_{upper}	Stress range bin upper value (MPa)
σ_1	Primary stresses due to bending, shear and torsion in the main hull girder; peak total stress (MPa)
σ_2	Secondary stresses due to local stiffener bending (MPa)
σ_{2L}	Plate/panel secondary stresses (MPa)
σ_3	Tertiary plate bending stress (MPa)
σ_δ	Stresses due to relative deflection between web frame and transverse bulkhead (MPa)
ω	Wave frequency (rad/s)
ω_e	Encounter frequency that accounts for the effects of speed and heading

PART A - OVERVIEW OF THE FATIGUE DESIGN GUIDE

A.1 INTRODUCTION

In March 1995, the US National Research Council's Committee on Marine Structures (CMS), on behalf of the Ship Structure Committee (SSC), convened a symposium in Washington, D.C., to address the unusually great number of bulk carrier losses and the continuing occurrence of fatigue cracking in relatively new and aging ships [Ref. A.1]. The symposium sought input from ship designers, fabricators, operators, and regulators as well as experts in fatigue, fracture, and structural reliability on how the Ship Structure Committee could best serve the marine community in preventing this type of failure in ships.

One outcome of this symposium was that practicing naval architects and engineers were having difficulty applying the results of much of the research and development work that had been carried out to investigate fatigue, fracture and structural reliability. As a consequence, the Ship Structure Committee initiated two related projects:

- development of a ship structural detail fatigue design Guide; and,
- development and presentation of a ship structure fatigue and fracture short course.

These two projects were awarded to Fleet Technology Limited under a competitive bidding process, under contract PO Number 97-0046.

This report is the result of the ship structure detail fatigue design Guide development project.

A.2 OBJECTIVE

The objective of the project was to develop a practical and rationally based "Fatigue Resistant Detail Design Guide" ("the Guide") that Engineers and Naval Architects with limited fatigue design experience could use to design cost-effective, fatigue-resistant, welded steel ship structural details.

The objective of the Guide is to provide a single source of information for practicing designers to use in addressing fatigue issues in the design of ship structure. In order to achieve this objective, the Guide was required to cater to all levels of design detail or phases in the design process. For this reason, the Guide presents three levels of structural design sophistication:

- structural detail geometry and layout guidance;
- a simple fatigue design procedure; and,
- an in-depth fatigue design procedure.

These can be used sequentially at successive stages in the ship design process. However, in many projects it may not be necessary to use the more detailed levels to achieve satisfactory fatigue performance.

A.3 BACKGROUND TO FATIGUE IN SHIP STRUCTURES

A.3.1 Metal Fatigue Process in Steel Ships

Metal fatigue is the progressive failure of metal under cyclic loading and as the name “fatigue” implies, it is a mode of degradation in which the steel is worked until it simply gets tired. This fatigue cracking process can be divided into three basic stages:

- (i) the initiation of microscopic cracks at local stress concentrations;
- (ii) the growth and coalescence of microscopic cracks into macroscopic cracks; and,
- (iii) the growth of macroscopic cracks to a critical size for failure (e.g., plastic collapse, fracture, excessive deflection, or loss of water-tightness).

The absolute and relative duration of these stages depend on the magnitude of the cyclic stresses or strains at the crack initiation site and along the crack propagation path, environmental effects, and the resistance of the material to the initiation and propagation of fatigue cracks. The magnitude of the local cyclic stress or strain is dependant on the magnitude of applied cyclic loads, severity of local stress concentration effects, structural redundancy, and the stress-strain response of a material under cyclic loading [e.g., Ref. A.2].

Fatigue cracks in steel ships generally initiate at local notches (e.g., weld toe, rat hole, or weld termination) in structural details that are located in highly stressed, primary or secondary structures [e.g., Ref. A.3, A.4]. Fatigue-prone areas in common types of ships are listed in [Tables A.3.1-A.3.5](#) [Ref. A.5]. The initiation and subsequent propagation of these cracks are primarily driven by wave induced cyclic loads including:

- (i) longitudinal bending, transverse bending, and torsion of the hull girder; and,
- (ii) fluctuating hydrostatic pressure on side shell plating, cargo hold boundaries, and tank walls, [e.g., Ref. A.6].

Other sources of cyclic loading include machine vibration, propeller-induced vibration, and deck loads . The exposure of unprotected surfaces to corrosive media (e.g., sea water or sour crude oil) can also accelerate the initiation and propagation of fatigue cracks, either directly through corrosion fatigue mechanisms or indirectly through the higher cyclic stresses that result from localized and general corrosion [Ref. A.7].

Although most fatigue cracks in ships are not detected by conventional inspection techniques until they are several inches long and through the thickness of plating, the majority of detected cracks do not pose an immediate threat for catastrophic rapid fracture because of the relatively good fracture toughness of modern ship steels, the inherent redundancy of ship structures, the use of crack arrestors, and the relatively low level of normal service loads. Nevertheless, any detected cracks are usually repaired at the earliest opportunity to preserve the water-tightness or oil tightness of the ship and to prevent a chain reaction of localized failures leading to overall structural failure. The latter scenario is believed to be responsible for some of the recent spate of bulk carrier losses [Ref. A.8].

For example, cracks at the corners of hatch corners and coamings in bulk carriers could allow water to leak into dry cargo holds. Sloshing of the resulting slurry could introduce significant dynamic loads on cargo hold plating, and internal mixing could produce explosive gases. As another example, cracking at the hold frame ends of bulk carriers could result in the detachment of side shell plating from internal framing. This could eventually lead to the separation of the end brackets from the slant of the topside tanks or bilge hopper tanks.

Table A.3.1.1: Highly Loaded Structural Elements - Tankers

Structural Member	Structural Detail	Load Type
Side, bottom and deck plating and longitudinals	Butt joints, deck openings and attachment to transverse webs, transverse bulkheads and intermediate longitudinal girders	Hull girder bending, stiffener lateral pressure load and support deformation
Transverse girder and stringer structures	Bracket toes, girder flange butt joints, curved girder flanges, panel knuckles including intersecting transverse girder webs, etc. Single lug slots for panel stiffeners, access and lightening holes	Sea pressure load combined with cargo or ballast pressure load
Longitudinal girders of deck and bottom structure	Bracket terminations of abutting transverse members (girders, stiffeners)	Hull girder bending and bending/deformation of longitudinal girder and abutting member

Table A.3.1.2: Highly Loaded Structural Elements - Bulk Carriers

Structural Member	Structural Detail	Load Type
Hatch corners	Hatch corner	Hull girder bending, hull girder torsional deformation
Hatch side coaming	Termination of end bracket	Hull girder bending
Main frames	End bracket terminations, weld main frame web to shell for un-symmetrical main frame profiles	External pressure load, ballast pressure load as applicable
Longitudinals of hopper tank and top wing tank	Connection to transverse webs and bulkheads	Hull girder bending, sea and ballast pressure load
Double bottom longitudinals	Connection to transverse webs and bulkheads	Hull girder bending stress, double bottom bending stress and sea, cargo and ballast pressure load
Transverse webs of double bottom, hopper and top wing tank	Slots for panel stiffener including stiffener connection members, knuckle of inner bottom and sloped hopper side including intersection with girder webs (floors). Single lug slots for panel stiffeners, access and lightening holes	Girder shear force, and bending moment, support force from panel stiffener due to sea, cargo and ballast pressure load

Table A.3.1.3: Highly Loaded Structural Elements - Ore Carriers

Structural Member	Structural Detail	Load Type
Upper deck plating	Hatch corners and side coaming terminations	Hull girder bending
Side-, bottom- and deck longitudinals	Butt joints and attachment to transverse webs, transverse bulkheads, hatch openings corners and intermediate longitudinal girders	Hull girder being, stiffener lateral pressure load and support deformation
Transverse girder and stringer structures	Bracket toes, girder flange butt joints, curved girder flanges, panel knuckles at intersection with transverse girder webs, etc. Single lug slots for panel stiffeners, access and lightening holes	Sea pressure load combined with cargo or ballast pressure
Transverse girders of wing tank	Single lug slots for panel stiffeners	Sea pressure load (in particular in ore loading condition)

Table A.3.1.4: Highly Loaded Structural Elements - Container Carriers

Hull Member	Structural Detail	Load Type
Side and bottom longitudinals	Butt joints and attachment to transverse webs, transverse bulkheads and intermediate longitudinal girders	Hull girder bending, torsion , stiffener lateral pressure load and support deformation
Upper deck	Plate and stiffener butt joints, hatch corner curvatures and support details welding on upper deck for container pedestals, etc.	Hull girder bending and torsional warping stress

Table A.3.1.5: Highly Loaded Structural Elements - Roll on/Roll off- and Car Carrier

Hull Member	Structural Detail	Load Type
Side and bottom longitudinals	Butt joints and attachment to transverse webs, transverse bulkheads and intermediate longitudinal girders	Hull girder bending, stiffener lateral pressure load and support deformation
Racking constraining girders, bulkheads, etc.	Stress concentration points at girder supports and at bulkhead openings	Transverse acceleration load

A.3.2 Historical Treatment of Metal Fatigue in Steel Ships

The design of a ship can be divided into three phases: conceptual design, preliminary design, and detail design. The principal dimensions, topology, and overall geometry of a ship are determined during the conceptual design phase, usually by non-structural considerations such as beam and draft limitations, cargo type, and cargo capacity.

The location, spacing, scantlings, and geometry of primary structure (e.g., side shell, decks, main bulkheads, main beams and girders) and secondary structure (e.g., stiffened panels, grillages, tank tops, short decks) are then selected during the preliminary design stage. These selections, and the resulting degree of structural continuity, optimization, and symmetry, determine the ultimate strength of the hull. In the next phase, detail design, the geometry and scantlings of details are selected within fabrication and maintenance constraints to minimize local stresses.

Until recently, fatigue cracking has not been explicitly considered in the detail and preliminary design phases except for damaged structure, novel structural configurations, and special types of ships such as LNG ships. The preliminary design of merchant ships has been largely based on static strength requirements in classification society rules [Ref. A.9- A.12], which are expressed in terms of empirical relationships for minimum scantlings and spacings, whereas the preliminary design of naval ships has been based on more rigorous static strength calculations with representative design loads [Ref. A.13, A.14]. Prior to the 1990's, designers and fabricators were able to rely on the margins in these static strength requirements and experience-based rules-of-thumb for detail design to achieve adequate fatigue performance in conventional ship structures. Fatigue cracks were rarely detected in ships less than 10 years old, and the frequency of cracking in older ships was generally acceptable to regulators and owners. Over the past two decades, however, significant changes in the age, design, fabrication, operation, and regulation of merchant and naval vessels have resulted in the need for rationally-based fatigue design approaches to address the following concerns:

- There has been a significant increase in the incidence of fatigue cracking in relatively new ships since the early 1980's. For example, fatigue cracks were detected in the summer of 1990 at the intersections of side shell longitudinals and transverse bulkheads in 15 second-generation VLCC's after only two to five years of service [Ref. A.15]. Similar cracking was reported in several classes of oil tankers operating on the Trans-Alaska Pipeline Service (TAPS) route in the mid-1980's [Ref. A.16 and A.17]. This change has been attributed to the introduction of more structurally optimized ships with thinner scantlings as a means to reduce weight, fabrication cost and operating cost. This optimization has been achieved through the greater use of high strength steels and the use of more sophisticated design tools [Ref. A.18, A.19]. Since the fatigue strength of as-welded steel joints is essentially independent of tensile strength, the stress concentrations of structural details must be adequately reduced to compensate for the higher design stresses, higher local bending stresses, and reduced margins for corrosion and wear in high strength steel structures with thinner scantlings. As a general precaution, classification societies introduced the so-called "k factor" on minimum scantling requirements which prevented design stresses for extreme loads from increasing in direct proportion to tensile strength. However, the continuing occurrence of fatigue cracking in ships demonstrated the need for more direct control of fatigue cracking. A number of owners with poor experiences have since insisted that the use of high strength steels in their ships be kept to a minimum, and shipyards have responded with standard designs using mainly low and medium strength steels.

- According to available statistics [e.g., Ref. A.20, A.21], the rate of reported fatigue cracking and other structural failures in current merchant and naval vessels increases significantly (i.e., the aging phase starts) after the first 10 years of service. However, approximately 60% of the world's tanker fleet and about 40% of the world's bulk carrier fleet (by tonnage) are over 15 years old, and many naval vessels are of the same vintage. As a result, maintenance costs and downtime are rising for operators. At the same time, many operators are facing reduced maintenance and operating budgets. This is forcing designers, fabricators, and operators of merchant and naval vessels to seek integrated approaches to the design, construction, and maintenance of ships with a view towards maximizing operational availability and minimizing life-cycle maintenance costs without compromising structural integrity [Ref. A.22, A.23].
- Strict environmental regulations have been introduced around the world since the grounding of the *Exxon Valdez*. These regulations will require nearly all new oil tankers to have double hulls by the year 2020 [Ref. A.24]. The double hull is intended to protect against oil spills caused by hull punctures. However, the cellular arrangement of double hull tankers makes it difficult to clean and/or ventilate ballast spaces. As a result, there is a potential risk for explosion if fatigue cracking permits cargo oil or vapours gas to leak into these spaces.
- In response to the aforementioned concerns, classification societies have recently introduced rationally-based procedures for the fatigue design of structural details in steel ships [e.g., Ref. A.25-A.27], and they have made these procedures mandatory for novel structural configurations and large ship designs (e.g., tankers and container ships longer than 190 m, bulk carriers longer than 150 m). Included in these procedures are: (i) simplified methods for quantifying the fatigue performance of structural details in common problem areas, (ii) spectral-based methods for quantifying the fatigue performance of structural details that cannot be properly analyzed with simplified methods, and, (iii) qualitative guidelines for optimizing the fabrication and fatigue performance of structural details. Similar procedures are being developed by several navies [Ref. A.28, A.29] and an ad hoc working group of the International Association of Classification Societies (IACS).

A.4 SCOPE OF THE GUIDE

A.4.1 Layout of the Guide

The design aid information in the Fatigue Resistant Detail Design Guide for Ship Structures is organized into four parts, entitled:

- Part A - Introduction
- Part B - Catalogue of Fatigue Resistant Details
- Part C - Fatigue Strength Assessment
- Part D - Fatigue Design Examples

Part A, the design Guide introduction, provides some background to illustrate the need for considering fatigue in the design process and describes the objectives and layouts of the remainder of the Guide.

Part B and Part C of the Guide present information to assist the designer to consider fatigue design throughout the detail design process. The three levels of design procedure sophistication which are presented in this Guide include:

- Level 1 - selection of fatigue resistant structural detail geometries and layouts;
- Level 2 - a simplified fatigue design procedure; and,
- Level 3 - an in-depth fatigue design procedure.

The increasing levels of the design procedures not only correspond to increased analytical detail and accuracy but also cost in terms of the time, effort and data required for implementation.

Part B of the Guide presents the Level 1 fatigue design aid in the form of a catalogue of fatigue resistant design details. The catalogue itemizes all key structural connections encountered in the midship area of various vessel types and provides geometric and structural arrangement suggestions to improve the fatigue performance of the detail of interest. The Level 1 structural detailing process may be used as a first step prior to a more sophisticated numerical fatigue strength assessment process or could be used alone as a simplified means of improving the fatigue performance of a detail without quantifying the net effect.

Part C of the fatigue design Guide presents rationally-based methods for quantifying the fatigue strength of welded structural details and provides detailed guidance and instructions for applying these methods. Level 2 and Level 3 fatigue design procedures are both presented in Part C due to their common and interchangeable procedural steps. The Level 2 fatigue strength assessment approach makes extensive use of empirical design equations to simplify the analysis process, whereas, the Level 3 fatigue strength assessment approach makes full use of the statistical information and numerical modeling techniques commonly available to designers.

Part D of the fatigue design Guide presents worked design examples used to demonstrate the application of the three levels of design procedure sophistication.

A.4.2 Application of the Guide as a Design Aid

The ship design process may be described schematically as shown in [Figure A.4.1](#), and the fatigue design Guide has been assembled to aid in the detail design process. The Guide includes information to promote good detail design early in the detail design stage, as well as analytical techniques, which may be used in an iterative manner to optimize the scantlings of selected details. The fatigue strength assessment techniques may also indicate a need for the designer to rework the overall structural design (i.e., return to the preliminary design stage) if acceptable fatigue performance cannot be achieved by optimization at the detailed level.

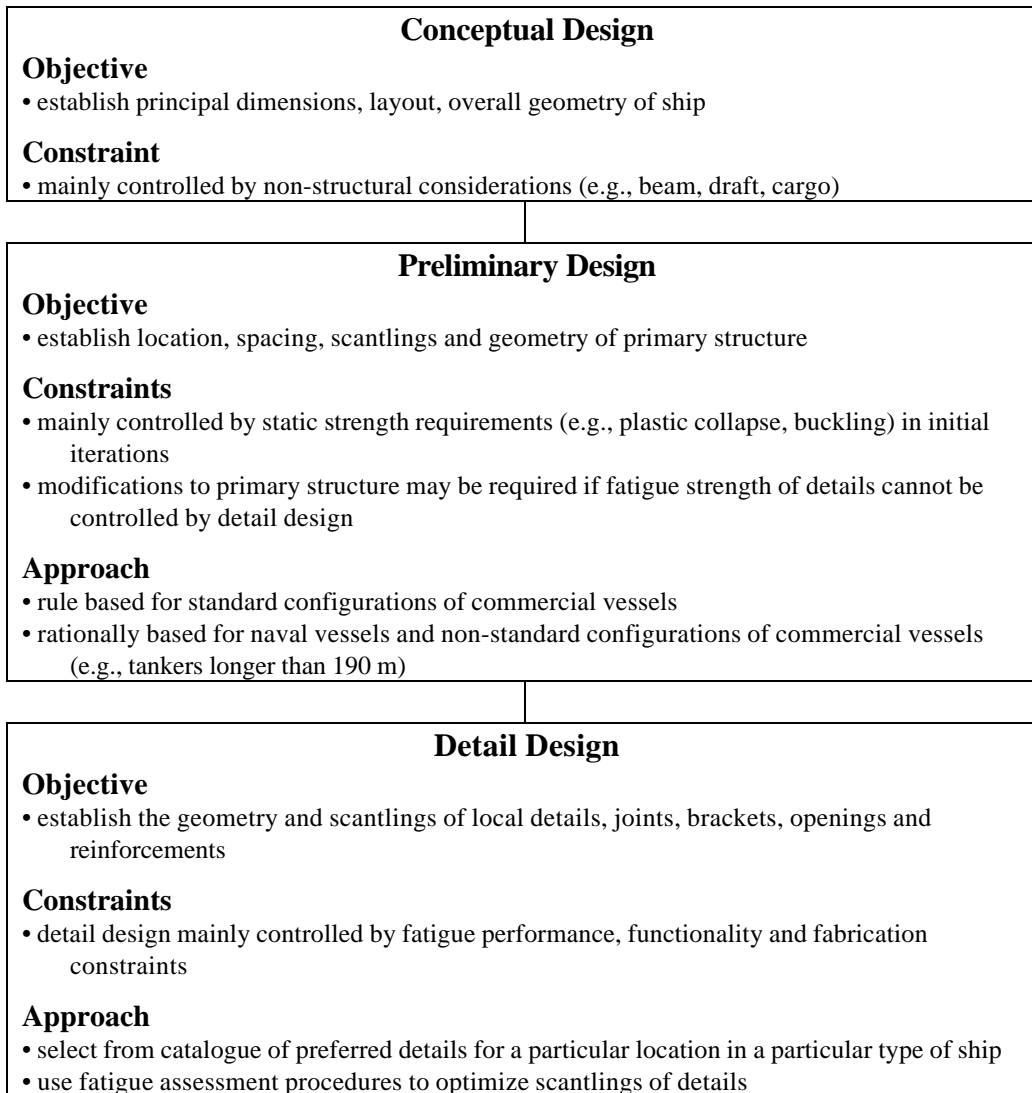


Figure A.4.4.1: Application of Fatigue-Design Guide in Ship Design Process

As previously noted for fatigue cracking in general, the service life of a ship may be divided into three distinct phases including:

- (i) the teething phase in which fatigue cracks initiate from fabrication defects at a decreasing rate with time;
- (ii) the stable phase in which fatigue cracks initiate randomly at a constant rate with time; and
- (iii) the aging phase in which the rate of fatigue cracking increases with time because of cumulative fatigue damage and other structural degradation (e.g., wear and corrosion).

These three phases lead to a cracking rate vs. time profile that is sometimes referred to as the “Bathtub Model” of service life.

This Guide is intended to produce designs for steel welded ship structural details in which the aging phase starts towards the end of a ship's service life and in which fatigue cracking rarely occurs during the teething and stable phases, provided the ships are well fabricated and maintained.

This Guide does not account for the possible onset of unstable fracture from a fatigue crack. In order to guard against this possibility, ships should be designed as damage tolerant structures (i.e., structures that can sustain maximum design loads without failure until damage is detected and repaired). Damage tolerance can be achieved by appropriate material selection at the design stage, the provision of multiple or redundant load paths, and the use of readily inspected structural details. In addition, damage tolerance analysis can be used to quantitatively assess the residual fatigue lives and residual strength of ship structures with fabrication defects or in-service cracks. Fracture toughness requirements for ship steels and welding consumables are given in various Naval and Classification Society documents [e.g., Ref. A.12, A.30, A.31], while procedures for assessing the damage tolerance and redundancy of ship structures are given in SSC Reports 402 [Ref. A.32] and SSC 354 [A.33], respectively.

Each part of this Guide builds upon the results of previous Ship Structure Committee projects [A.32-A.41] and recent efforts of classification societies and navies [e.g., Ref. A.27]. Wherever possible, the developers of this Guide have tried to incorporate the best features of the previous work and to reconcile major procedural differences. In order to maintain a practical focus, the Guide has been reviewed by designers and fabricators of ships, and their recommendations have been incorporated. Sufficient commentary, guidance, and references have been included to make the Guide self-contained. However, the developers of this Guide have assumed that its users will be trained in ship structural analysis and will have at least a rudimentary knowledge of metal fatigue.

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PART B - CATALOGUE OF FATIGUE RESISTANT DETAILS

*For this section, the authors have drawn heavily on the work of the Classification Societies, specifically **Lloyd’s Register of Shipping, Det Norske Veritas, American Bureau of Shipping, and the International Association of Classification Societies (IACS)**, as well as Canadian and UK Navy information. We are indebted to Lloyd’s Register of Shipping, Det Norske Veritas, the American Bureau of Shipping, IACS, Saint John Shipbuilding Limited and the Canadian Navy for permission to use their source materials and drawings. These resources are identified in each of the data sheets herein. We recommend the references in this section for those wishing to explore this area further.*

B.1 INTRODUCTION

B.1.1 Background

The fatigue resistance of structural details subjected to a given cyclic load is primarily geometry dependent. That is, discontinuous load paths, rapid geometric transitions or misalignments and poor weld geometries are the most significant factors which influence the fatigue life of a structural detail subjected to a given cyclic load environment. Therefore, the most effective way of extending the fatigue life of a structural connection is through proper detailing.

The most common reason for poor fatigue resistance is inappropriate detailed design. [Figures B.1.1 and B.1.2](#), extracted from previous SSC Reports [Ref. B.1], show typical crack patterns that have been found in tankers and bulk carriers, respectively. These instances of fatigue damage can be avoided with additional attention to detailing for fatigue resistance in the design process.

Design for good fatigue resistance can proceed through a series of levels, or steps. The first level in the fatigue design process involves the selection of fatigue-tolerant details. This section of the Fatigue Design Guide presents a catalogue of structural details for a range of vessel types, with suggested “good practice” to improve fatigue performance. A detail “rating” system is included in the catalogue to indicate the relative performance and costs associated with alternative structural details. The purpose of this catalogue is to provide the practicing naval architect with a readily accessible, guide on how to improve the fatigue performance of a structural connection.

B.1.2 Objective

The Guide is intended to be utilized by engineers and naval architects in ship structural design (and fabrication) to promote good design practice, reduce the likelihood of premature fatigue failure, and identify the relative fatigue resistance of structural details.

The objective of Part B of the Guide - the Catalogue - is to provide guidance in the preliminary design of structural connections to improve their fatigue performance. The Catalogue therefore, provides a Level 1 fatigue design process.

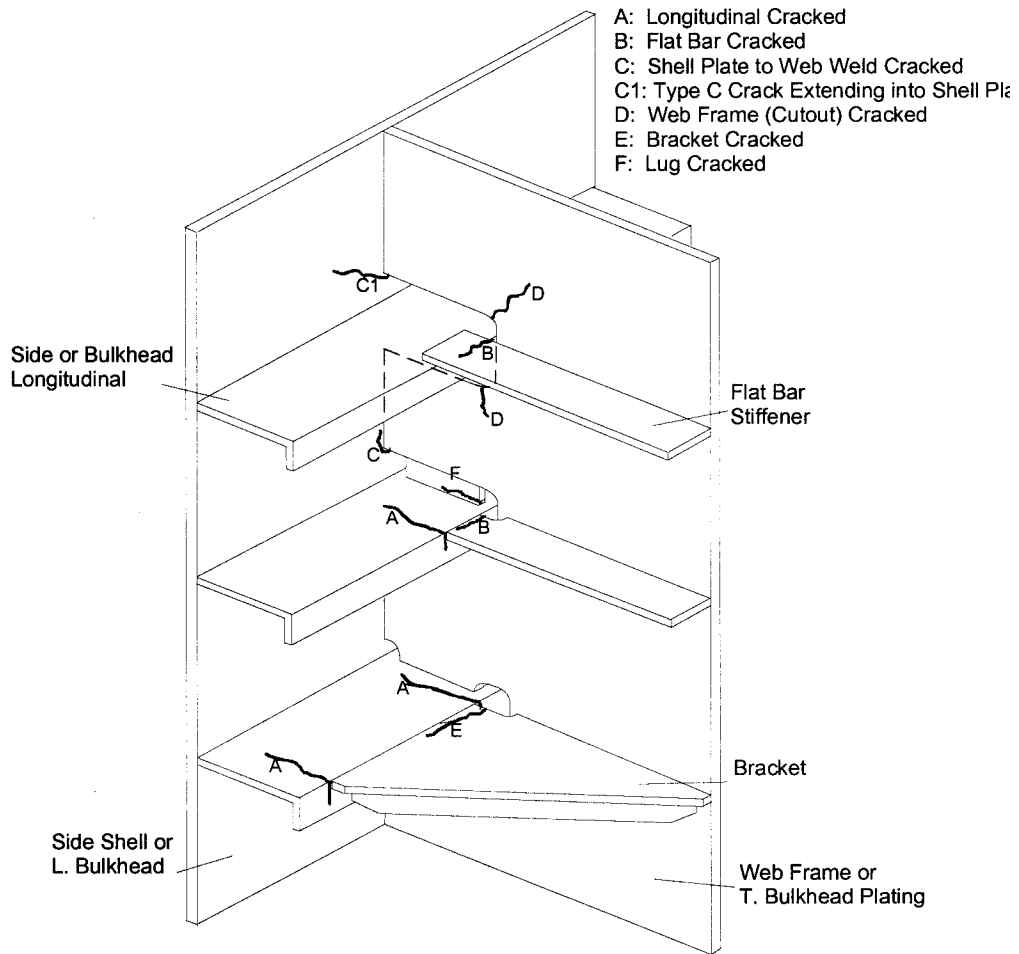


Figure B.1.1: Fatigue Crack Locations and Orientation in Typical Tanker Structure

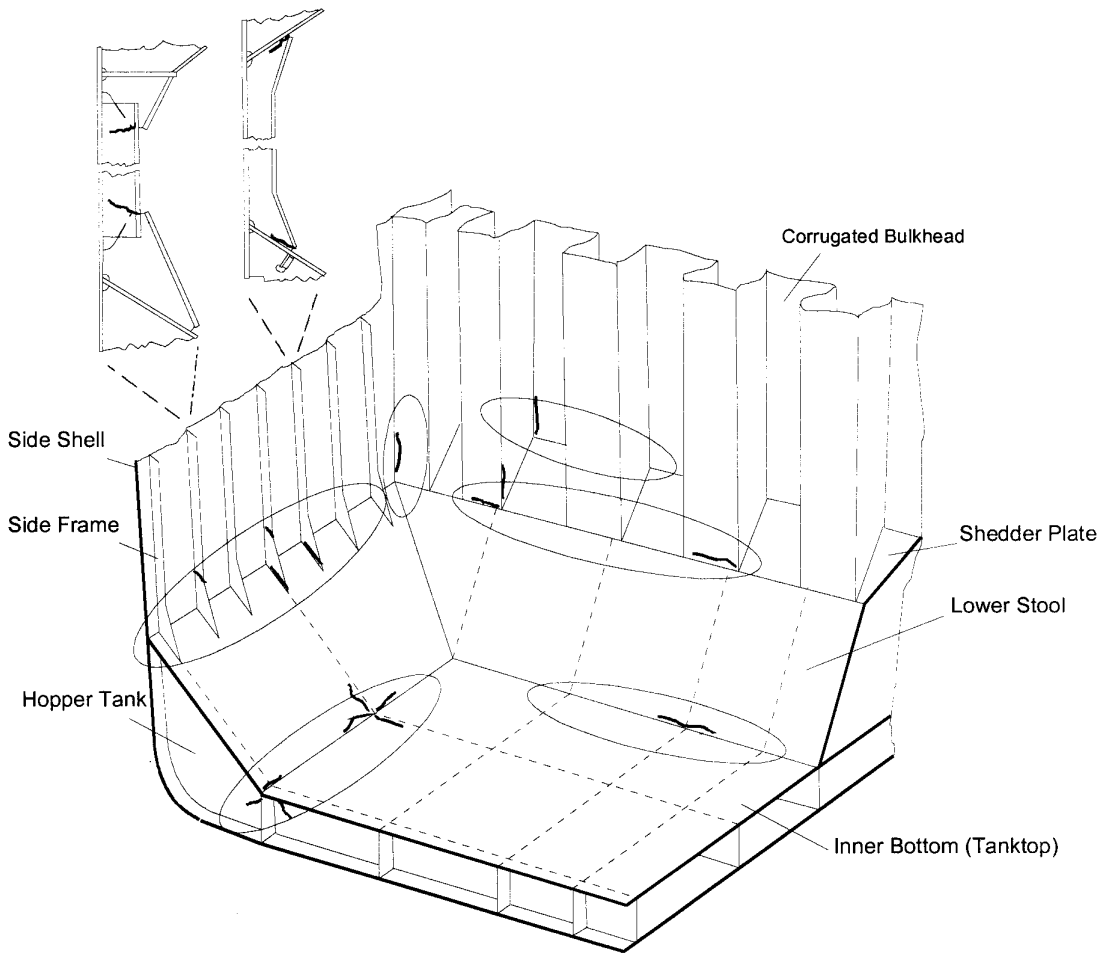


Figure B.1.2: Fatigue Crack Locations and Orientation in Typical Bulk Carrier Structure

B.1.3 Scope

With literally hundreds of structural detail configurations in existence, this Catalogue can only provide examples of the most common details. For these details the Catalogue provides geometric limitations and arrangement alternatives which may be used directly to improve the fatigue performance of ship structural details. However the Catalogue can also be used to indicate the *type* of improvements that may be considered for details not presented by applying the principles that are shown herein.

The Catalogue is presented in four sections, each section providing information on a different ship type. The four generic ship types dealt with in the catalogue are:

- Double Hull Tankers
- Bulk Carriers
- Container Ships
- Warships

The Catalogue focuses on details associated with the midship section of the vessels of interest, since this is generally the most critical area. All welded connections which are potentially prone to fatigue failure are presented. The Catalogue illustrates good detailing practice and/or alternative configurations for those connections considered to be at high risk. The evaluation of a connection as being at high or low risk is solely based on the geometry of the connection and thus the stress concentration it represents. The severity of the loadings seen by any specific detail will be determined by location-specific factors, and thus an inherently high-risk connection may or may not require modification depending on its application. Many structural details are common to more than one ship type. The catalogue illustrates the detail for each ship type so that the user has complete information within each section. Similar details, used in different ship types (e.g., tanker, bulk carrier, etc.), will have the same detail identification number. For example, a transverse floor, bottom longitudinal connection is Detail # 2 for both tankers and bulk carriers.

The remainder of this section of the Guide provides background and guidance to the designer on how to interpret the information contained in the Catalogue (Section B.5). This information is presented in the following parts:

- Discussion of critical and non-critical details (Section B.2)
- Definitions of terminology used in the catalogue (Section B.3)
- A description of the catalogue layout (Section B.4)
- **The Catalogue of details** (Section B.5)
- Guidance on good design and construction practice (Section B.7)
- A list of the relevant reference material (Section B.8)

B.2 CRITICAL AND NON-CRITICAL DETAILS

The criticality of a structural detail is determined by assessing the risk to the continued safe operation of the vessel, posed by failure of the detail. Risk, in most general terms, is a combined measure of the probability (or likelihood) of failure and the consequence (or cost) of that failure. The designer is encouraged to consider both the probability of failure (in terms of fatigue strength of the detail) and the potential consequence of failure (which may be evaluated in terms of structural location or function).

B.2.1 Probability of Failure (Fatigue Resistance)

The primary objective of the Catalogue is the improvement of standard structural connections. For those midship connections considered prone to fatigue damage, design improvements or alternative arrangements are presented. In some instances, several levels of fatigue performance improvements are presented. Their relative fabrication and maintenance costs are rated to allow the designer to weigh the additional costs associated with reducing the probability of fatigue failure.

Those details that are not prone to fatigue problems, due to their geometric configuration, may be considered non-critical. The connection between the longitudinal and the bottom of a ship is an

example of a welded connection that, if good fabrication and maintenance practices are observed, should not be considered a likely location for fatigue cracking.

B.2.2 Consequences of Failure (Structural Function)

While the Catalogue does not specifically categorize the details based on the consequence of failure, the user is encouraged to examine the consequence of failure of the detail and associated structure when deciding if it is worthwhile to apply the fatigue performance improvements suggested. Since the cost of improving details can be significant, this is a most important consideration.

For example, a fatigue-prone detail in side or bottom plating warrants the cost associated with improved design practice more than the same detail in, say, a longitudinal bulkhead. The failure of a detail in the outer shell of a single hull tanker **in way** of a cargo tank can result in massive pollution, and therefore, the consequence of failure is great. The side shell plating in double hull tankers or bulk carriers is another critical area. The upper deck near amidships in way of large openings of warships should be considered a critical area. With these considerations in mind, the designer should use cost-benefit judgements to select the detail design improvements that provide appropriate levels of safety against fatigue failure.

B.2.3 Warship Details

Structural details on warships have been developed to meet the specific requirements of shock and impact, and, as a result, occasionally embody details that are designed to enhance the impact resistance. An example is the doubler plate on web frame/deck beam connections. However, this doubler plate is not necessarily an enhancement for fatigue purposes. The catalogue tries to present structural arrangements for warships that do not compromise the integrity of the connection while improving the fatigue performance of the detail.

B.3 CATALOGUE TERMINOLOGY

To ensure that the user understands the terminology used in the fatigue resistant design detail catalogue, the following list of definitions is supplied.

Critical Area - That area of the ship encompassing the global structure and comprising major structural elements such as bulkheads, decks, and in which the critical detail is included.

Critical Detail - The specific detail for which the fatigue performance is critical.

Good Practice - Detail design and fabrication practice which will improve the resistance of the detail to fatigue crack initiation and growth.

Detail Attributes - The measures that are being used to “rank” the alternative detail configurations.

Ranking - A simple measure of the relative strength of each attribute among the detail options described. Number 1 is always the best ranking for a detail.

Fatigue Performance - A measure of the resistance of the detail to fatigue crack initiation and growth. The lower the number, the better the fatigue performance.

Ease of Inspection - Ease with which inspection devices (e.g., ultra-sonic probes) can access and detect cracks, and ease with which the configuration (e.g., alignment) can be checked after construction for detection of defects. The lower the number, the easier to inspect.

Maintenance Cost - This is a measure of the degree to which the detail can be kept clean of damaging material, corrosive fluids, ease of painting and maintenance of an intact paint coating, as well as access for repair of defects. The lower the number, the easier to maintain.

Fabrication Cost - Reflects material, cutting and welding content, ease of fabrication. The most expensive detail has the highest number.

B.4 LAYOUT OF THE CATALOGUE

B.4.1 Introduction

The Catalogue is arranged in a hierarchical fashion starting with a midship section in which the global structural location of interest is identified. From this, the connection structural elements are selected to identify the detail of interest. The hierarchical layout of the catalogue is described in [Figure B.4.1.1](#) along with the nine-step process involved in making full use of its information.

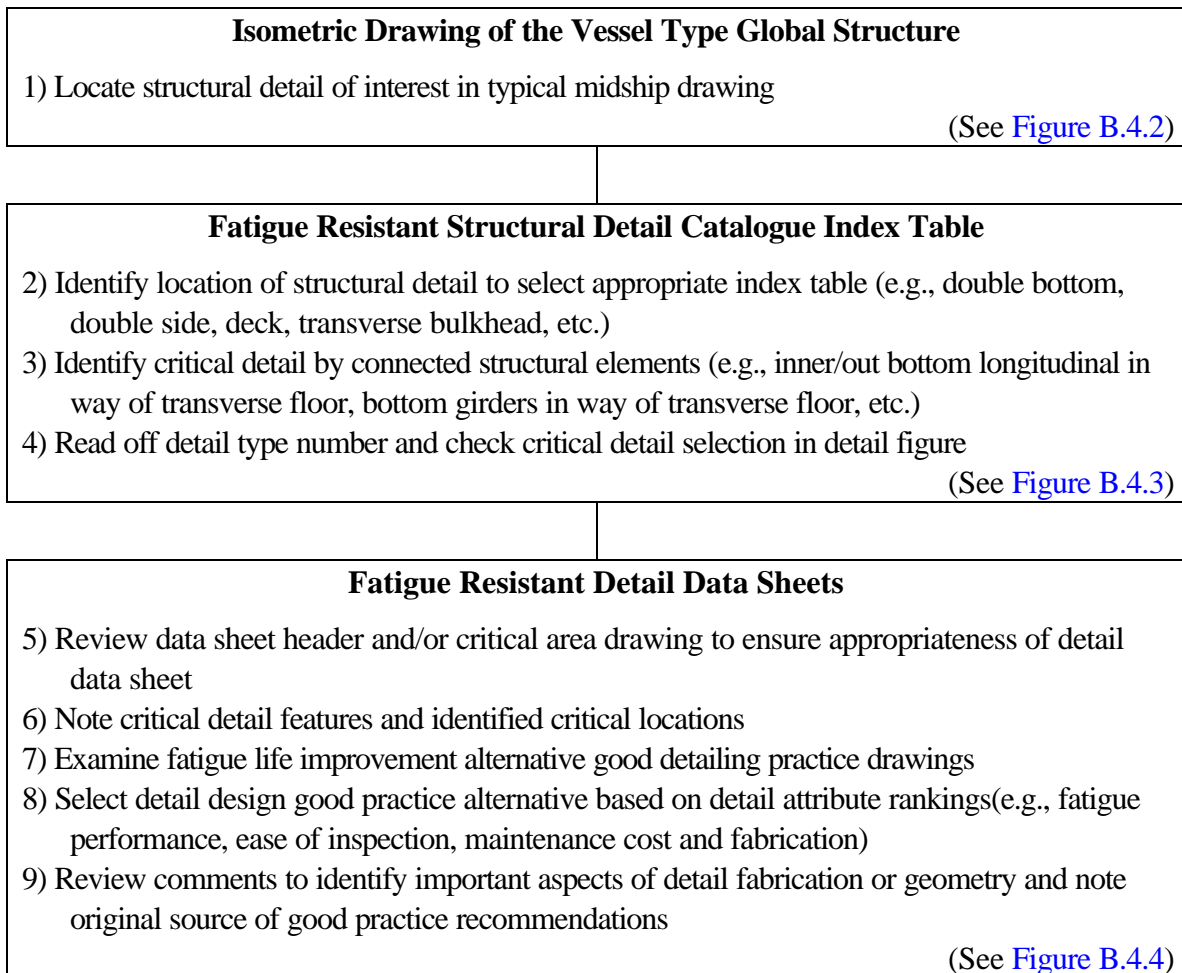


Figure B.4.1.1: Fatigue Design Guide Structural Detail Catalogue Layout

At the top level, the catalogue is divided into four, stand-alone sections by ship type as follows:

- Double Hull Tanker
- Bulk Carrier
- Container Ship
- Warship

Each vessel type specific section begins with an isometric illustration of a typical midship area of the vessel type in question (see [Figure B.4.1.2](#)) with the principal structural elements defined. This structural representation should aid in locating the structural detail of interest and relating it to the terminology used to describe it in the fatigue resistant design detail catalogue.

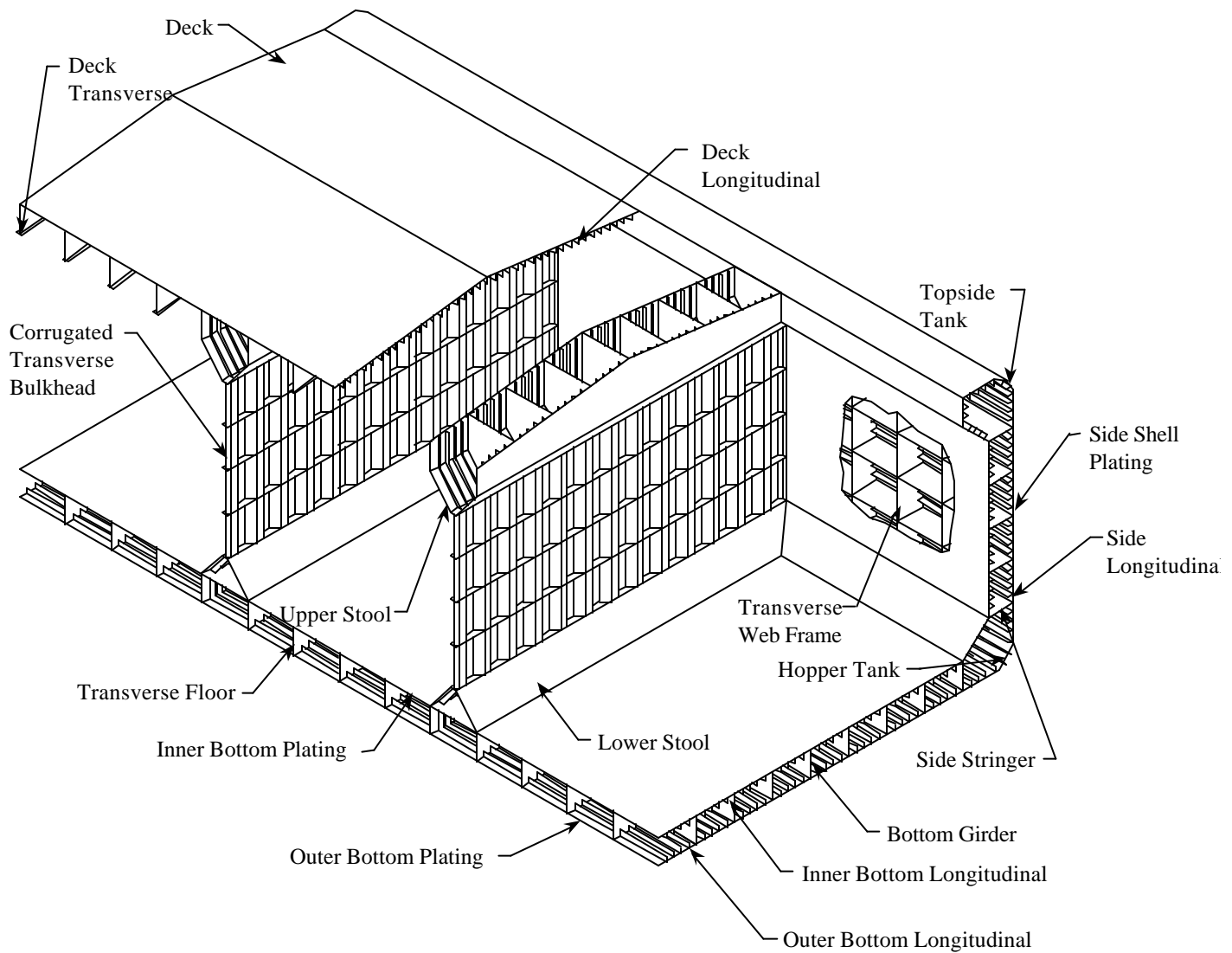


Figure B.4.1.2: Typical Double Hull Tanker Midship Global Structural Arrangement

The fatigue resistant structural detail catalogue index tables, similar to that shown in [Figure B.4.1.3](#), group the critical structural details by location (e.g., double bottoms, double sides, deck and transverse bulkheads). The terminology for the location names used in the catalogue index tables are given in the global midship structure illustration (see [Figure B.4.1.4](#)) located at the beginning of each vessel type catalogue section.

A catalogue index table is available for all significant structural locations. For the double hull tanker structure, catalogue index tables have been assembled for:

- double bottom structure,
- double side structure, and
- transverse bulkhead structure,
- deck structure.

The catalogue index tables are arranged to describe all of the connections in the structural location being described. Since a structural connection is defined as the connection of two or more structural members, the catalogue index table identifies connection details based on the structural elements or members joined in its first two columns.

The third and fourth columns of the catalogue index tables contain a detail type reference and a figure illustrating the structure surrounding the detail of interest. The detail type reference number directs the user how to find more information on developing a fatigue resistant structural detail for the connection of interest. Since experience has shown that not all structural member connection details are prone to fatigue damage, the detail type reference may be one of the following two forms:

- 1) for connections where fatigue is not typically a critical design issue, the detail type reference directs the user to [Section B.7](#) which discusses good fabrication and assembly practice to preclude fatigue cracking; or,
- 2) for connections which are considered susceptible to fatigue damage, the detail type reference directs the user to appropriate fatigue resistant detail data sheets, similar to that in [Figure B.4.4](#).

While each vessel type section is completely self-contained, similar fatigue-susceptible details used in different vessel types have been given the same detail reference number. Once a critical fatigue detail is identified the alternative designs can be reviewed by locating the associated fatigue resistant detail data sheet within the catalogue.

The data sheets contain information which the engineer or naval architect will use to select appropriate detail configurations and to understand the implications of the selection. The data sheet has five information areas including:

- a data sheet title block;
- a critical area and detail illustration;
- detail design good practice recommendations;
- a detail attribute ranking matrix; and,
- a detail specific comment section.

Critical Structural Details		Detail Type#	Detail Figure
Inner/Outer Bottom Longitudinals	Transverse Floor Stiffener - Asymmetric Longitudinal - Symmetric Longitudinal	1 2	
	Transverse Bulkhead - Plane Oil Tight Transverse Bulkhead - Corrugated Oil Tight Transverse Bulkhead	3 4	
Bottom Longitudinals	Transverse Ring Web - Transverse Ring Web Tripping Bracket	5	
Bottom Girders	Transverse Bulkhead - Plane Oil Tight Transverse Bulkhead - Plane Oil Tight Transverse Bulkhead Vertical Stiffener End Brackets and High Strength Steel Girders	6 7	
Inner/Outer Bottom Plating	Hopper Tank and Transverse Floor - Welded Knuckle - Radiused Knuckle	8 9	
	Bottom Longitudinal	Section B.7	
	Bottom Girder	Section B.7	
	Transverse Floor	Section B.7	
	Transverse Bulkhead	Section B.7	

Figure B.4.1.3: Typical Fatigue Resistant Structural Detail Catalogue Index Table



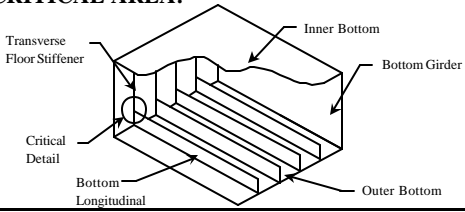
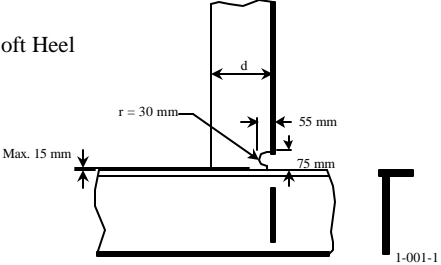
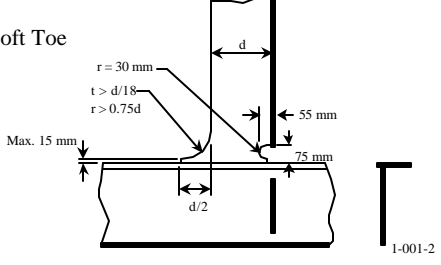
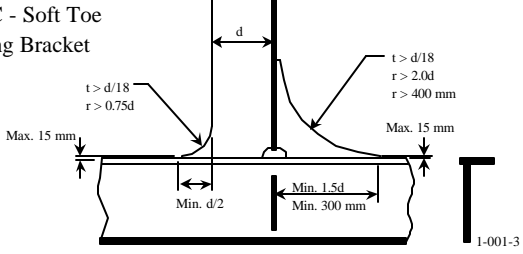
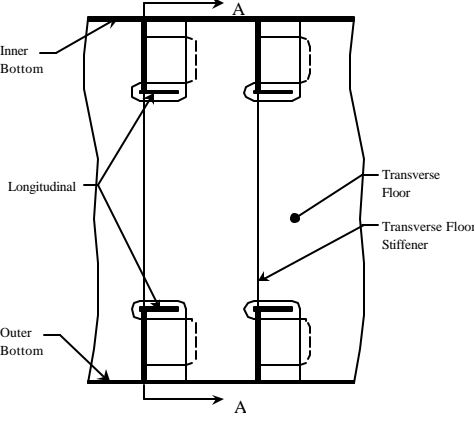
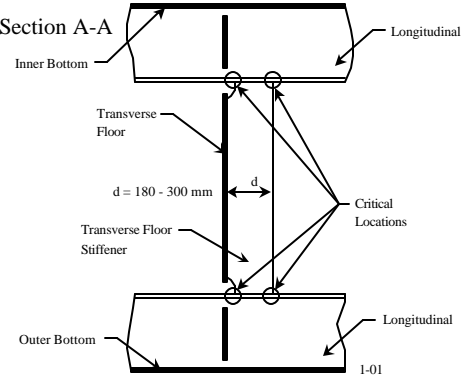
<p>Vessel Type: Double Hull Oil Tankers Critical Area: Double Bottom Critical Detail: Intersection of Inner/Outer Bottom Longitudinal and Transverse Floor Stiffener Detail Description: Asymmetric Longitudinal</p>	 Ship Structure Committee 	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">1</h1> <p>Rev. 001-250199</p>																							
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Heel</p>  <p>Detail Design B - Soft Toe and Soft Heel</p>  <p>Detail Design C - Soft Toe and Soft Backing Bracket</p> 																								
<p>CRITICAL DETAIL:</p>  <p>Section A-A</p> 	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2">Detail Attributes</th> <th colspan="3">Design Detail Rankings</th> </tr> <tr> <th>A</th> <th>B</th> <th>C</th> </tr> </thead> <tbody> <tr> <td>Fatigue Performance</td> <td>3</td> <td>2</td> <td>1</td> </tr> <tr> <td>Ease of Inspection</td> <td>1</td> <td>1</td> <td>2</td> </tr> <tr> <td>Maintenance Cost</td> <td>1</td> <td>2</td> <td>3</td> </tr> <tr> <td>Fabrication Cost</td> <td>1</td> <td>2</td> <td>3</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings			A	B	C	Fatigue Performance	3	2	1	Ease of Inspection	1	1	2	Maintenance Cost	1	2	3	Fabrication Cost	1	2	3
Detail Attributes	Design Detail Rankings																								
	A	B	C																						
Fatigue Performance	3	2	1																						
Ease of Inspection	1	1	2																						
Maintenance Cost	1	2	3																						
Fabrication Cost	1	2	3																						
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between the transverse floor stiffener and the backing bracket is to be maintained. - A wrap around weld, free of undercut or notches, around the heel and toe connections of the floor stiffener and backing bracket. 																									

Figure B.4.1.4: Typical Fatigue Resistant Detail Data Sheets

The data sheet title block includes descriptions of the vessel type, critical area, critical detail and any further detail description data to identify the connection of interest along with the detail type reference number. The critical area and detail illustrations are included in the data sheet to locate and describe fatigue damage susceptible elements of the critical detail. This is accomplished with a global structure illustration of the critical area including near by structural elements to help visualize the detail of interest. Below the critical area illustration, the data sheet includes a close-up of the critical detail illustrating the critical locations at which fatigue cracking is most likely to occur.

The drawings on the right side of the data sheet illustrate “good practice” recommendations to improve the fatigue performance of the structural connection. The recommendations include both general items such as: soft toe and heel geometries, scallop sizes and ensuring member/load path continuity, along with detail specific suggestions such as the addition of stiffening elements or flange tapering/chamfering geometry.

Limitations on geometry or detail critical tolerances and dimensions, are also included in the drawings where appropriate.

Most of the data sheet “good practice” recommendations include more than one fatigue resistant detail alternative. These alternatives are offered as stepwise improvements in the fatigue performance of the detail, but there are costs associated with improved fatigue strength. The detail attribute ranking matrix, at the bottom right of the data sheet, tries to qualitatively express the relative costs of the detail improvement alternatives in terms of:

- fatigue performance,
- ease of inspection, and
- maintenance cost,
- fabrication cost.

While the relative rankings in the detail attribute matrix are subjective, they serve their intended purpose; which is to highlight the additional collateral costs or burdens associated with improving fatigue performance. The relative cost rankings could be used to help in the selection of a detail for a specific application as discussed in [Section B.3](#).

Located at the bottom of the data sheet is a comment block that provides detail specific alignment or fabrication practice recommendations to improve the fatigue performance of the connection.

B.5 FATIGUE RESISTANT DETAIL CATALOGUE

B.5.1 Tanker Structure Details

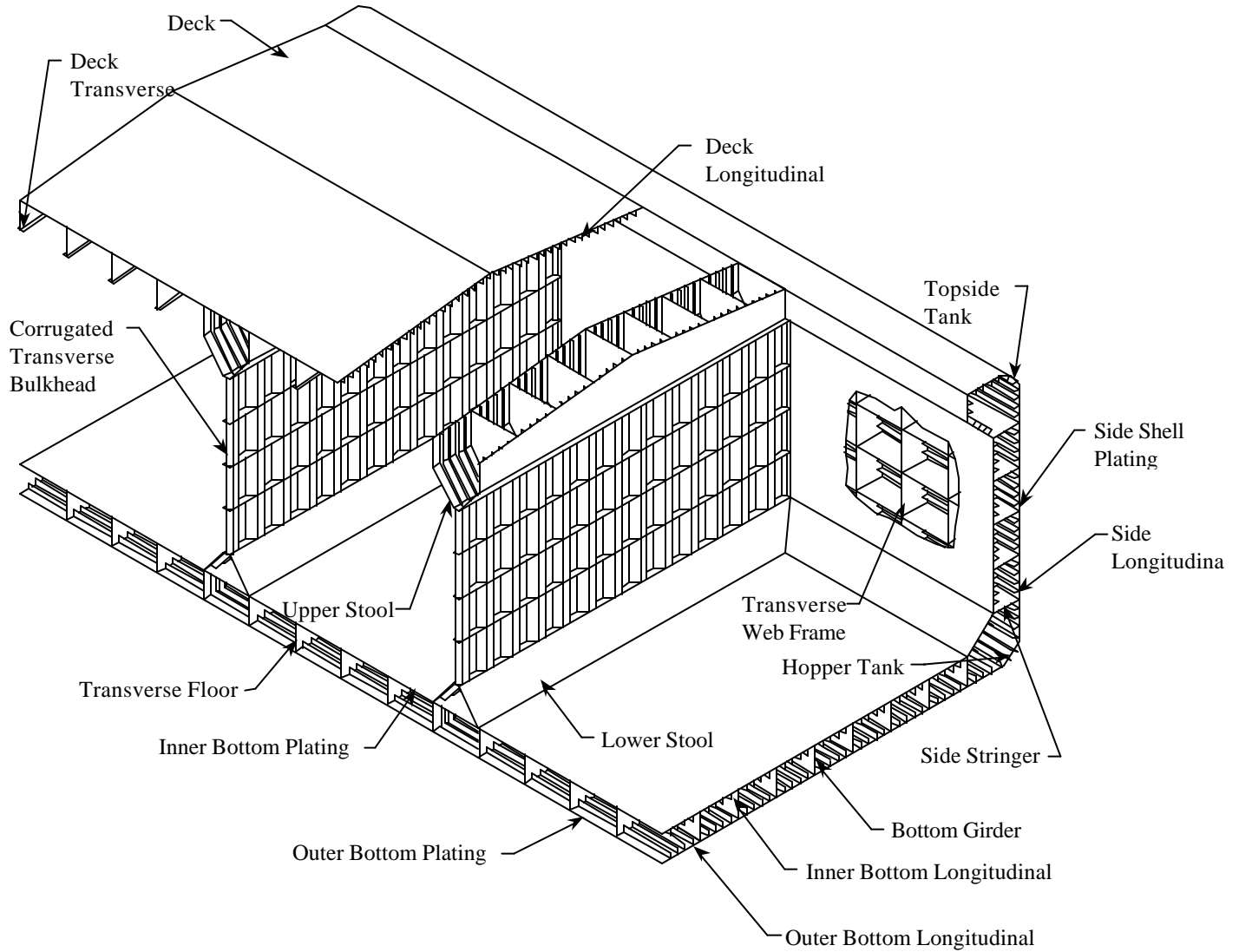


Figure B.5.1: Typical Midship Section Nomenclature for Double Hull Tankers

Table B.5.1 (a): Critical Structural Details for Double Bottom

Critical Structural Details		Detail Type#	Detail Figure
Inner/Outer Bottom Longitudinals	Transverse Floor Stiffener - Asymmetric Longitudinal - Symmetric Longitudinal	1 2	<p>Transverse Bulkhead Transverse Floor Bottom Girder Critical Detail #3 & #4 Transverse Floor Stiffener Critical Detail #1 & #2 Outer Bottom Bottom Longitudinal</p>
	Transverse Bulkhead - Plane Oil Tight Transverse Bulkhead - Corrugated Oil Tight Transverse Bulkhead	3 4	
Bottom Longitudinals	Transverse Ring Web - Transverse Ring Web Tripping Bracket	5	<p>Critical Detail #5 Transverse Ring Web Bottom Longitudinal</p>
Bottom Girders	Transverse Bulkhead - Plane Oil Tight Transverse Bulkhead - Plane Oil Tight Transverse Bulkhead Vertical Stiffener End Brackets and High Strength Steel Girders	6 7	<p>Transverse Bulkhead Critical Detail #6 & #7 Bottom Girder Outer Bottom</p>
Inner/Outer Bottom Plating	Hopper Tank and Transverse Floor - Welded Knuckle - Radiused Knuckle	8 9	<p>Critical Detail #8 & #9 Side Shell Hopper Tank Inner Bottom Outer Bottom Bottom Girder</p>
	Bottom Longitudinal	Section B.7	<p>Inner Bottom Non Critical Detail Bottom Girder</p>
	Bottom Girder	Section B.7	<p>Outer Bottom Bottom Longitudinal</p>
	Transverse Floor	Section B.7	<p>Transverse Bulkhead Non Critical Detail Transverse Floor</p>
Transverse Bulkhead	Section B.7	<p>Outer Bottom</p>	

Table B.5.1 (b): Critical Structural Details for Double Side

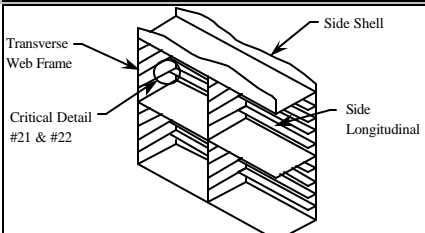
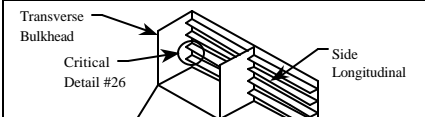
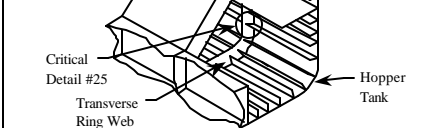
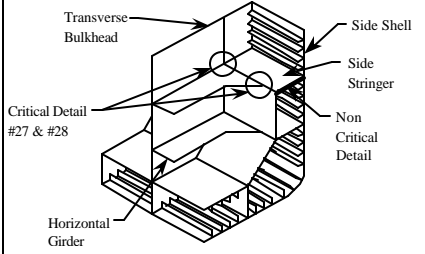
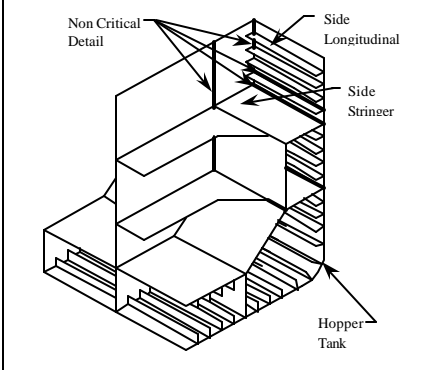
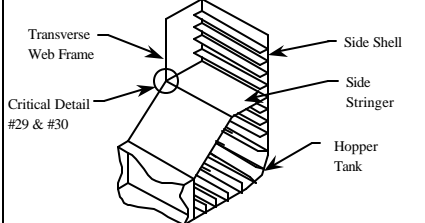


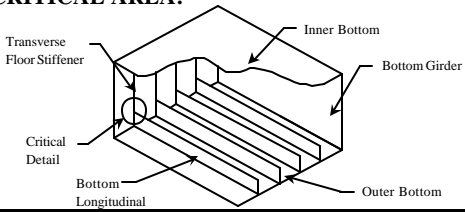
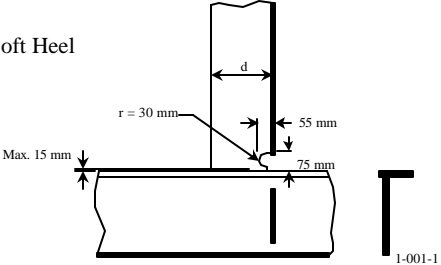
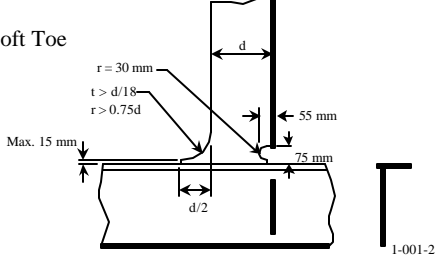
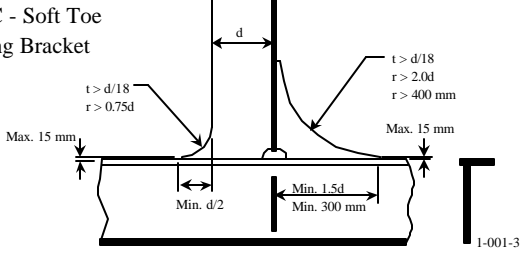
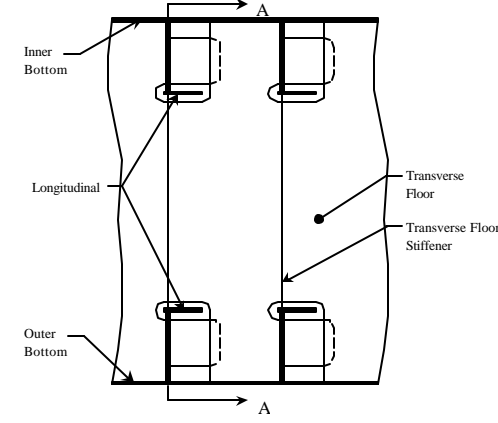
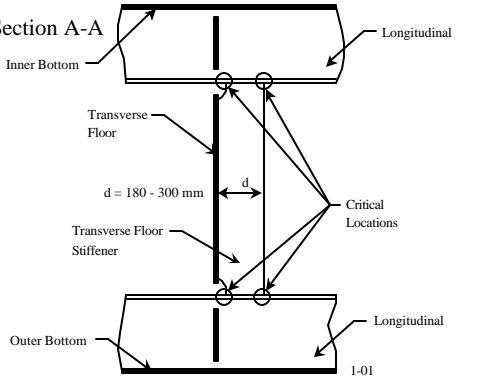
Critical Structural Details		Detail #	Detail Figure
Side Shell Longitudinals	Transverse Web Frame - Asymmetric Longitudinal - Symmetric Longitudinal	21 22	
	Transverse Ring Web - Transverse Ring Web Tripping Bracket	25	
	Transverse Bulkhead - Transverse Bulkhead Horizontal Stiffener	26	
Side Stringers	Transverse Web Frames	Section B.7	
	Transverse Bulkhead - Plane Oil Tight - Corrugated Oil Tight	27 28	
Side Shell Plating	Hopper/Topside Tank	Section B.7	
	Side Longitudinals	Section B.7	
	Side Stringers	Section B.7	
	Transverse Web Frames	Section B.7	
	Transverse Bulkhead	Section B.7	
Transverse Web Frames	Hopper/Topside Tanks - Welded Side Longitudinal Bulkhead Plating - Knuckled Side Longitudinal Bulkhead Plating	29 30	



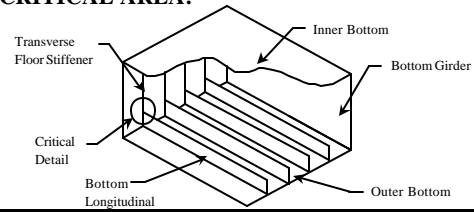
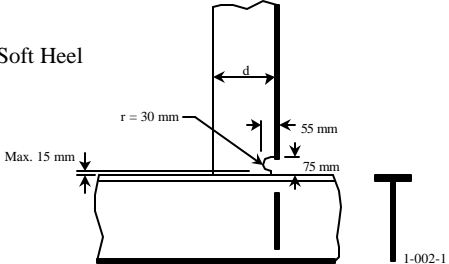
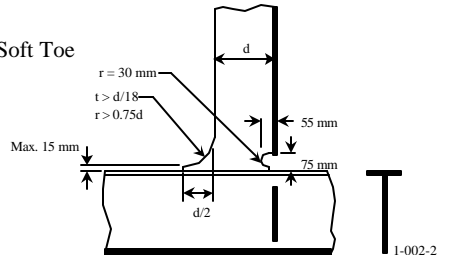
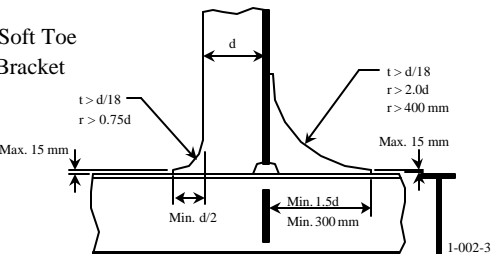
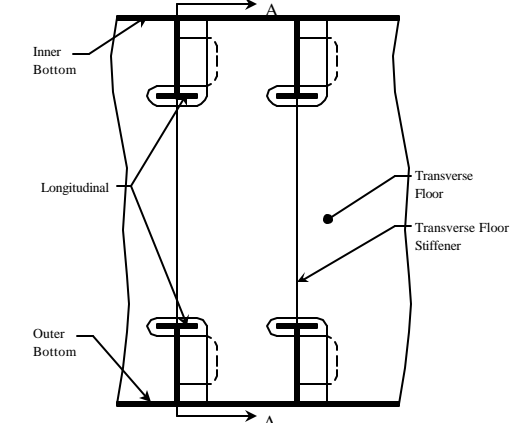
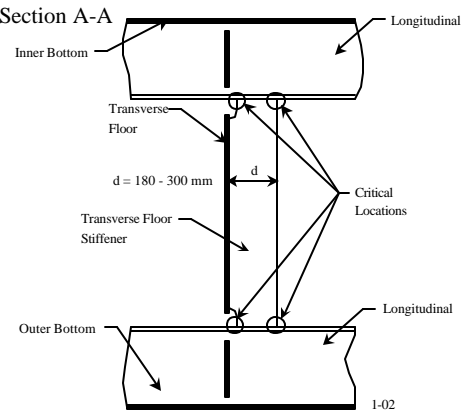
Table B.5.1 (c): Critical Structural Details for Transverse Bulkhead



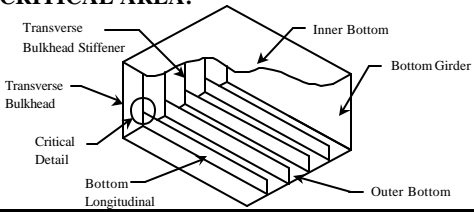
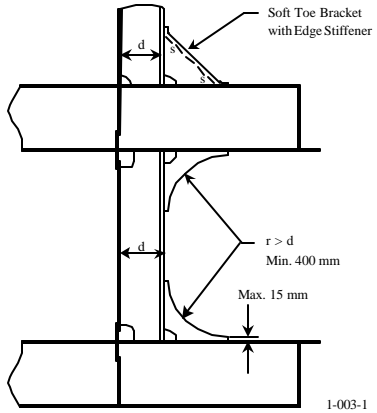
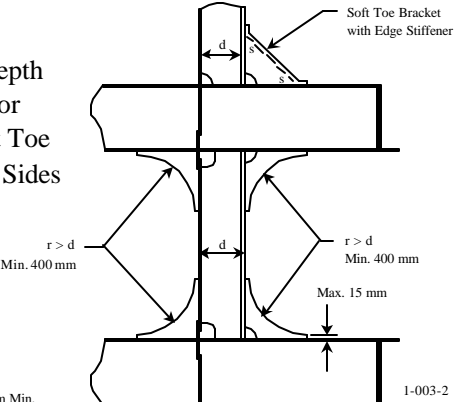
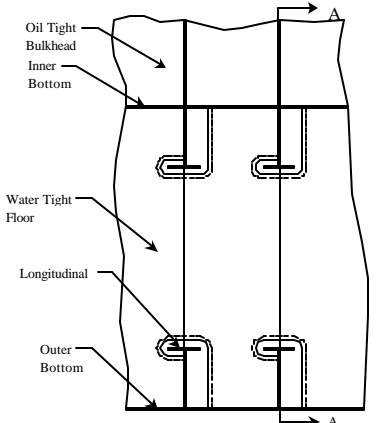
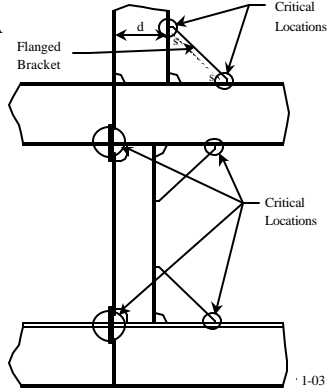
Critical Structural Details		Detail #	Detail Figure
Upper/Lower Bulkhead Stool	Deck/Bottom Plating	Section B.7	
	Topside/Hopper Tanks	Section B.7	
Transverse Bulkhead	Upper/Lower Bulkhead Stool	Section B.7	
Transverse Bulkhead Plating	at Corrugations	Section B.7	



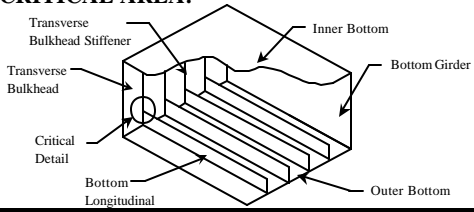
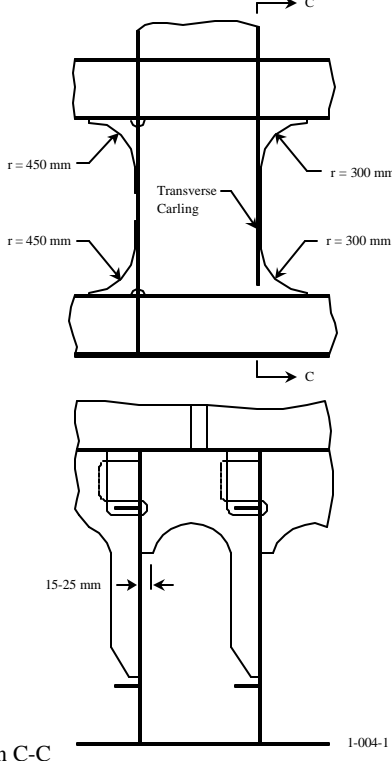
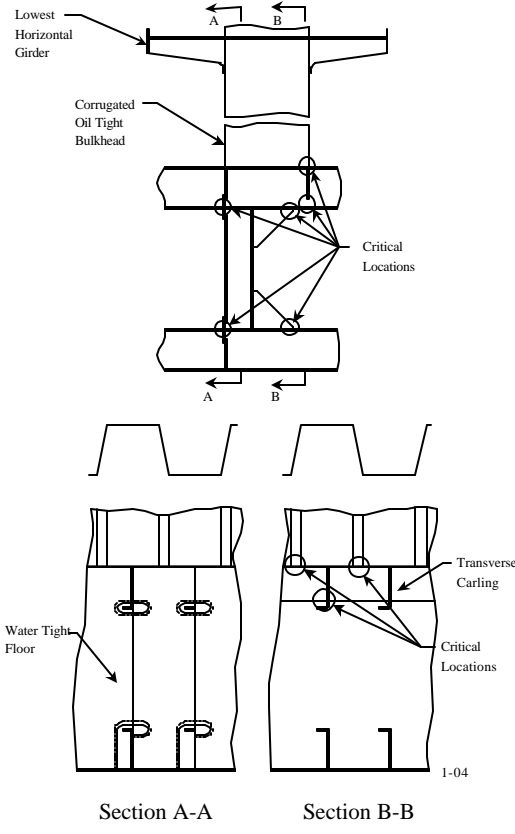
Table B.5.1 (d): Critical Structural Details for Deck



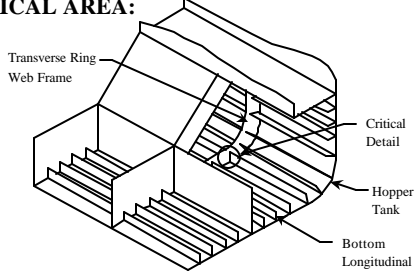
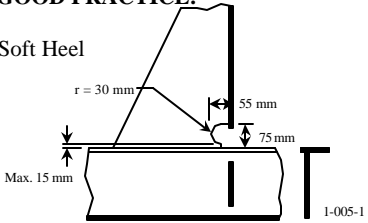
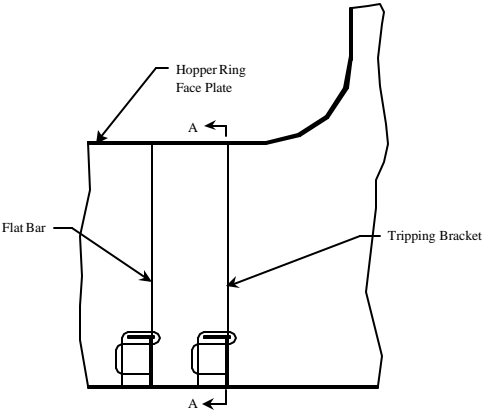
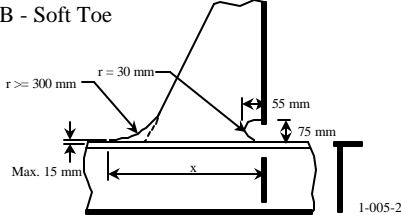
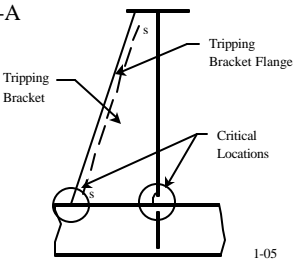
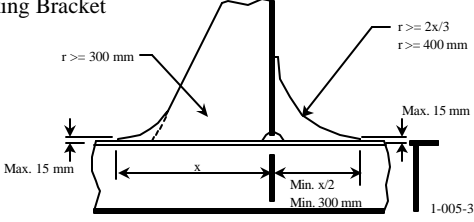
Critical Structural Details		Detail #	Detail Figure
Deck Plating	Topside Tank	Section B.7	
	Side Shell Plating	Section B.7	
	Transverse Web Frame	Section B.7	
Deck Transverse End Brackets	Transverse Web Frames - Toe Connection	61	
Deck Transverse	Deck Longitudinals	Section B.7	



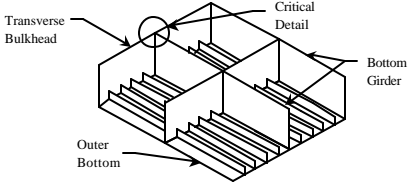
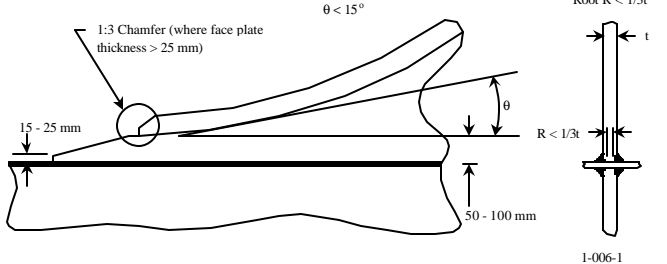
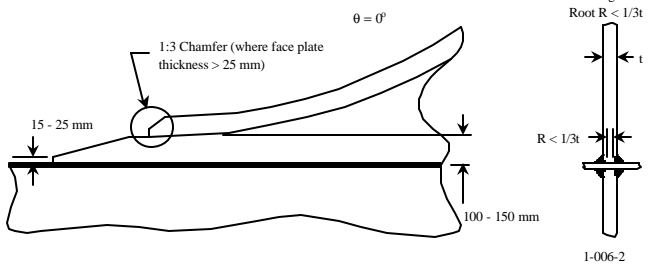
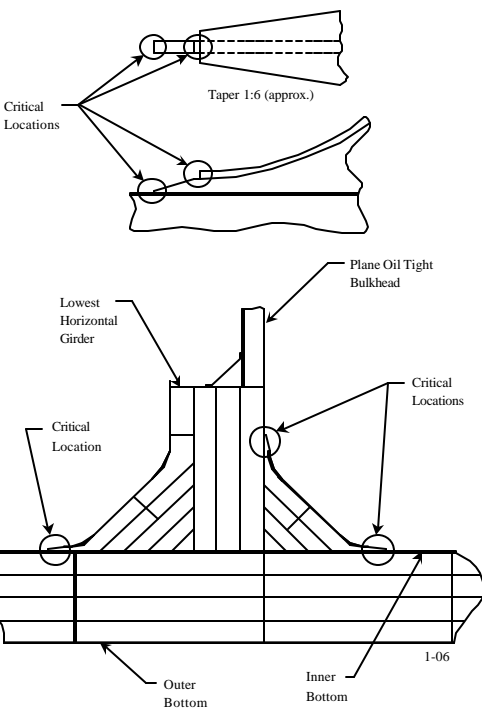
<p>Vessel Type: Double Hull Oil Tankers Critical Area: Double Bottom Critical Detail: Intersection of Inner/Outer Bottom Longitudinal and Transverse Floor Stiffener Detail Description: Asymmetric Longitudinal</p>	 Ship Structure Committee 	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">1</h1> <p>Rev. 001-250199</p>																							
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Heel</p>  <p>Detail Design B - Soft Toe and Soft Heel</p>  <p>Detail Design C - Soft Toe and Soft Backing Bracket</p> 																								
<p>CRITICAL DETAIL:</p>  <p>Section A-A</p> 	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2">Detail Attributes</th> <th colspan="3">Design Detail Rankings</th> </tr> <tr> <th>A</th> <th>B</th> <th>C</th> </tr> </thead> <tbody> <tr> <td>Fatigue Performance</td> <td>3</td> <td>2</td> <td>1</td> </tr> <tr> <td>Ease of Inspection</td> <td>1</td> <td>1</td> <td>2</td> </tr> <tr> <td>Maintenance Cost</td> <td>1</td> <td>2</td> <td>3</td> </tr> <tr> <td>Fabrication Cost</td> <td>1</td> <td>2</td> <td>3</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings			A	B	C	Fatigue Performance	3	2	1	Ease of Inspection	1	1	2	Maintenance Cost	1	2	3	Fabrication Cost	1	2	3
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<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between the transverse floor stiffener and the backing bracket is to be maintained. - A wrap around weld, free of undercut or notches, around the heel and toe connections of the floor stiffener and backing bracket. 																									



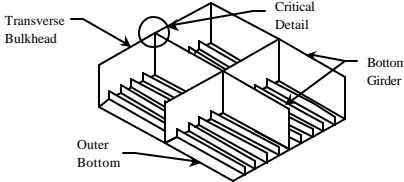
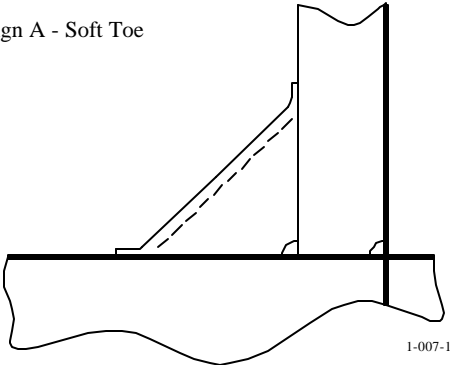
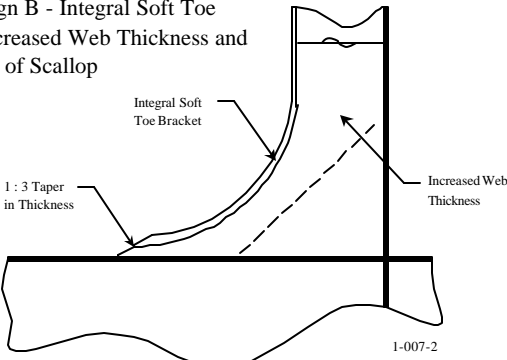
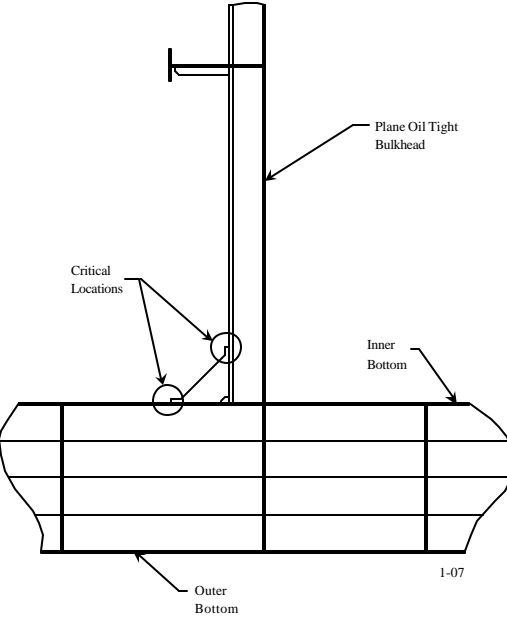
<p>Vessel Type: Double Hull Oil Tankers Critical Area: Double Bottom Critical Detail: Intersection of Inner/Outer Bottom Longitudinal and Transverse Floor Stiffener Detail Description: Symmetric Longitudinal</p>	 Ship Structure Committee  FLEET TECHNOLOGY LTD	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">2</h1> <p>Rev. 001-250199</p>																							
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Heel</p>  <p>Detail Design B - Soft Toe and Soft Heel</p>  <p>Detail Design C - Soft Toe and Soft Backing Bracket</p> 																								
<p>CRITICAL DETAIL:</p>  <p>Section A-A</p> 	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2">Detail Attributes</th> <th colspan="3">Design Detail Rankings</th> </tr> <tr> <th>A</th> <th>B</th> <th>C</th> </tr> </thead> <tbody> <tr> <td>Fatigue Performance</td> <td>3</td> <td>2</td> <td>1</td> </tr> <tr> <td>Ease of Inspection</td> <td>1</td> <td>1</td> <td>2</td> </tr> <tr> <td>Maintenance Cost</td> <td>1</td> <td>2</td> <td>3</td> </tr> <tr> <td>Fabrication Cost</td> <td>1</td> <td>2</td> <td>3</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings			A	B	C	Fatigue Performance	3	2	1	Ease of Inspection	1	1	2	Maintenance Cost	1	2	3	Fabrication Cost	1	2	3
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Fabrication Cost	1	2	3																						
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between the transverse floor stiffener and the backing bracket is to be maintained. - A wrap around weld, free of undercut or notches, around the heel and toe connections of the floor stiffener and backing bracket. 																									



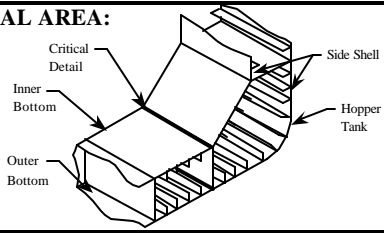
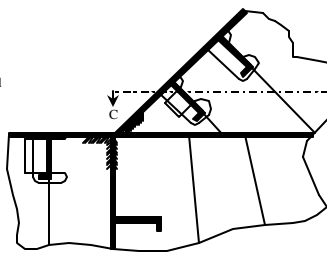
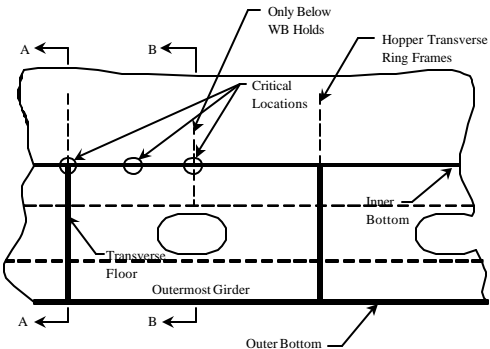
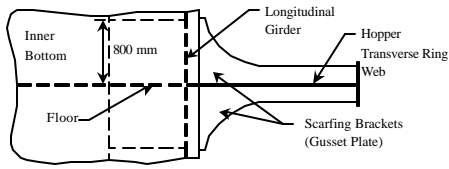
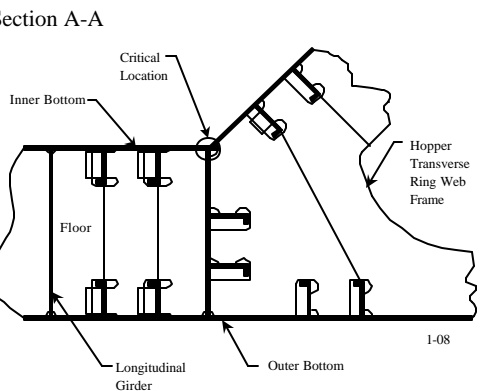
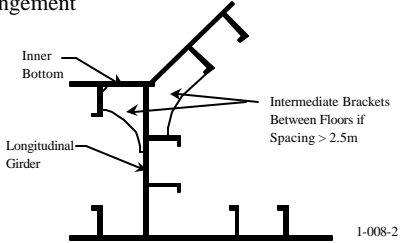
<p>Vessel Type: Double Hull Oil Tankers Critical Area: Double Bottom Critical Detail: Intersection of Inner/Outer Bottom Longitudinal and Transverse Bulkhead Detail Description: Plane Oil Tight Transverse Bulkhead</p>	 Ship Structure Committee  FLEET TECHNOLOGY LTD	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">3</h1> <p style="font-size: 10px;">Rev. 001-250199</p>																	
<p>CRITICAL AREA:</p>  <p>Labels: Transverse Bulkhead Stiffener, Inner Bottom, Bottom Girder, Transverse Bulkhead, Critical Detail, Bottom Longitudinal, Outer Bottom.</p>	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Increased Web Depth of Watertight Floor Stiffener and Soft Toe Brackets on Back Side</p>  <p style="text-align: right;">1-003-1</p> <p>Detail Design B - Increased Web Depth of Watertight Floor Stiffener and Soft Toe Brackets on Both Sides</p>  <p style="text-align: right;">1-003-2</p> <p><small>Note: - Higher tensile steel bracket - Bracket thickness = d/50 or 12 mm Min.</small></p>																		
<p>CRITICAL DETAIL:</p>  <p>Labels: Oil Tight Bulkhead, Inner Bottom, Water Tight Floor, Longitudinal, Outer Bottom.</p> <p>Section A-A</p>  <p>Labels: Flanged Bracket, Critical Locations, Critical Locations.</p> <p style="text-align: right;">1-03</p>	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2" style="background-color: #cccccc;">Detail Attributes</th> <th colspan="2" style="background-color: #cccccc;">Design Detail Rankings</th> </tr> <tr> <th style="background-color: #cccccc;">A</th> <th style="background-color: #cccccc;">B</th> </tr> </thead> <tbody> <tr> <td style="text-align: left;">Fatigue Performance</td> <td>2</td> <td>1</td> </tr> <tr> <td style="text-align: left;">Ease of Inspection</td> <td>1</td> <td>2</td> </tr> <tr> <td style="text-align: left;">Maintenance Cost</td> <td>1</td> <td>2</td> </tr> <tr> <td style="text-align: left;">Fabrication Cost</td> <td>1</td> <td>2</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings		A	B	Fatigue Performance	2	1	Ease of Inspection	1	2	Maintenance Cost	1	2	Fabrication Cost	1	2
Detail Attributes	Design Detail Rankings																		
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Fatigue Performance	2	1																	
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Maintenance Cost	1	2																	
Fabrication Cost	1	2																	
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between the floor stiffener, soft toe bracket and the web of the longitudinal is to be maintained. - A wrap around weld, free of undercut or notches, around the floor stiffener heel and the heel and toe of the bracket connection to the longitudinal. 																			



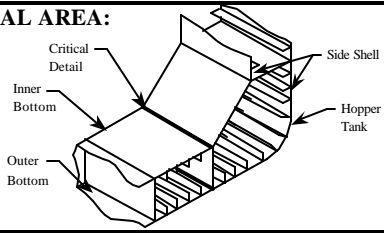
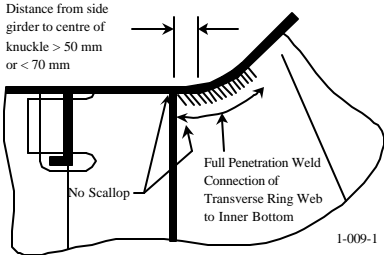
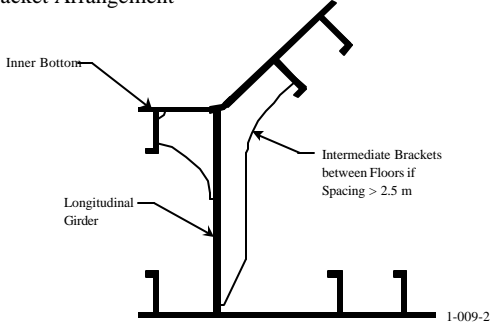
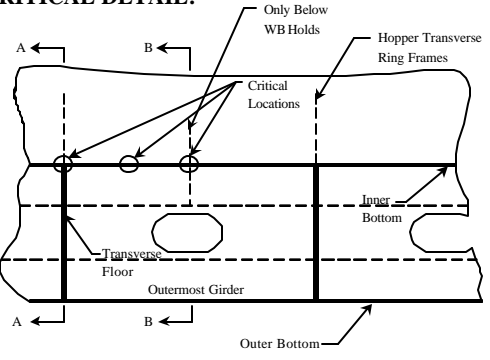
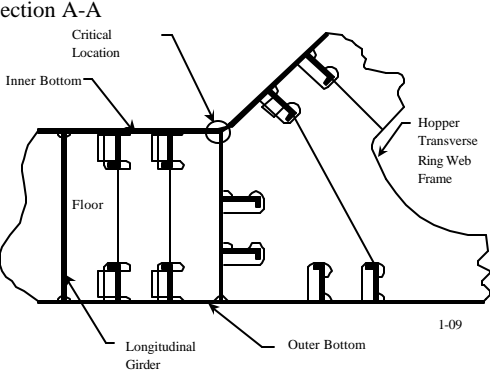
<p>Vessel Type: Double Hull Oil Tankers Critical Area: Double Bottom Critical Detail: Intersection of Inner/Outer Bottom Longitudinal and Transverse Bulkhead Detail Description: Corrugated Oil Tight Transverse Bulkhead Supported by Horizontal Girders</p>	 	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">4</h1> <p style="font-size: 8px;">Rev. 001-250199</p>																	
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Increased Web Depth of Watertight Floor Stiffeners with Transverse Carlings and Soft Toe Brackets</p>  <p style="text-align: center;">Section C-C 1-004-1</p>																		
<p>CRITICAL DETAIL:</p>  <p style="text-align: center;">Section A-A Section B-B</p> <p style="text-align: right; font-size: 8px;">1-04</p>	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2" style="background-color: #cccccc;">Detail Attributes</th> <th colspan="2" style="background-color: #cccccc;">Design Detail Rankings</th> </tr> <tr> <th style="background-color: #cccccc;">A</th> <th style="background-color: #cccccc;">Original Detail</th> </tr> </thead> <tbody> <tr> <td style="text-align: left;">Fatigue Performance</td> <td>1</td> <td>2</td> </tr> <tr> <td style="text-align: left;">Ease of Inspection</td> <td>2</td> <td>1</td> </tr> <tr> <td style="text-align: left;">Maintenance Cost</td> <td>2</td> <td>1</td> </tr> <tr> <td style="text-align: left;">Fabrication Cost</td> <td>2</td> <td>1</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings		A	Original Detail	Fatigue Performance	1	2	Ease of Inspection	2	1	Maintenance Cost	2	1	Fabrication Cost	2	1
Detail Attributes	Design Detail Rankings																		
	A	Original Detail																	
Fatigue Performance	1	2																	
Ease of Inspection	2	1																	
Maintenance Cost	2	1																	
Fabrication Cost	2	1																	
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Due to the complexity of the connection, this detail warrants further detailed analysis. - Alignment between the longitudinal diaphragm, the soft toe backing brackets, the web of the longitudinals, and the flange of the corrugated bulkhead to the transverse carling is to be maintained. - A wrap around weld, free of undercut or notches, around the toe of the bracket connection to the longitudinal. 																			



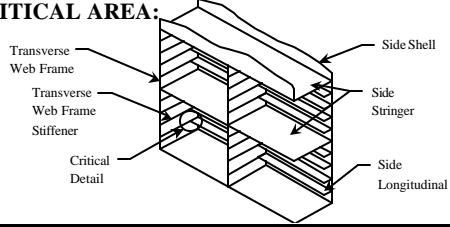
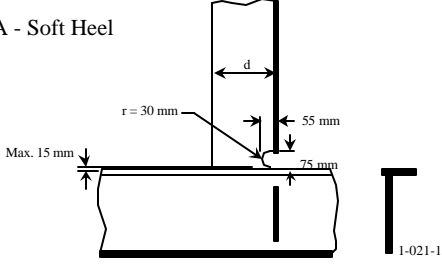
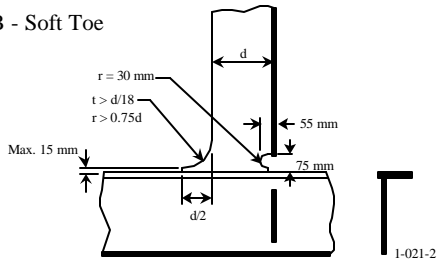
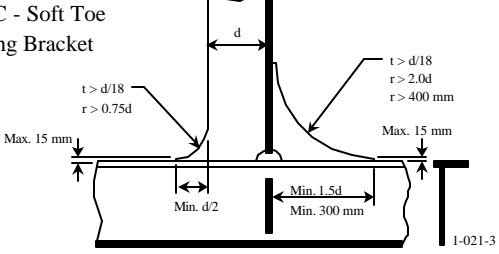
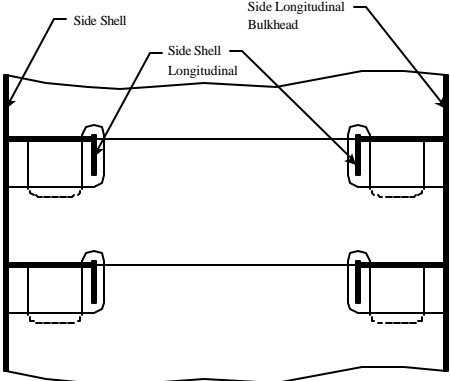
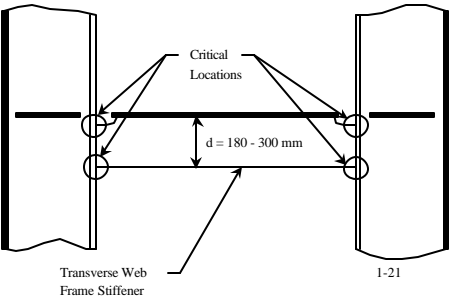
<p>Vessel Type: Double Hull Oil Tankers Critical Area: Double Bottom Critical Detail: Intersection of Bottom Longitudinal and Transverse Ring Web Detail Description: Transverse Ring Web Tripping Bracket</p>	 Ship Structure Committee 	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">5</h1> <p style="font-size: 10px;">Rev. 001-260698</p>																							
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Heel</p>  <p style="text-align: right;">1-005-1</p>																								
<p>CRITICAL DETAIL:</p> 	<p>Detail Design B - Soft Toe and Soft Heel</p>  <p style="text-align: right;">1-005-2</p>																								
<p>Section A-A</p>  <p style="text-align: right;">1-05</p>	<p>Detail Design C - Soft Toe and Soft Backing Bracket</p>  <p style="text-align: right;">1-005-3</p>																								
<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2" style="background-color: #cccccc;">Detail Attributes</th> <th colspan="3" style="background-color: #cccccc;">Design Detail Rankings</th> </tr> <tr> <th style="background-color: #cccccc;">A</th> <th style="background-color: #cccccc;">B</th> <th style="background-color: #cccccc;">C</th> </tr> </thead> <tbody> <tr> <td style="background-color: #cccccc;">Fatigue Performance</td> <td>3</td> <td>2</td> <td>1</td> </tr> <tr> <td style="background-color: #cccccc;">Ease of Inspection</td> <td>1</td> <td>1</td> <td>2</td> </tr> <tr> <td style="background-color: #cccccc;">Maintenance Cost</td> <td>1</td> <td>2</td> <td>3</td> </tr> <tr> <td style="background-color: #cccccc;">Fabrication Cost</td> <td>1</td> <td>2</td> <td>3</td> </tr> </tbody> </table>			Detail Attributes	Design Detail Rankings			A	B	C	Fatigue Performance	3	2	1	Ease of Inspection	1	1	2	Maintenance Cost	1	2	3	Fabrication Cost	1	2	3
Detail Attributes	Design Detail Rankings																								
	A	B	C																						
Fatigue Performance	3	2	1																						
Ease of Inspection	1	1	2																						
Maintenance Cost	1	2	3																						
Fabrication Cost	1	2	3																						
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between the tripping bracket, the backing bracket and the web of the bottom longitudinal is to be maintained. - A wrap around weld, free of undercut or notches, around the heel and toe of the bracket connection to the longitudinal face plate. 																									
<p>Ref: Lloyd's Register</p>																									



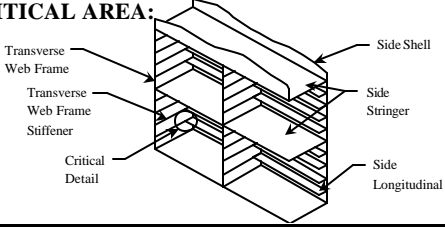
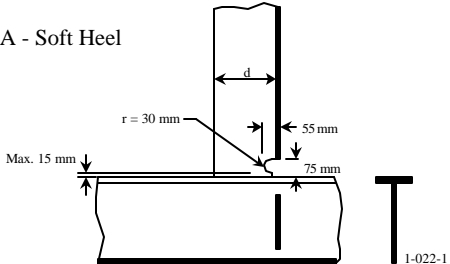
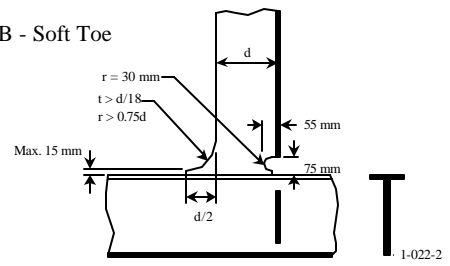
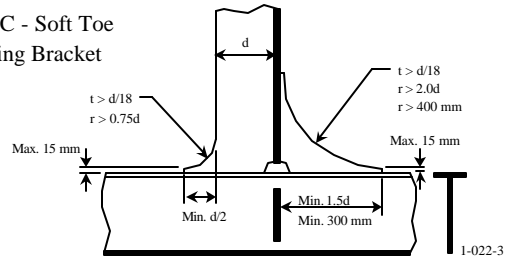
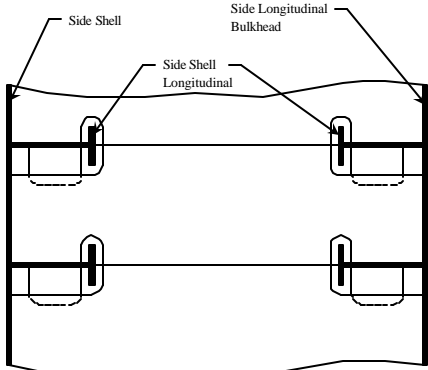
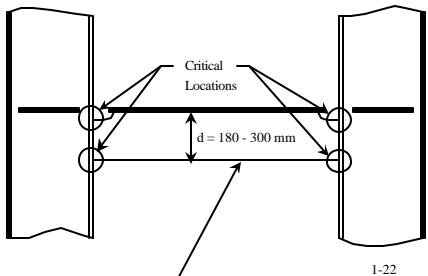
<p>Vessel Type: Double Hull Oil Tankers Critical Area: Double Bottom Critical Detail: Intersection of Bottom Girder and Transverse Bulkhead Detail Description: Plane Oil Tight Transverse Bulkhead</p>	 Ship Structure Committee 	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">6</h1> <p style="font-size: 10px;">Rev. 001-250199</p>																	
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Toe</p>  <p>Detail Design B - Parallel Toe</p> 																		
<p>CRITICAL DETAIL:</p> 	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2" style="background-color: #cccccc;">Detail Attributes</th> <th colspan="2" style="background-color: #cccccc;">Design Detail Rankings</th> </tr> <tr> <th style="background-color: #cccccc;">A</th> <th style="background-color: #cccccc;">B</th> </tr> </thead> <tbody> <tr> <td style="text-align: left;">Fatigue Performance</td> <td>2</td> <td>1</td> </tr> <tr> <td style="text-align: left;">Ease of Inspection</td> <td>2</td> <td>2</td> </tr> <tr> <td style="text-align: left;">Maintenance Cost</td> <td>2</td> <td>2</td> </tr> <tr> <td style="text-align: left;">Fabrication Cost</td> <td>1</td> <td>2</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings		A	B	Fatigue Performance	2	1	Ease of Inspection	2	2	Maintenance Cost	2	2	Fabrication Cost	1	2
Detail Attributes	Design Detail Rankings																		
	A	B																	
Fatigue Performance	2	1																	
Ease of Inspection	2	2																	
Maintenance Cost	2	2																	
Fabrication Cost	1	2																	
<p>COMMENTS:</p> <ul style="list-style-type: none"> - The nominal distance between the centers of the bracket toe thickness and bottom girder web thickness should not exceed 1/3 of the inner bottom thickness. - Extent of full penetration is 2-3 longitudinal spacings, or back to the first bracket web stiffener. A wrap around weld, free of undercut or notches, in way of the toe of the bracket connection to the inner bottom plating. <p>Ref: Lloyd's Register</p>																			



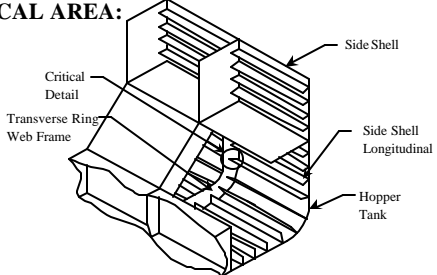
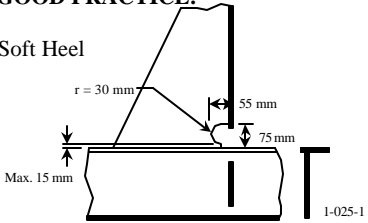
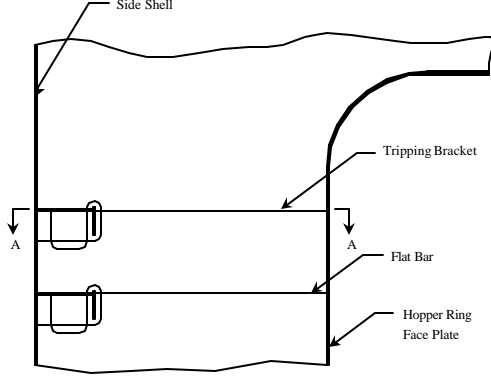
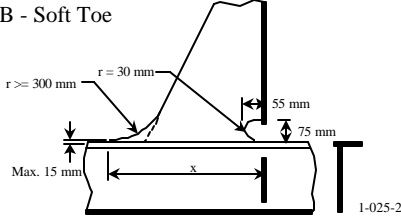
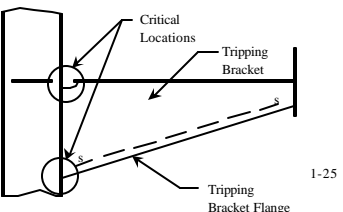
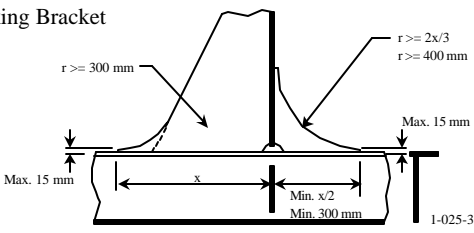
<p>Vessel Type: Double Hull Oil Tankers Critical Area: Double Bottom Critical Detail: Intersection of Bottom Girder and Transverse Plane Oil Tight Bulkhead Detail Description: Bulkhead Vertical Stiffener End Brackets and Higher Tensile Steel Bottom Girders</p>	  Detail 7 <small>Rev. 001-250199</small>																	
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Toe</p>  <p style="text-align: right;">1-007-1</p> <p>Detail Design B - Integral Soft Toe Bracket, Increased Web Thickness and Elimination of Scallop</p>  <p style="text-align: right;">1-007-2</p>																	
<p>CRITICAL DETAIL:</p>  <p style="text-align: right;">1-07</p>	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2">Detail Attributes</th> <th colspan="2">Design Detail Rankings</th> </tr> <tr> <th>A</th> <th>B</th> </tr> </thead> <tbody> <tr> <td>Fatigue Performance</td> <td>2</td> <td>1</td> </tr> <tr> <td>Ease of Inspection</td> <td>1</td> <td>1</td> </tr> <tr> <td>Maintenance Cost</td> <td>1</td> <td>2</td> </tr> <tr> <td>Fabrication Cost</td> <td>1</td> <td>2</td> </tr> </tbody> </table>	Detail Attributes	Design Detail Rankings		A	B	Fatigue Performance	2	1	Ease of Inspection	1	1	Maintenance Cost	1	2	Fabrication Cost	1	2
Detail Attributes	Design Detail Rankings																	
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Fatigue Performance	2	1																
Ease of Inspection	1	1																
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Fabrication Cost	1	2																
<p>COMMENTS:</p> <p>- The nominal distance between the centers of the bracket toe thickness and bottom girder web thickness should not exceed 1/3 of the inner bottom thickness.</p>																		
<p>Ref: Lloyd's Register</p>																		



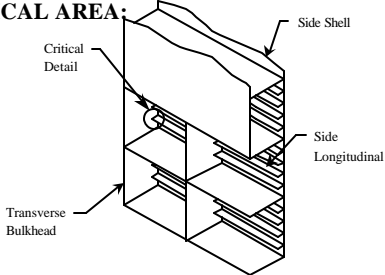
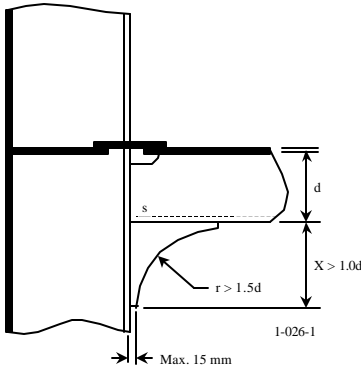
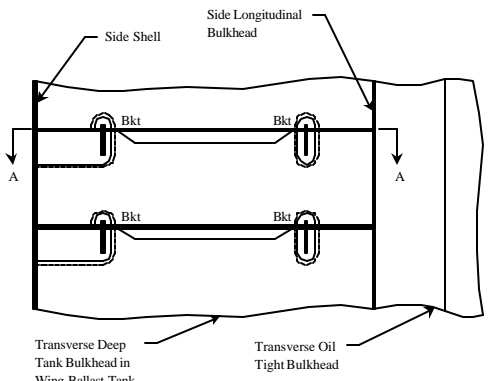
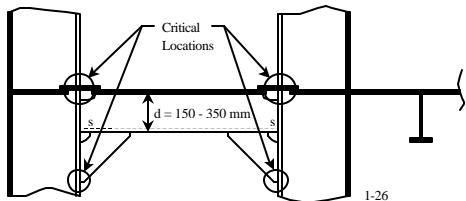
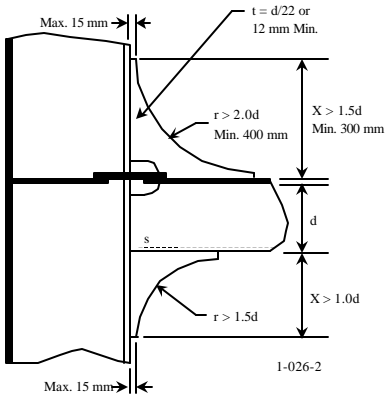
<p>Vessel Type: Double Hull Oil Tankers Critical Area: Double Bottom Critical Detail: Intersection of Inner Bottom Plating, Hopper Tank and Transverse Floor Detail Description: Welded Knuckle Connection</p>	 Ship Structure Committee  FTL FLEET TECHNOLOGY LTD	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">8</h1> <p style="font-size: 8px;">Rev. 001-250199</p>																	
<p>CRITICAL AREA:</p>  <p>Labels: Critical Detail, Inner Bottom, Outer Bottom, Side Shell, Hopper Tank</p>	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Elimination of Scallops and Extension of Inner</p> <p>No scallop. Full penetration weld connection of floor and web to inner bottom girder and hopper</p>  <p style="text-align: right;">1-008-1</p>																		
<p>CRITICAL DETAIL:</p>  <p>Labels: Only Below WB Holds, Hopper Transverse Ring Frames, Critical Locations, Inner Bottom, Transverse Floor, Outermost Girder, Outer Bottom</p>	<p>Section C-C</p>  <p>Labels: Inner Bottom, 1800 mm, Floor, Longitudinal Girder, Hopper Transverse Ring Web, Scarfing Brackets (Gusset Plate)</p>																		
<p>Section A-A</p>  <p>Labels: Critical Location, Inner Bottom, Floor, Longitudinal Girder, Outer Bottom</p> <p style="text-align: right;">1-08</p>	<p>Detail Design B - Detail Design A with Additional Intermediate Bracket Arrangement</p> <p>Section B-B</p>  <p>Labels: Inner Bottom, Longitudinal Girder, Intermediate Brackets Between Floors if Spacing > 2.5m</p> <p style="text-align: right;">1-008-2</p>																		
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Fatigue Performance	2	1																	
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Maintenance Cost	1	2																	
Fabrication Cost	1	2																	
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between the transverse floor and the hopper transverse ring web and between sloping plating and hopper side girder is to be maintained. Maximum misalignment is to be less than $t/3$ where t is the thinner of the webs to be aligned and misalignment is the overhang of the thinner thickness. - Full penetration welding for the connections near the knuckle between double bottom floor to inner bottom and girder, inner bottom to hopper and transverse ring web to hopper and girder and where full collars are fitted. 																			



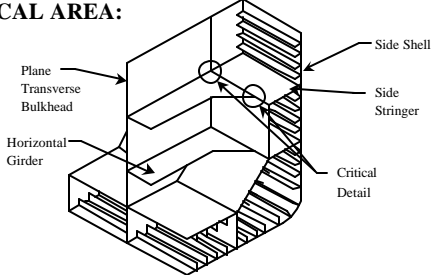
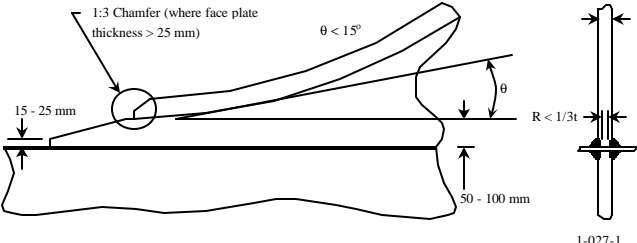
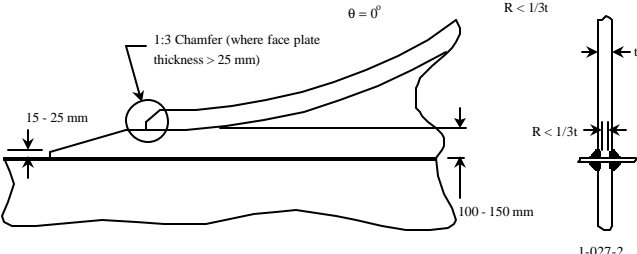
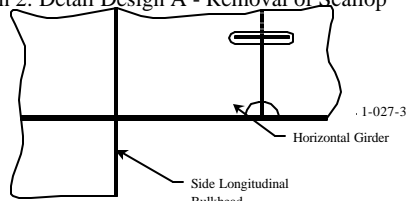
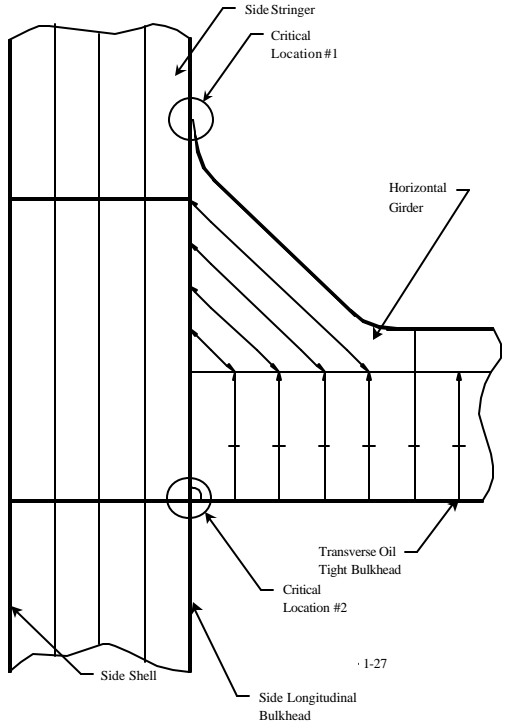
<p>Vessel Type: Double Hull Oil Tankers</p> <p>Critical Area: Double Bottom</p> <p>Critical Detail: Intersection of Inner Bottom Plating, Hopper Tank and Transverse Floor</p> <p>Detail Description: Radiused Knuckle Connection</p>	 <p>Ship Structure Committee</p>  <p>FLEET TECHNOLOGY LTD</p>	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">9</h1> <p style="font-size: 8px;">Rev. 001-250199</p>																	
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Elimination of Scallops</p>  <p>Detail Design B - Detail Design A with Additional Intermediate Bracket Arrangement</p> 																		
<p>CRITICAL DETAIL:</p>  <p>Section A-A</p> 	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2" style="background-color: #cccccc;">Detail Attributes</th> <th colspan="2" style="background-color: #cccccc;">Design Detail Rankings</th> </tr> <tr> <th style="background-color: #cccccc;">A</th> <th style="background-color: #cccccc;">B</th> </tr> </thead> <tbody> <tr> <td style="text-align: left;">Fatigue Performance</td> <td>2</td> <td>1</td> </tr> <tr> <td style="text-align: left;">Ease of Inspection</td> <td>1</td> <td>2</td> </tr> <tr> <td style="text-align: left;">Maintenance Cost</td> <td>1</td> <td>2</td> </tr> <tr> <td style="text-align: left;">Fabrication Cost</td> <td>1</td> <td>2</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings		A	B	Fatigue Performance	2	1	Ease of Inspection	1	2	Maintenance Cost	1	2	Fabrication Cost	1	2
Detail Attributes	Design Detail Rankings																		
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Fatigue Performance	2	1																	
Ease of Inspection	1	2																	
Maintenance Cost	1	2																	
Fabrication Cost	1	2																	
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between double bottom floor and the hopper transverse ring web. Maximum misalignment is to be less than $t/3$ where t is the thinner of the webs to be aligned and misalignment is the overhang of the thinner thickness. - Full penetration welding for the connections near the knuckle between inner bottom and transverse ring web. 																			



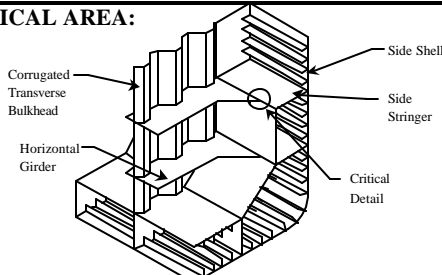
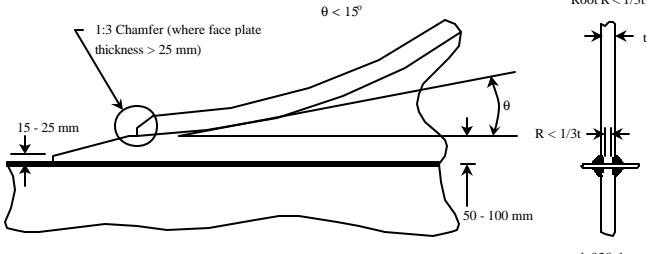
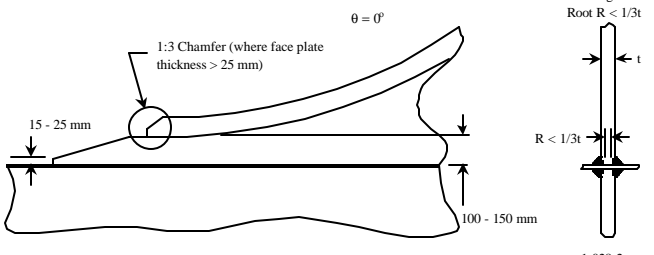
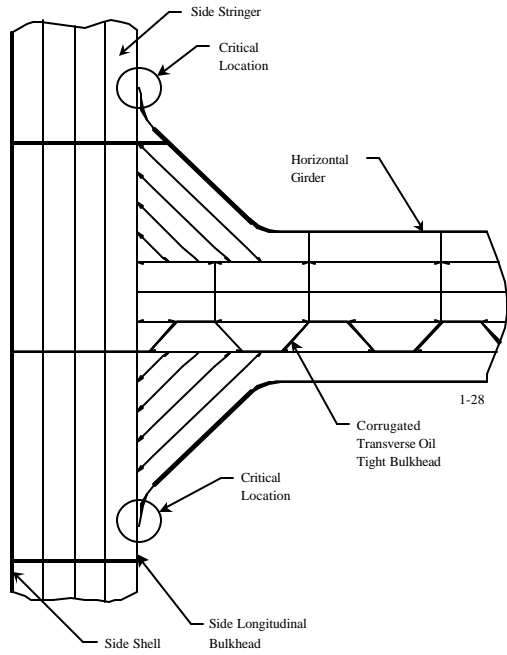
<p>Vessel Type: Double Hull Oil Tankers</p> <p>Critical Area: Double Side</p> <p>Critical Detail: Intersection of Side Shell Longitudinal and Transverse Web Frame Stiffener</p> <p>Detail Description: Asymmetric Longitudinal</p>	 Ship Structure Committee 	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">21</h1> <p style="font-size: 10px;">Rev. 001-250199</p>																							
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Heel</p>  <p>Detail Design B - Soft Toe and Soft Heel</p>  <p>Detail Design C - Soft Toe and Soft Backing Bracket</p> 																								
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<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between the transverse web frame stiffener, the backing bracket and the web of the longitudinal is to be maintained. - A wrap around weld, free of undercut or notches, around the heel and toe connections of the stiffeners and the backing bracket to the longitudinal face plate. 																									



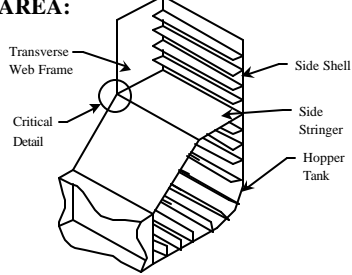
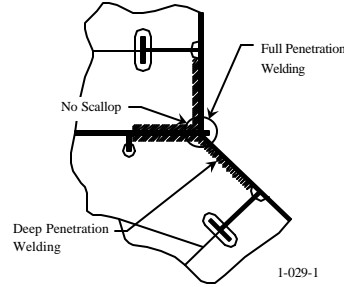
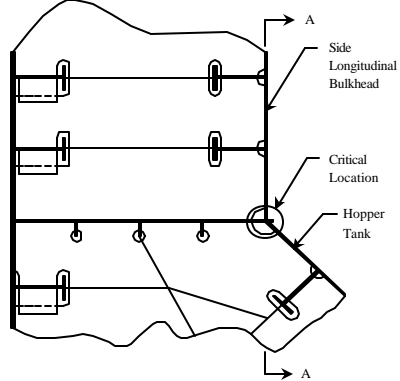
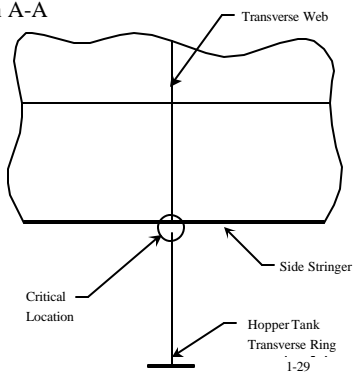
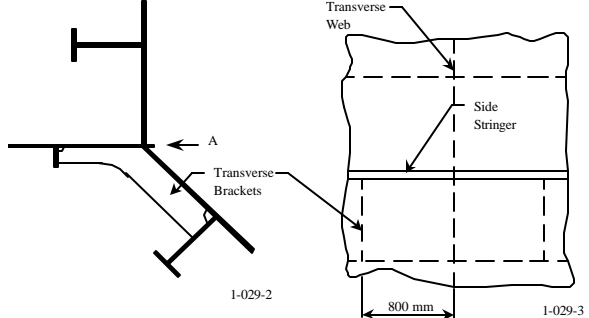
<p>Vessel Type: Double Hull Oil Tankers Critical Area: Double Side Critical Detail: Intersection of Side Shell Longitudinal and Transverse Web Frame Stiffener Detail Description: Symmetric Longitudinal</p>	 Ship Structure Committee 	<p>Detail</p> <h1 style="font-size: 2em; margin: 0;">22</h1> <p style="font-size: 0.8em;">Rev. 001-250199</p>																							
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Heel</p>  <p>Detail Design B - Soft Toe and Soft Heel</p>  <p>Detail Design C - Soft Toe and Soft Backing Bracket</p> 																								
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<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between the transverse web frame stiffener, the backing bracket and the web of the longitudinal is to be maintained. - A wrap around weld, free of undercut or notches, around the heel and toe connections of the stiffeners and the backing bracket to the longitudinal face plate. 																									



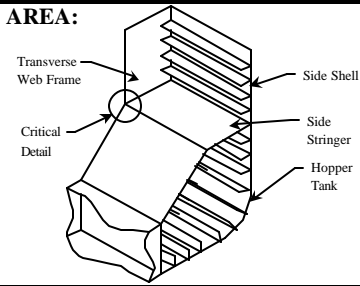
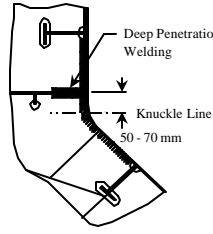
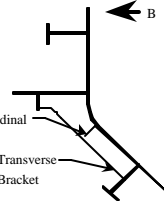
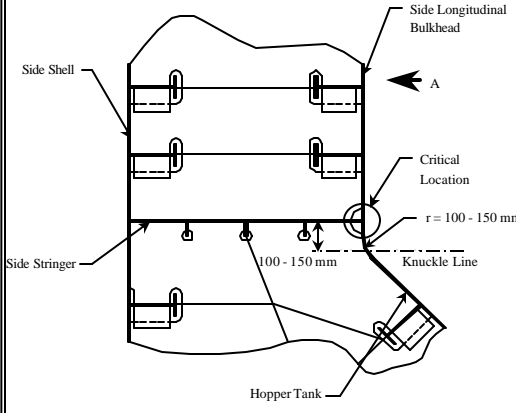
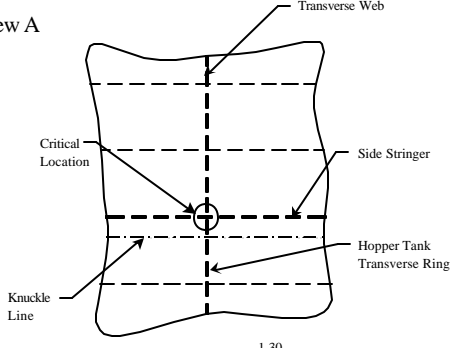
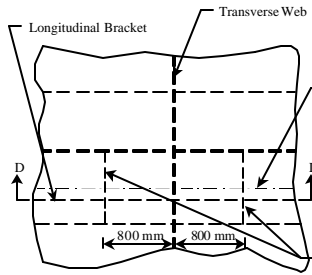
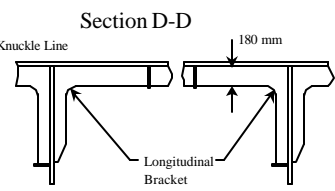
<p>Vessel Type: Double Hull Oil Tankers Critical Area: Double Side Critical Detail: Intersection of Side Shell Longitudinal and Transverse Ring Web Detail Description: Transverse Ring Web Tripping Bracket</p>	 Ship Structure Committee 	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">25</h1> <p>Rev. 001-250199</p>																							
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Heel</p>  <p style="text-align: right;">1-025-1</p>																								
<p>CRITICAL DETAIL:</p> 	<p>Detail Design B - Soft Toe and Soft Heel</p>  <p style="text-align: right;">1-025-2</p>																								
 <p style="text-align: center;">Section A-A</p> <p style="text-align: right;">1-25</p>	<p>Detail Design C - Soft Toe and Soft Backing Bracket</p>  <p style="text-align: right;">1-025-3</p>																								
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<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between the tripping bracket, the backing bracket and the web of the side longitudinal is to be maintained. - A wrap around weld, free of undercut or notches, around the heel and toe of the bracket connection to the longitudinal face plate. 																									
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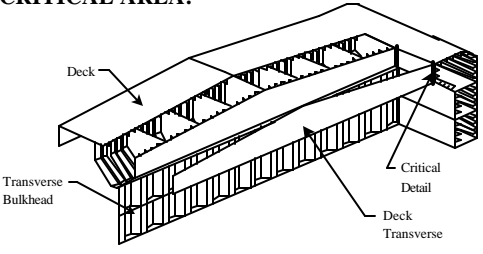
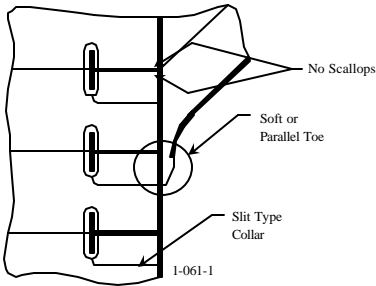
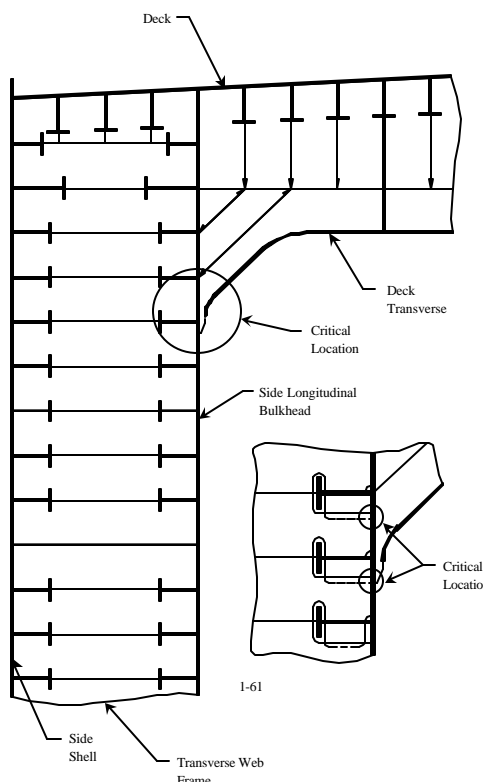
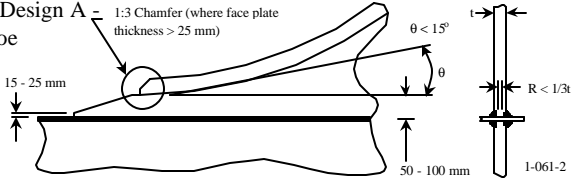
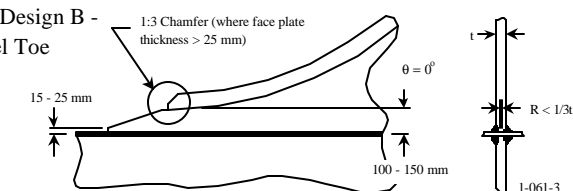
<p>Vessel Type: Double Hull Oil Tankers Critical Area: Double Side Critical Detail: Intersection of Side Shell Longitudinal and Transverse Bulkhead Detail Description: Transverse Bulkhead Horizontal Stiffener</p>	 Ship Structure Committee 	<p>Detail</p> <h1 style="font-size: 2em; margin: 0;">26</h1> <p style="font-size: 0.8em;">Rev. 001-020299</p>																	
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Toe</p>  <p style="text-align: right;">1-026-1</p>																		
<p>CRITICAL DETAIL:</p>  <p style="text-align: center;">Section A-A</p> 	<p>Detail Design B - Soft Toe and Soft Backing Bracket</p>  <p style="text-align: right;">1-026-2</p>																		
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Maintenance Cost	1	2																	
Fabrication Cost	1	2																	
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between the web of the bulkhead horizontal stiffener, the soft toe brackets and the web of the side longitudinal is to be maintained. - A wrap around weld, free of undercut or notches, around the heel connections of bulkhead stiffener and toe connection of brackets. <p>Ref. Lloyd's Register</p>																			

<p>Vessel Type: Double Hull Oil Tankers Critical Area: Double Side Critical Detail: Intersection of Side stringer and Transverse Bulkhead Horizontal Girder Detail Description: Plane Oil Tight Transverse Bulkhead</p>	 Ship Structure Committee 	<p>Detail</p> <h1 style="font-size: 2em; margin: 0;">27</h1> <p>Rev. 001-250199</p>																	
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Critical Location 1: Detail Design A - Soft Toe</p>  <p>Critical Location 1: Detail Design B - Parallel Toe</p>  <p>Critical Location 2: Detail Design A - Removal of Scallop</p> 																		
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Ease of Inspection	1	2																	
Maintenance Cost	1	2																	
Fabrication Cost	1	2																	
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Nominal distance between the centers of thickness of two abutting members should not exceed 1/3 of table member thickness. - Deep penetration welding for connection of toe brackets in Location 1. Fillet welding for side stringer webs in way of bracket toes in Location 1 and all connection in Location 2. - A wrap around weld, free of undercut or notches, around the plate thickness. <p>Ref: Lloyd's Register</p>																			

<p>Vessel Type: Double Hull Oil Tankers Critical Area: Double Side Critical Detail: Intersection of Side stringer and Transverse Bulkhead Horizontal Girder Detail Description: Corrugated Oil Tight Transverse Bulkhead</p>	 Ship Structure Committee 	<p>Detail</p> <h1 style="font-size: 2em; margin: 0;">28</h1> <p>Rev. 001-250199</p>																	
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Toe</p>  <p style="text-align: right;">1-028-1</p> <p>Detail Design B - Parallel Toe</p>  <p style="text-align: right;">1-028-2</p>																		
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Maintenance Cost	1	2																	
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<p>COMMENTS:</p> <ul style="list-style-type: none"> - Nominal distance between the centers of bracket toe thickness and side stringer web thickness should not exceed 1/3 of side longitudinal bulkhead thickness. - Deep penetration welding for connection of toe brackets to side longitudinal bulkhead. Fillet welding for side stringer webs in way of bracket toes to side longitudinal bulkhead. A wrap around weld, free of undercut or notches, around the plate thickness. <p>Ref: Lloyd's Register</p>																			

<p>Vessel Type: Double Hull Oil Tankers Critical Area: Double Side Critical Detail: Intersection of Transverse Web Frame and Hopper/Topside Tank Detail Description: Welded Side Longitudinal Bulkhead and Hopper Sloping Plating</p>	 Ship Structure Committee  FLEET TECHNOLOGY LTD	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">29</h1> <p style="font-size: 10px;">Rev. 001-020299</p>																	
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Elimination of Scallops</p>  <p style="text-align: right;">1-029-1</p>																		
<p>CRITICAL DETAIL:</p>  <p style="text-align: center;">Section A-A</p>  <p style="text-align: right;">1-29</p>	<p>Detail Design B - Detail Design A with Additional Intermediate Transverse Brackets</p>  <p style="text-align: right;">1-029-2</p> <p style="text-align: right;">View A</p> <p style="text-align: right;">1-029-3</p>																		
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<p>COMMENTS:</p> <ul style="list-style-type: none"> - Nominal distance between centers of thickness of the two abutting members should not exceed 1/3 of the table member thickness. - Full penetration welding for the connection of the hopper sloping plating to side longitudinal bulkhead and connection of side stringer to side longitudinal bulkhead. Deep penetration welding for connection of transverse web to side longitudinal bulkhead and to side stringer, hopper transverse webs to sloping plating, to side longitudinal bulkhead and to side stringers in way of hopper corners. <p>Ref: Lloyd's Register</p>																			

<p>Vessel Type: Double Hull Oil Tankers Critical Area: Double Side Critical Detail: Intersection of Transverse Web Frame and Hopper/Topside Tank Detail Description: Knuckled Side Longitudinal Bulkhead and Hopper Sloping Plating</p>	 Ship Structure Committee  FLEET TECHNOLOGY LTD	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">30</h1> <p style="font-size: 10px;">Rev. 001-250199</p>																	
<p>CRITICAL AREA:</p>  <p>Transverse Web Frame Side Shell Side Stringer Hopper Tank Critical Detail</p>	<p>DETAIL DESIGN GOOD PRACTICE:</p> <div style="display: flex; justify-content: space-around;"> <div style="width: 45%;"> <p>Detail Design A - Removal of Scallops and Closer Knuckle Distance from Side Girder</p>  <p>Deep Penetration Welding Knuckle Line 50 - 70 mm</p> </div> <div style="width: 45%;"> <p>Detail Design B - Detail Design A with Additional Longitudinal Transverse Brackets</p>  <p>Longitudinal Bracket Transverse Bracket</p> </div> </div>																		
<p>CRITICAL DETAIL:</p>  <p>Side Longitudinal Bulkhead Side Shell Side Stringer Hopper Tank Critical Location $r = 100 - 150 \text{ mm}$ Knuckle Line 100 - 150 mm</p> <p>View A</p>  <p>Transverse Web Side Stringer Hopper Tank Transverse Ring Knuckle Line Critical Location</p> <p style="text-align: center;">1-30</p>	 <p>Longitudinal Bracket Transverse Web Knuckle Line Transverse Brackets 800 mm 800 mm</p> <p>View B</p>  <p>Section D-D Knuckle Line Longitudinal Bracket 180 mm</p> <p style="text-align: right;">1-030-1</p>																		
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<p>COMMENTS:</p> <ul style="list-style-type: none"> - Nominal distance between centers of thickness of the two abutting members should not exceed 1/3 of the table member thickness. - Deep penetration welding for the connection of side stringers to side longitudinal bulkhead, connection of transverse web to side longitudinal bulkhead and to side stringer, connection of hopper transverse web to sloped side longitudinal bulkhead and to side stringer in way of hopper corners. <p>Ref: Lloyd's Register</p>																			

<p>Vessel Type: Double Hull Oil Tankers Critical Area: Deck Critical Detail: Intersection of Deck Transverse End Brackets and Transverse Web Frames Detail Description: Toe Connection</p>	 Ship Structure Committee 	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">61</h1> <p style="font-size: 8px;">Rev. 001-250199</p>																	
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design - Modification from Lapped Collars to Slit Type Collars, Removal of Scallops and Improved Bracket Toe</p> 																		
<p>CRITICAL DETAIL:</p> 	<p>Detail Design A - Soft Toe</p>  <p>Detail Design B - Parallel Toe</p> 																		
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Nominal distance between centers of bracket toe thickness and transverse web thickness should not exceed 1/3 of the side longitudinal bulkhead thickness. - Deep penetration welding for the connection of bracket toes to side longitudinal bulkhead. Fillet weldin for the connection of transverse webs in way of bracket toes to side longitudinal bulkhead. 																			
<p>Ref: Lloyd's Register</p>																			
<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2" style="background-color: #cccccc;">Detail Attributes</th> <th colspan="2" style="background-color: #cccccc;">Design Detail Rankings</th> </tr> <tr> <th style="background-color: #cccccc;">A</th> <th style="background-color: #cccccc;">B</th> </tr> </thead> <tbody> <tr> <td style="text-align: left;">Fatigue Performance</td> <td>2</td> <td>1</td> </tr> <tr> <td style="text-align: left;">Ease of Inspection</td> <td>1</td> <td>2</td> </tr> <tr> <td style="text-align: left;">Maintenance Cost</td> <td>1</td> <td>2</td> </tr> <tr> <td style="text-align: left;">Fabrication Cost</td> <td>1</td> <td>2</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings		A	B	Fatigue Performance	2	1	Ease of Inspection	1	2	Maintenance Cost	1	2	Fabrication Cost	1	2	
Detail Attributes	Design Detail Rankings																		
	A	B																	
Fatigue Performance	2	1																	
Ease of Inspection	1	2																	
Maintenance Cost	1	2																	
Fabrication Cost	1	2																	

B.5.2 Bulk Carrier Structure Details

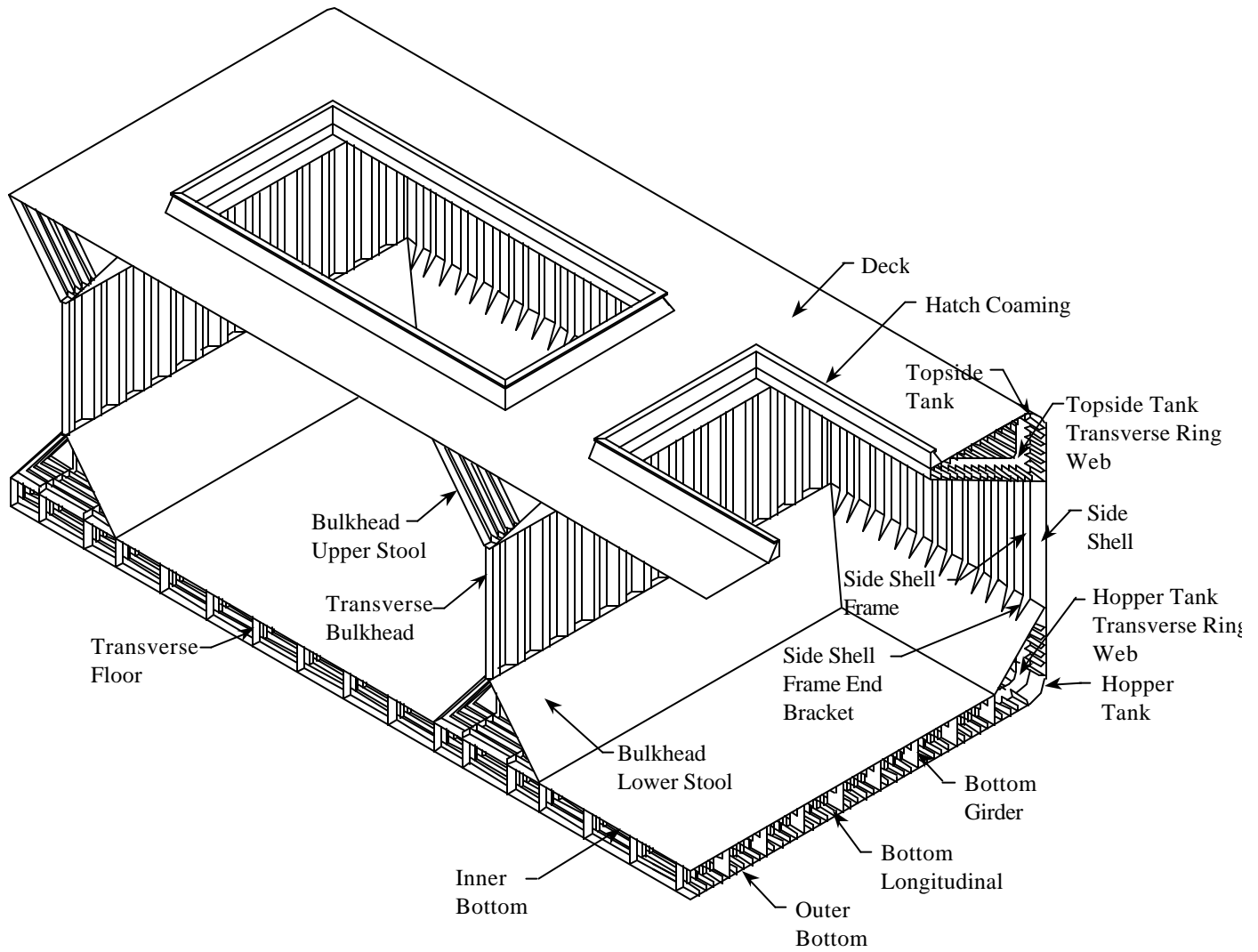


Figure B.5.2: Typical Midship Section Nomenclature for Bulk Carriers

Table B.5.2 (a): Critical Structural Details for Double Bottom

Critical Structural Details		Detail Type#	Detail Figure
Inner/Outer Bottom Longitudinals	Transverse Floor Stiffener	1	
	Transverse Bulkhead	Section B.7	
Bottom Girders	Transverse Floor	Section B.7	
	Transverse Bulkhead	Section B.7	
Inner/Outer Bottom Plating	Hopper Tank - Welded Knuckle Connection - Radiused Knuckle Connection	8 9	
	Bottom Longitudinal	Section B.7	
	Bottom Girder	Section B.7	
	Transverse Floor	Section B.7	
	Transverse Bulkhead	Section B.7	

Table B.5.2 (b): Critical Structural Details for Side Shell (includes hopper and topside tanks)



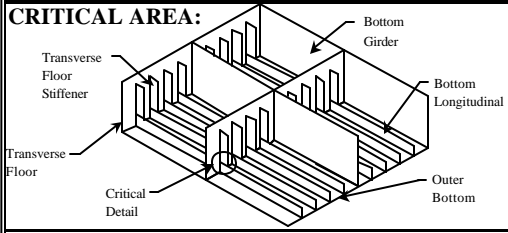
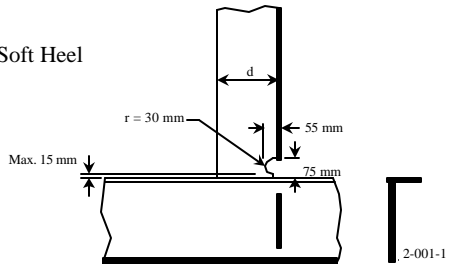
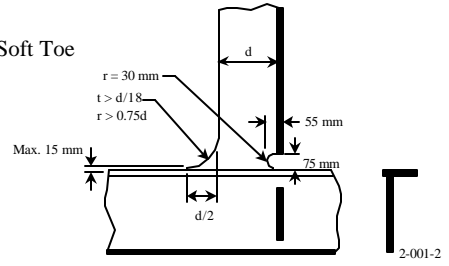
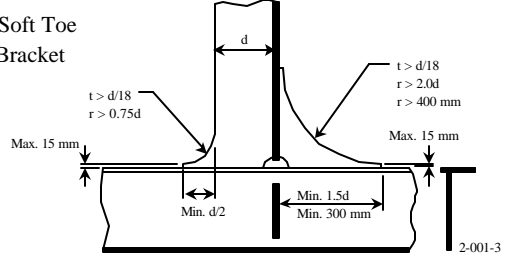
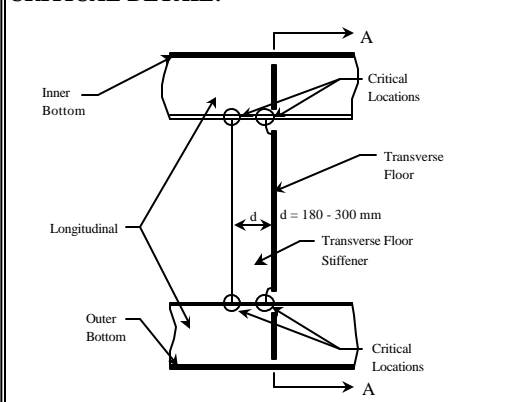
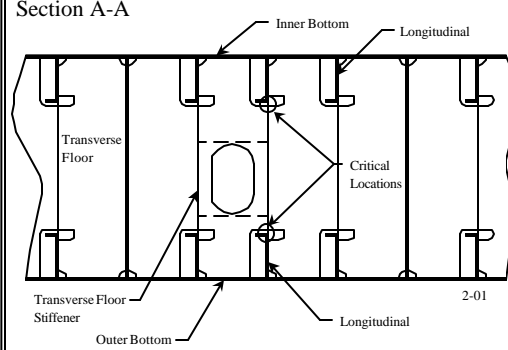
Critical Structural Details		Detail #	Detail Figure
Hopper/Topside Tank Sloping Plating	Side Shell Frame End Brackets	21	
	Side Shell Frames	22	
Hopper/Topside Tank Transverse Ring Web Flat Bar Stiffener	Hopper/Topside Tank Longitudinals	23	
	Bottom/Deck Longitudinals		
	Side Shell Longitudinals		
Hopper/Topside Tank Transverse Ring Web Tripping Bracket	Hopper/Topside Tank Longitudinals	24	
	Bottom/Deck Longitudinals		
	Side Shell Longitudinals		
Side Shell Frame	End Brackets	Section B.7	
	Side Shell Plating	Section B.7	



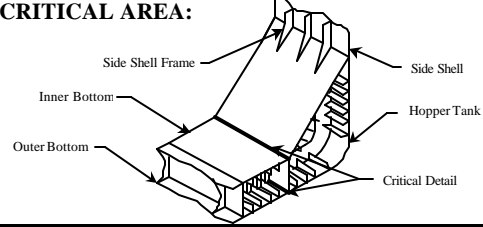
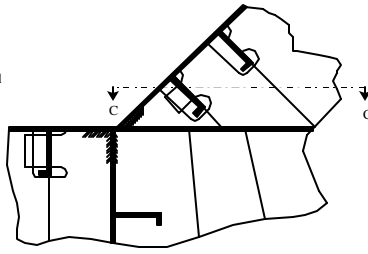
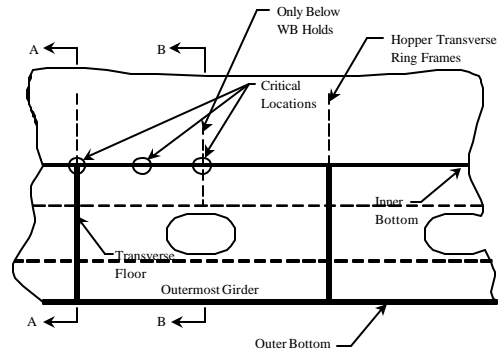
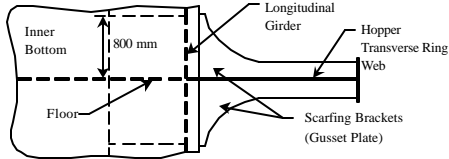
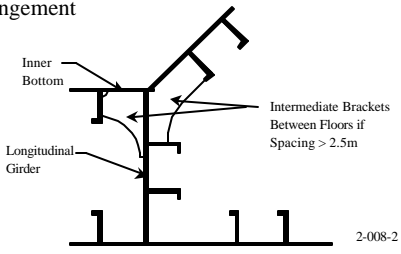
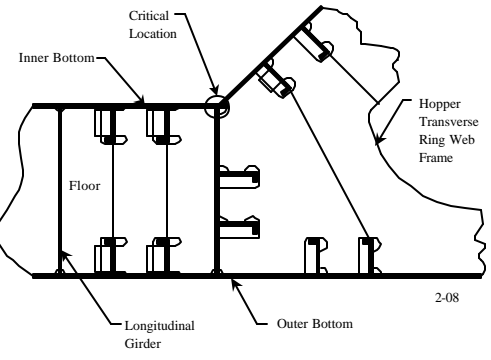
Table B.5.2 (c): Critical Structural Details for Transverse Bulkhead



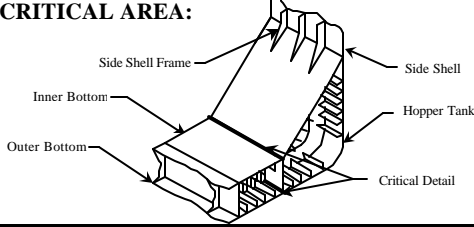
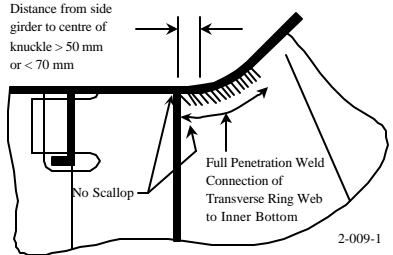
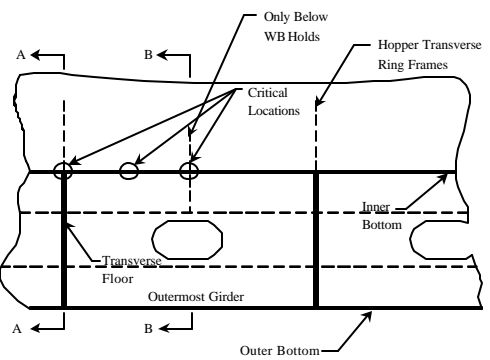
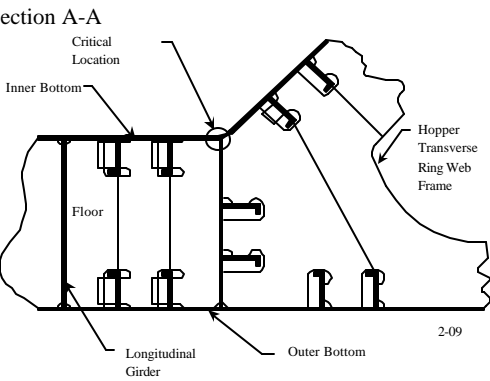
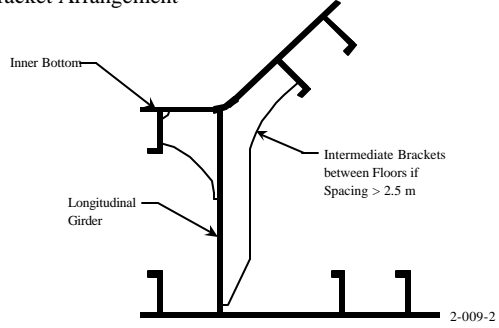
Critical Structural Details		Detail #	Detail Figure
Lower Bulkhead Stool	Inner/Outer Bottom Longitudinals	41	
	Inner Bottom	42	
	Transverse Bulkhead Shelf Plate	43	
Transverse Bulkhead	Topside Tank and Upper Stool	44	



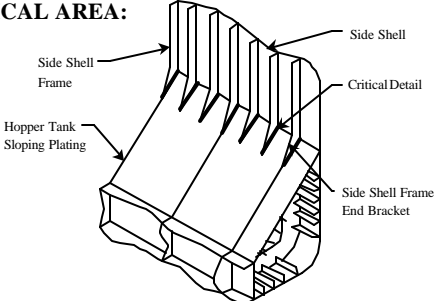
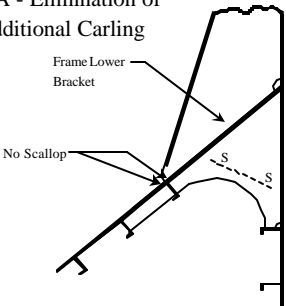
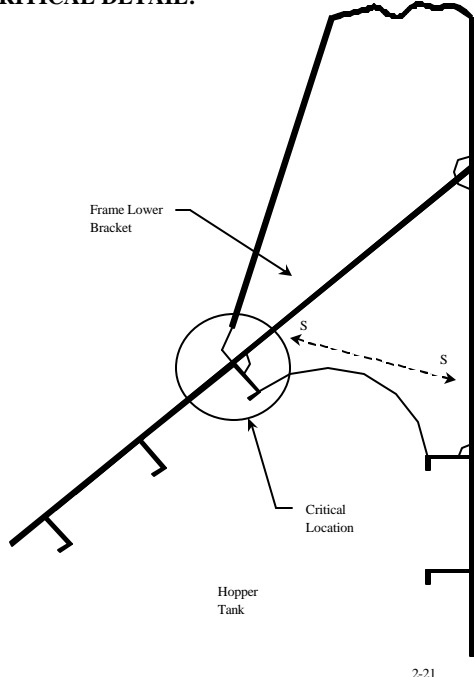
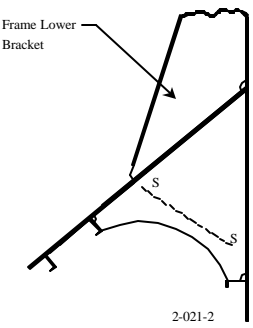
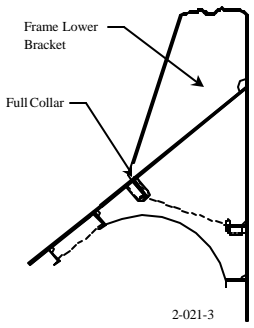
Table B.5.2 (d): Critical Structural Details for Deck



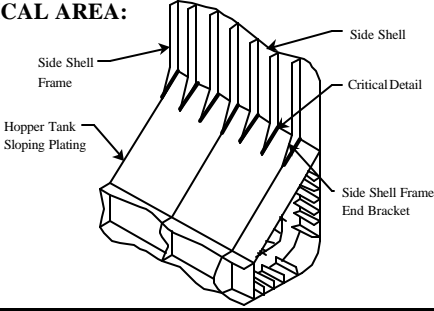
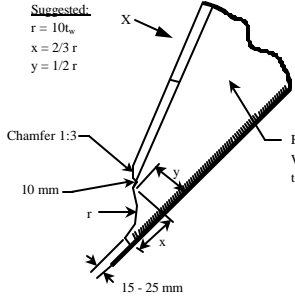
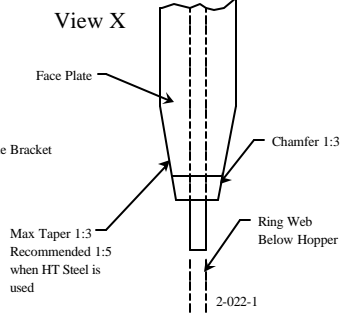
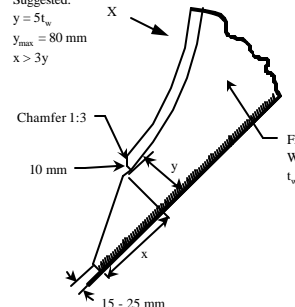
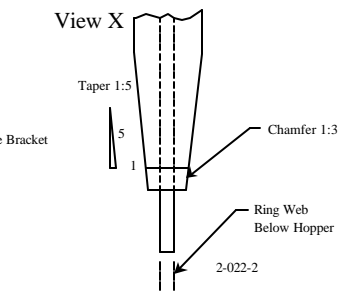
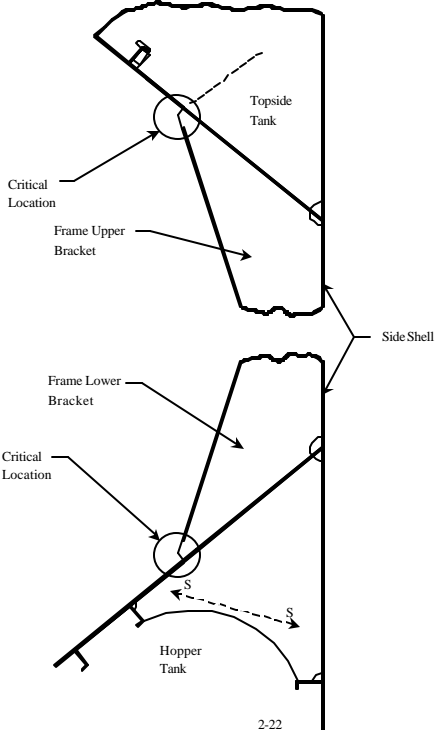
Critical Structural Details		Detail #	Detail Figure
Deck Plating	Topside Tank	Section B.7	
	Transverse Web Frame	Section B.7	
	Longitudinal Hatch Coaming End Bracket	61	
Deck Transverse Stiffener	Deck Longitudinals	62	
Deck Hatch Corners/End Beams	Topside Tank Transverse Ring Web	63	



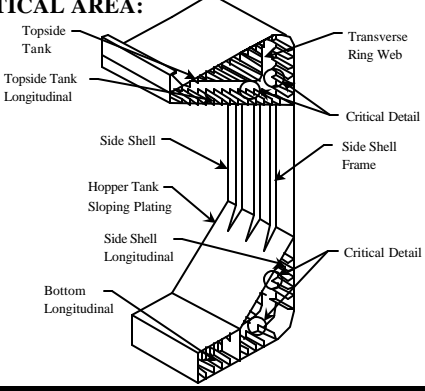
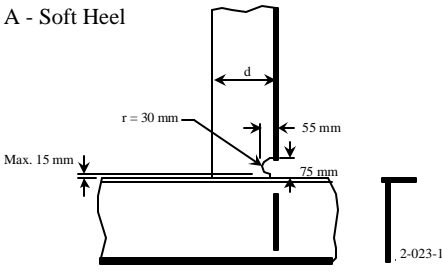
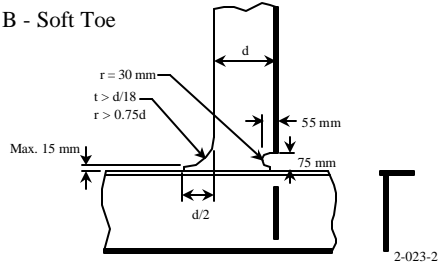
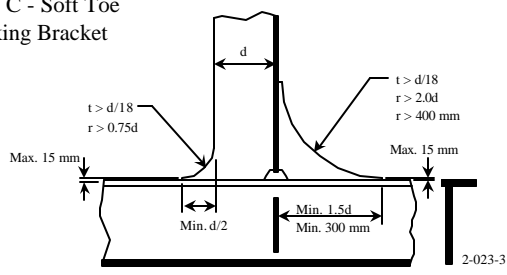
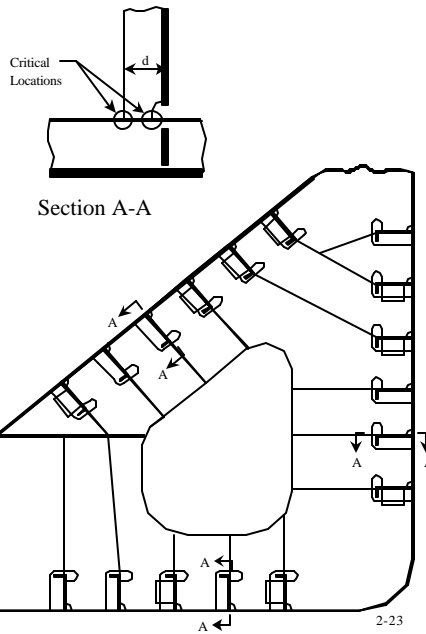
<p>Vessel Type: Bulk Carriers Critical Area: Double Bottom Critical Detail: Intersection of Inner/Outer Bottom Longitudinal and Transverse Floor Stiffener Detail Description:</p>	 Ship Structure Committee  FLEET TECHNOLOGY LTD	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">1</h1> <p>Rev. 001-260199</p>																							
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Heel</p>  <p>Detail Design B - Soft Toe and Soft Heel</p>  <p>Detail Design C - Soft Toe and Soft Backing Bracket</p> 																								
<p>CRITICAL DETAIL:</p>  <p>Section A-A</p> 	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2">Detail Attributes</th> <th colspan="3">Design Detail Rankings</th> </tr> <tr> <th>A</th> <th>B</th> <th>C</th> </tr> </thead> <tbody> <tr> <td>Fatigue Performance</td> <td>3</td> <td>2</td> <td>1</td> </tr> <tr> <td>Ease of Inspection</td> <td>1</td> <td>1</td> <td>2</td> </tr> <tr> <td>Maintenance Cost</td> <td>1</td> <td>2</td> <td>3</td> </tr> <tr> <td>Fabrication Cost</td> <td>1</td> <td>2</td> <td>3</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings			A	B	C	Fatigue Performance	3	2	1	Ease of Inspection	1	1	2	Maintenance Cost	1	2	3	Fabrication Cost	1	2	3
Detail Attributes	Design Detail Rankings																								
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Fatigue Performance	3	2	1																						
Ease of Inspection	1	1	2																						
Maintenance Cost	1	2	3																						
Fabrication Cost	1	2	3																						
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between the longitudinal stiffener web and transverse floor stiffener and the backing bracket is to be maintained. - A wrap around weld, free of undercut or notches, around the heel and toe connections of the floor stiffener and backing bracket. <p>Ref: Lloyd's Register</p>																									



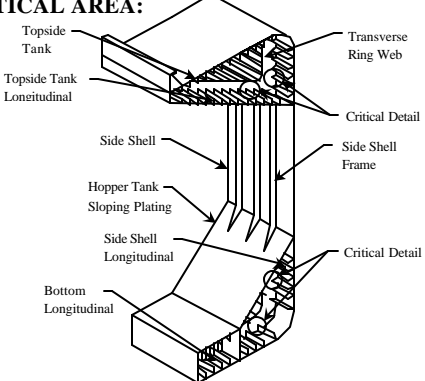
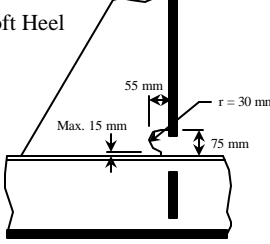
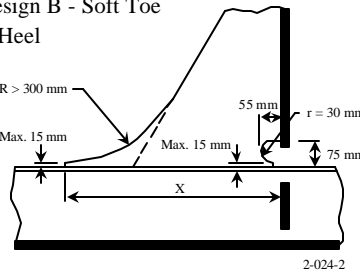
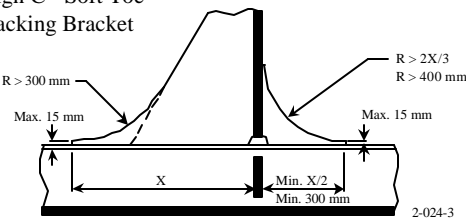
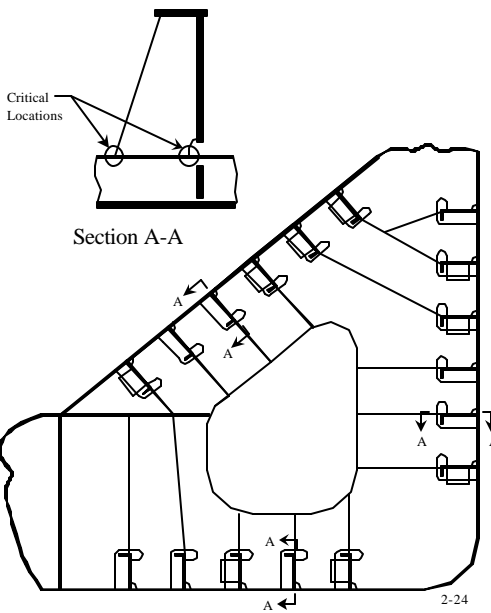
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<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Elimination of Scallops and Extension of Inner</p> <p>No scallop. Full penetration weld connection of floor and web to inner bottom girder and hopper</p>  <p style="text-align: right;">2-008-1</p>																	
<p>CRITICAL DETAIL:</p>  <p style="text-align: right;">2-08</p>	<p>Section C-C</p>  <p style="text-align: right;">2-008-1</p> <p>Detail Design B - Detail Design A with Additional Intermediate Bracket Arrangement</p> <p>Section B-B</p>  <p style="text-align: right;">2-008-2</p>																	
<p>Section A-A</p>  <p style="text-align: right;">2-08</p>	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2">Detail Attributes</th> <th colspan="2">Design Detail Rankings</th> </tr> <tr> <th>A</th> <th>B</th> </tr> </thead> <tbody> <tr> <td>Fatigue Performance</td> <td>2</td> <td>1</td> </tr> <tr> <td>Ease of Inspection</td> <td>1</td> <td>2</td> </tr> <tr> <td>Maintenance Cost</td> <td>1</td> <td>2</td> </tr> <tr> <td>Fabrication Cost</td> <td>1</td> <td>2</td> </tr> </tbody> </table>	Detail Attributes	Design Detail Rankings		A	B	Fatigue Performance	2	1	Ease of Inspection	1	2	Maintenance Cost	1	2	Fabrication Cost	1	2
Detail Attributes	Design Detail Rankings																	
	A	B																
Fatigue Performance	2	1																
Ease of Inspection	1	2																
Maintenance Cost	1	2																
Fabrication Cost	1	2																
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between the transverse floor and the hopper transverse ring web and between sloping plating and hopper side girder is to be maintained. Maximum misalignment is to be less than $t/3$ where t is the thinner of the webs to be aligned and misalignment is the overhang of the thinner thickness. - Full penetration welding for the connections near the knuckle between double bottom floor to inner bottom and girder, inner bottom to hopper and transverse ring web to hopper and girder and where full collars are fitted. 																		

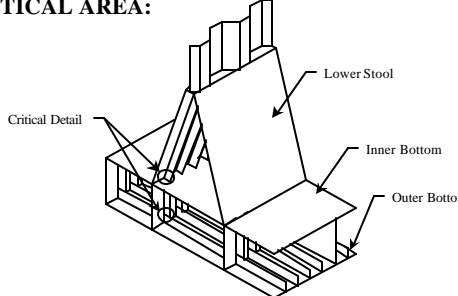
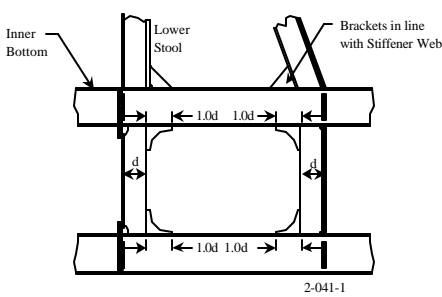
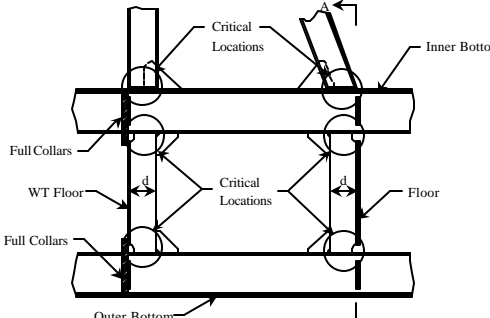
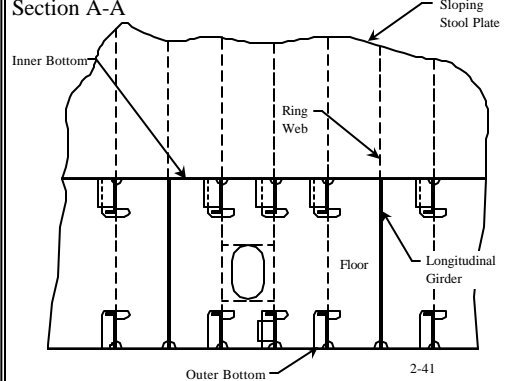
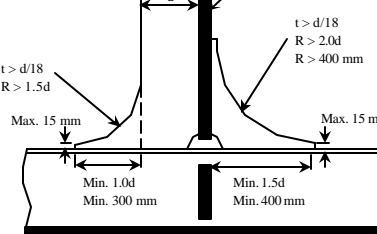
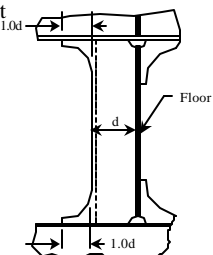
<p>Vessel Type: Bulk Carriers Critical Area: Double Bottom Critical Detail: Intersection of Inner/Outer Bottom Plating and Hopper Tank Detail Description: Radiused Knuckle Connection</p>	 Ship Structure Committee  FLEET TECHNOLOGY LTD	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">9</h1> <p style="font-size: 8px;">Rev. 001-260199</p>																
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Elimination of Scallops</p>  <p>2-009-1</p>																	
<p>CRITICAL DETAIL:</p>  <p>Section A-A</p>  <p>2-09</p>	<p>Detail Design B - Detail Design A with Additional Intermediate Bracket Arrangement</p>  <p>2-009-2</p>																	
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Detail Attributes	Design Detail Rankings																	
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Fatigue Performance	2	1																
Ease of Inspection	1	2																
Maintenance Cost	1	2																
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<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between double bottom floor and the hopper transverse ring web. Maximum misalignment is to be less than $t/3$ where t is the thinner of the webs to be aligned and misalignment is the overhang of the thinner thickness. - Full penetration welding for the connections near the knuckle between inner bottom and transverse ring web. 																		
<p>Ref: Lloyd's Register</p>																		



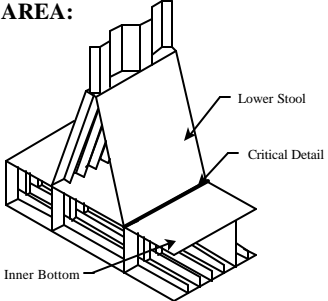
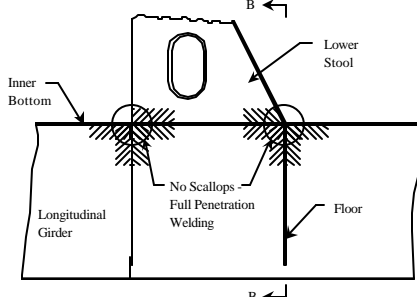
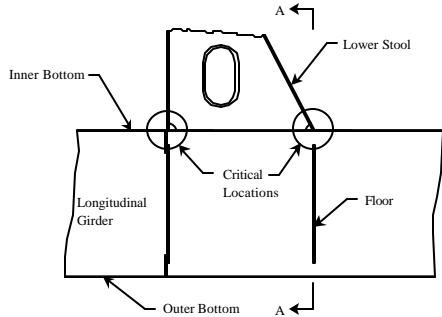
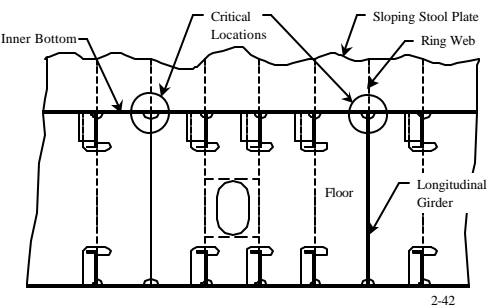
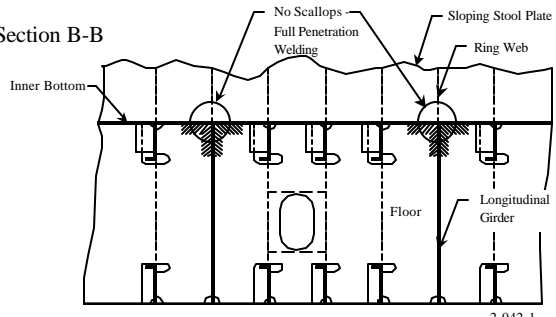
<p>Vessel Type: Bulk Carriers Critical Area: Side Shell Critical Detail: Intersection of Hopper/Topside Tank Sloping Plating and Side Shell Frame End Brackets Detail Description: Hopper Tank Backing Bracket</p>	 Ship Structure Committee 	<p>Detail</p> <h1 style="font-size: 2em; margin: 0;">21</h1> <p style="font-size: 0.8em; margin: 0;">Rev. 001-260199</p>																							
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Elimination of Scallop and Additional Carling</p>  <p style="text-align: right; font-size: 0.8em;">2-021-1</p>																								
<p>CRITICAL DETAIL:</p>  <p style="text-align: right; font-size: 0.8em;">2-21</p>	<p>Detail Design B - Full Bracket</p>  <p style="text-align: right; font-size: 0.8em;">2-021-2</p> <p>Detail Design C - Detail Design A and B with Full Collar</p>  <p style="text-align: right; font-size: 0.8em;">2-021-3</p>																								
<table border="1" style="margin: auto; border-collapse: collapse;"> <thead> <tr> <th rowspan="2" style="padding: 5px;">Detail Attributes</th> <th colspan="3" style="padding: 5px;">Design Detail Rankings</th> </tr> <tr> <th style="padding: 5px;">A</th> <th style="padding: 5px;">B</th> <th style="padding: 5px;">C</th> </tr> </thead> <tbody> <tr> <td style="padding: 5px;">Fatigue Performance</td> <td style="text-align: center; padding: 5px;">3</td> <td style="text-align: center; padding: 5px;">2</td> <td style="text-align: center; padding: 5px;">1</td> </tr> <tr> <td style="padding: 5px;">Ease of Inspection</td> <td style="text-align: center; padding: 5px;">1</td> <td style="text-align: center; padding: 5px;">1</td> <td style="text-align: center; padding: 5px;">2</td> </tr> <tr> <td style="padding: 5px;">Maintenance Cost</td> <td style="text-align: center; padding: 5px;">2</td> <td style="text-align: center; padding: 5px;">1</td> <td style="text-align: center; padding: 5px;">3</td> </tr> <tr> <td style="padding: 5px;">Fabrication Cost</td> <td style="text-align: center; padding: 5px;">2</td> <td style="text-align: center; padding: 5px;">1</td> <td style="text-align: center; padding: 5px;">3</td> </tr> </tbody> </table>			Detail Attributes	Design Detail Rankings			A	B	C	Fatigue Performance	3	2	1	Ease of Inspection	1	1	2	Maintenance Cost	2	1	3	Fabrication Cost	2	1	3
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Fabrication Cost	2	1	3																						
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between frame bracket and supporting bracket is to be maintained. Maximum misalignment is to be less than $t/3$ where t is the thinner of the webs to be aligned and misalignment is overhang of the thinner thickness. - Ensure start and stop of welding is far away as practicable from the unsupported edge corners in the supporting brackets. 																									
<p>Ref: Lloyd's Register</p>																									

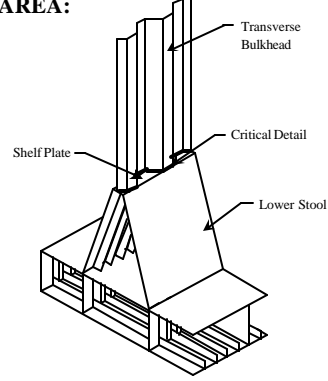
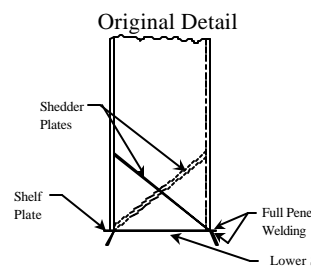
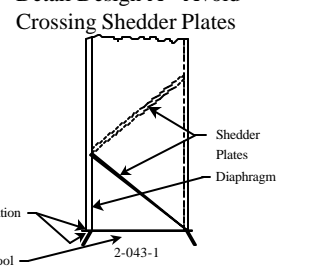
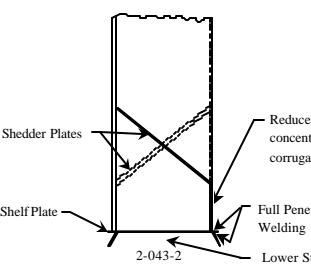
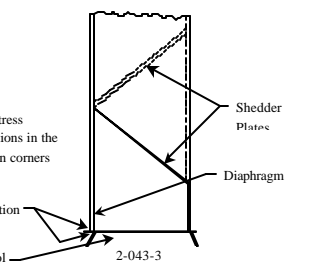
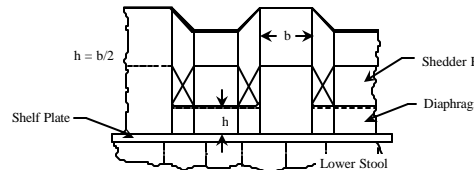
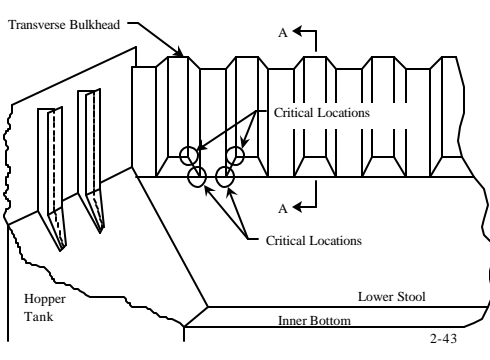
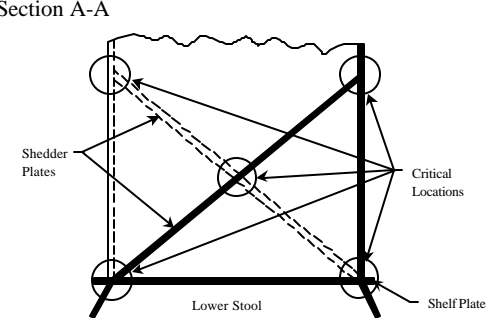
<p>Vessel Type: Bulk Carriers Critical Area: Side Shell Critical Detail: Intersection of Hopper/Topside Tank Sloping Plating and Side Shell Frame Detail Description: Hold Brackets</p>	 Ship Structure Committee 	<p>Detail</p> <h1 style="font-size: 2em; margin: 0;">22</h1> <p style="font-size: 0.8em;">Rev. 001-260199</p>																	
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <div style="display: flex; justify-content: space-between;"> <div style="width: 45%;"> <p>Detail Design A - Soft Toe</p> <p>Suggested: $r = 10t_w$ $x = 2/3 r$ $y = 1/2 r$</p>  </div> <div style="width: 45%;"> <p>View X</p>  <p>Max Taper 1:3 Recommended 1:5 when HT Steel is used</p> </div> </div> <div style="display: flex; justify-content: space-between; margin-top: 20px;"> <div style="width: 45%;"> <p>Detail Design B - Extended Toe</p> <p>Suggested: $y = 5t_w$ $y_{min} = 80 \text{ mm}$ $x > 3y$</p>  </div> <div style="width: 45%;"> <p>View X</p>  </div> </div>																		
<p>CRITICAL DETAIL:</p> 	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2" style="background-color: #cccccc;">Detail Attributes</th> <th colspan="2" style="background-color: #cccccc;">Design Detail Rankings</th> </tr> <tr> <th style="background-color: #cccccc;">A</th> <th style="background-color: #cccccc;">B</th> </tr> </thead> <tbody> <tr> <td style="text-align: left;">Fatigue Performance</td> <td>2</td> <td>1</td> </tr> <tr> <td style="text-align: left;">Ease of Inspection</td> <td>1</td> <td>1</td> </tr> <tr> <td style="text-align: left;">Maintenance Cost</td> <td>1</td> <td>1</td> </tr> <tr> <td style="text-align: left;">Fabrication Cost</td> <td>1</td> <td>2</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings		A	B	Fatigue Performance	2	1	Ease of Inspection	1	1	Maintenance Cost	1	1	Fabrication Cost	1	2
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<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between side shell frame lower and upper brackets and transverse ring webs or supporting brackets is to be maintained. Maximum misalignment is to be less than $t/3$ where t is the thinner of the webs to be aligned and misalignment is overhang of the thinner thickness. - Ensure start and stop of welding is far away as practicable from the toe of the frame brackets. A wrap around weld, free from undercut and notches, around the toe of the end bracket connections to hopper plating. 																			
<p>Ref: Lloyd's Register</p>																			



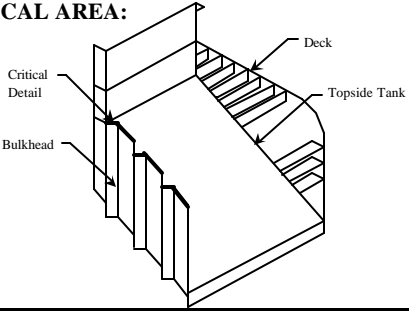
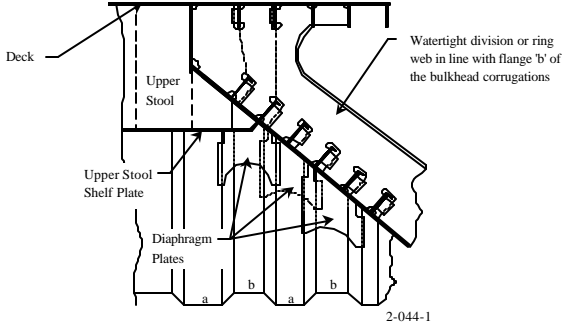
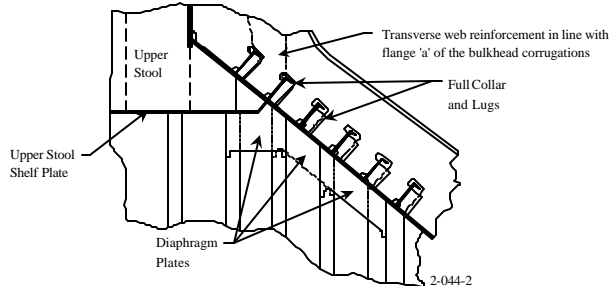
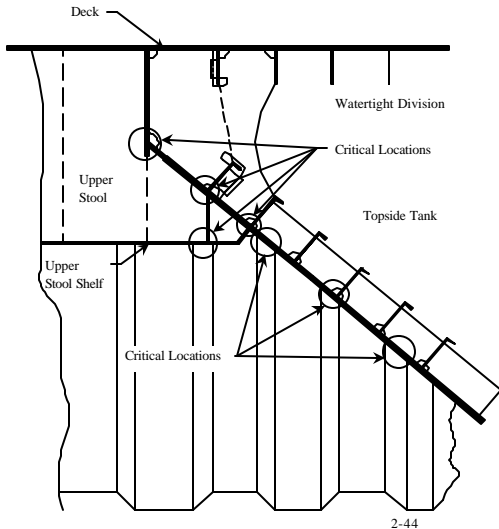
<p>Vessel Type: Bulk Carriers Critical Area: Side Shell Critical Detail: Intersection of Hopper/Topside Tank Transverse Web Flat Bar Stiffener and Hopper/Topside tank, Bottom/Deck, and Side Shell Longitudinal</p>	 Ship Structure Committee 	<p>Detail</p> <h1 style="font-size: 2em; margin: 0;">23</h1> <p>Rev. 001-260199</p>																							
<p>Detail Description:</p>																									
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Heel</p>  <p>Detail Design B - Soft Toe and Soft Heel</p>  <p>Detail Design C - Soft Toe and Soft Backing Bracket</p> 																								
<p>CRITICAL DETAIL:</p>  <p style="text-align: center;">Section A-A</p>	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2">Detail Attributes</th> <th colspan="3">Design Detail Rankings</th> </tr> <tr> <th>A</th> <th>B</th> <th>C</th> </tr> </thead> <tbody> <tr> <td>Fatigue Performance</td> <td>3</td> <td>2</td> <td>1</td> </tr> <tr> <td>Ease of Inspection</td> <td>1</td> <td>1</td> <td>2</td> </tr> <tr> <td>Maintenance Cost</td> <td>1</td> <td>2</td> <td>2</td> </tr> <tr> <td>Fabrication Cost</td> <td>1</td> <td>2</td> <td>3</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings			A	B	C	Fatigue Performance	3	2	1	Ease of Inspection	1	1	2	Maintenance Cost	1	2	2	Fabrication Cost	1	2	3
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<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between longitudinal web stiffener, transverse ring web stiffener and backing bracket is to be maintained. - Ensure start and stop of welding is far away as practicable from the heel and toe. A wrap around weld, free from undercut and notches, around the heel and toe connections of stiffener and bracket to longitudinal. 																									
<p>Ref: Lloyd's Register</p>																									



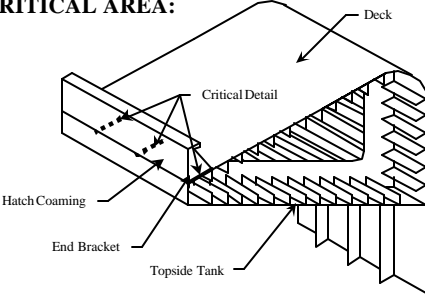
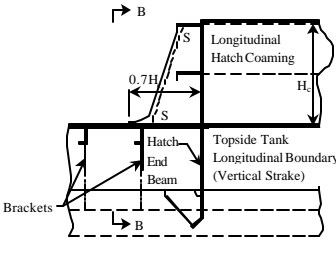
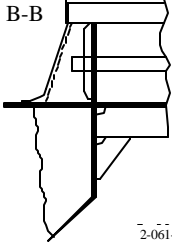
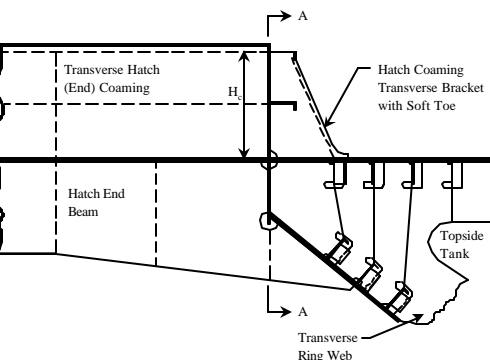
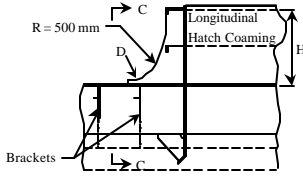
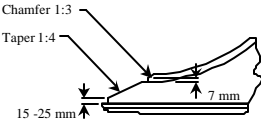
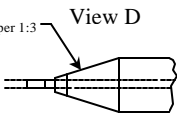
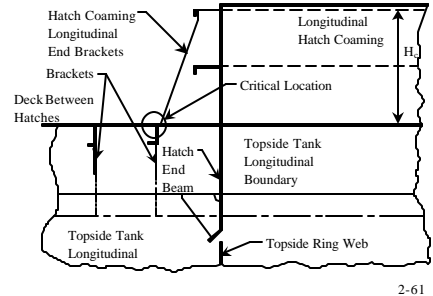
<p>Vessel Type: Bulk Carriers Critical Area: Side Shell Critical Detail: Intersection of Hopper/Topside Tank Transverse Web Tripping Bracket and Hopper/Topside tank, Bottom/Deck, and Side Shell Longitudinal</p>	 	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">24</h1> <p>Rev. 001-260199</p>																							
<p>Detail Description:</p>																									
<p>CRITICAL AREA:</p>  <p>Labels: Topside Tank, Transverse Ring Web, Critical Detail, Side Shell, Side Shell Frame, Hopper Tank Sloping Plating, Side Shell Longitudinal, Bottom Longitudinal, Critical Detail.</p>	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Heel</p>  <p style="text-align: right;">2-024-1</p> <p>Detail Design B - Soft Toe and Soft Heel</p>  <p style="text-align: right;">2-024-2</p> <p>Detail Design C - Soft Toe and Soft Backing Bracket</p>  <p style="text-align: right;">2-024-3</p>																								
<p>CRITICAL DETAIL:</p>  <p>Labels: Critical Locations, Section A-A</p> <p style="text-align: right;">2-24</p>																									
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<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between longitudinal web stiffener, transverse ring web tripping bracket and backing bracket is to be maintained. - Ensure start and stop of welding is far away as practicable from the heel and toe. A wrap around weld, free from undercut and notches, around the heel and toe connections of brackets. 																									
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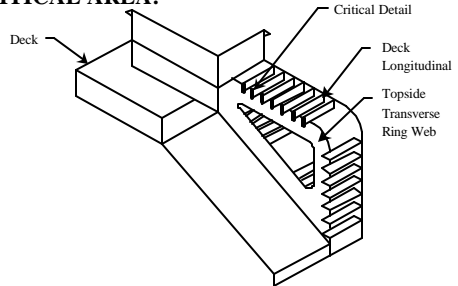
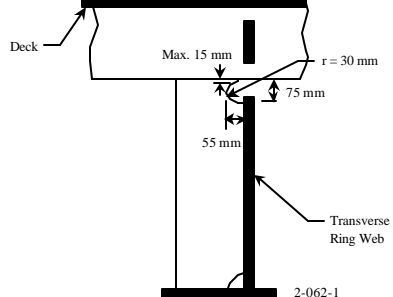
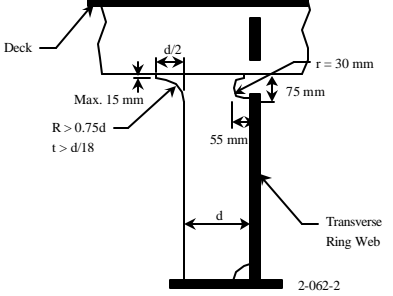
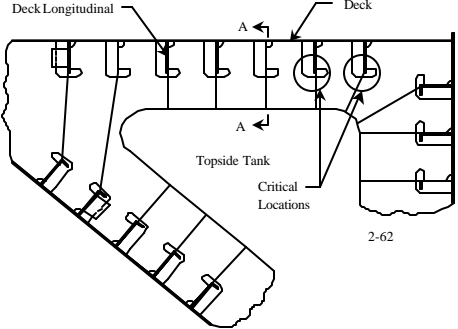
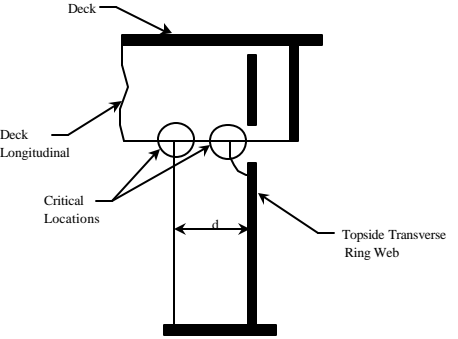
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<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p style="text-align: center;">Detail Design A - Soft Toe</p>  <p style="text-align: center;">2-041-1</p>																							
<p>CRITICAL DETAIL:</p>  <p style="text-align: center;">Section A-A</p>  <p style="text-align: center;">2-41</p>	<p style="text-align: center;">Detail Design B - Soft Toe and Soft Backing Bracket</p>  <p style="text-align: center;">2-041-2</p> <p style="text-align: center;">Detail Design C - Single Plate Integral Brackets, Additional Stability Stiffener and Soft Bracket</p> 																							
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<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between brackets and stiffener webs is to be maintained. - Maximum misalignment is to be less than $t/3$ where t is the thinner of the webs to be aligned and misalignment is overhang of the thinner thickness. - Fillet welding between inner bottom and floors. - Ensure start and stop of welding is far away as practicable from the toe of brackets or corners. A wrap around weld, free from undercut and notches, around the toe connections of the stiffeners and backing brackets to longitudinal. 																								



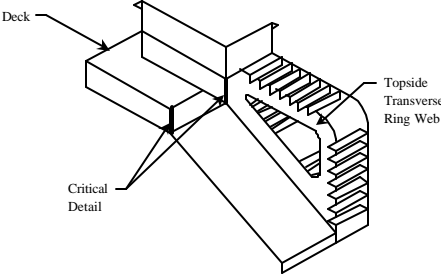
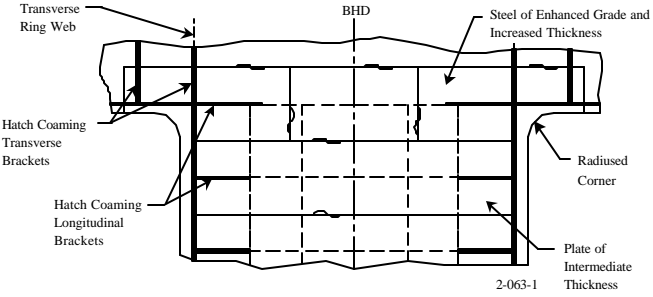
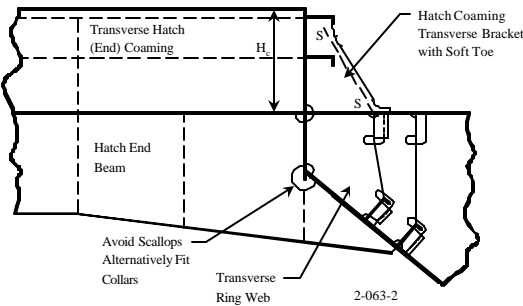
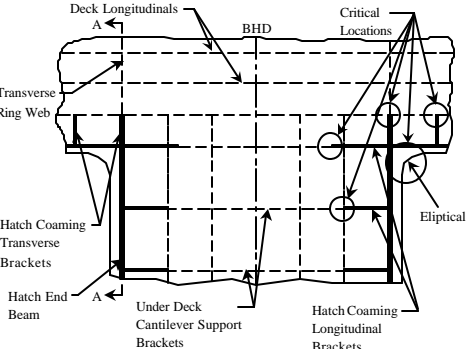
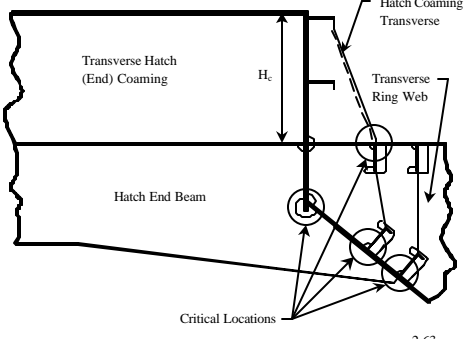
<p>Vessel Type: Bulk Carriers Critical Area: Transverse Bulkhead Critical Detail: Intersection of Lower Bulkhead Stool and Inner Bottom Girder Detail Description:</p>	 Ship Structure Committee 	<p>Detail</p> <h1 style="font-size: 2em; margin: 0;">42</h1> <p style="font-size: 0.8em; margin: 0;">Rev. 001-260199</p>																
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design</p> 																	
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Maintenance Cost	2	1																
Fabrication Cost	2	1																
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between double bottom girders and lower stool webs and between floors and stool plates is to be maintained. - Full penetration welding between inner Bottom and stool plates and near the corner intersections of primary members, floor plates, girders and lower stool webs of ballast hold side. - Fillet welding between inner bottom and stool plates for dry cargo hold side. - Ensure stop and stop of welding is as far as practicable from the corners. <p>Ref: Lloyd's Register</p>																		

<p>Vessel Type: Bulk Carriers Critical Area: Transverse Bulkhead Critical Detail: Intersection of Transverse Bulkhead Stool and Lower Transverse Bulkhead Shelf Plate Detail Description:</p>	<div style="text-align: right;"> <p>Ship Structure Committee</p> <p>FLEET TECHNOLOGY LTD</p> </div> <div style="text-align: right; font-size: 2em; font-weight: bold; margin-top: 10px;">Detail 43</div> <p style="text-align: right; font-size: 0.8em;">Rev. 001-260199</p>																							
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <div style="display: flex; justify-content: space-around;"> <div style="text-align: center;"> <p>Original Detail</p>  </div> <div style="text-align: center;"> <p>Detail Design A - Avoid Crossing Shedder Plates</p>  </div> </div> <div style="display: flex; justify-content: space-around; margin-top: 20px;"> <div style="text-align: center;"> <p>Detail Design B</p>  </div> <div style="text-align: center;"> <p>Detail Design C</p>  </div> </div> <div style="text-align: center; margin-top: 20px;">  </div>																							
<p>CRITICAL DETAIL:</p>  <p style="text-align: right; font-size: 0.8em;">2-43</p>	<p>Section A-A</p> 																							
<table border="1" style="width: 100%; border-collapse: collapse;"> <thead> <tr> <th rowspan="2" style="background-color: #cccccc;">Detail Attributes</th> <th colspan="3" style="background-color: #cccccc;">Design Detail Rankings</th> </tr> <tr> <th style="background-color: #cccccc;">A</th> <th style="background-color: #cccccc;">B</th> <th style="background-color: #cccccc;">C</th> </tr> </thead> <tbody> <tr> <td>Fatigue Performance</td> <td style="text-align: center;">3</td> <td style="text-align: center;">2</td> <td style="text-align: center;">1</td> </tr> <tr> <td>Ease of Inspection</td> <td style="text-align: center;">1</td> <td style="text-align: center;">1</td> <td style="text-align: center;">1</td> </tr> <tr> <td>Maintenance Cost</td> <td style="text-align: center;">1</td> <td style="text-align: center;">1</td> <td style="text-align: center;">1</td> </tr> <tr> <td>Fabrication Cost</td> <td style="text-align: center;">1</td> <td style="text-align: center;">2</td> <td style="text-align: center;">3</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings			A	B	C	Fatigue Performance	3	2	1	Ease of Inspection	1	1	1	Maintenance Cost	1	1	1	Fabrication Cost	1	2	3
Detail Attributes	Design Detail Rankings																							
	A	B	C																					
Fatigue Performance	3	2	1																					
Ease of Inspection	1	1	1																					
Maintenance Cost	1	1	1																					
Fabrication Cost	1	2	3																					
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between lower stool sloping plates and corrugation faces is to be maintained. - Full penetration welding at the connections of the bulkhead corrugations, diaphragm and the stool sloping plates to the lower stool shelf plate. Ensure start and stop of welding is far away as practicable from the corners of the corrugation. 																								
<p>Ref: Lloyd's Register</p>																								

<p>Vessel Type: Bulk Carriers Critical Area: Transverse Bulkhead Critical Detail: Intersection of Transverse Bulkhead and Topside Tank and Upper Stool Detail Description:</p>	 Ship Structure Committee 	<p>Detail</p> <h1 style="font-size: 2em; margin: 0;">44</h1> <p style="font-size: 0.8em;">Rev. 001-260199</p>																	
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A- Diaphragm Plates between Corrugations</p>  <p style="text-align: right; font-size: 0.8em;">2-044-1</p> <p>Detail Design B - Diaphragm Plates and Transverse Web Reinforcement</p>  <p style="text-align: right; font-size: 0.8em;">2-044-2</p>																		
<p>CRITICAL DETAIL:</p>  <p style="text-align: right; font-size: 0.8em;">2-44</p>	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr style="background-color: #cccccc;"> <th rowspan="2">Detail Attributes</th> <th colspan="2">Design Detail Rankings</th> </tr> <tr style="background-color: #cccccc;"> <th>A</th> <th>B</th> </tr> </thead> <tbody> <tr> <td style="text-align: left;">Fatigue Performance</td> <td>2</td> <td>1</td> </tr> <tr> <td style="text-align: left;">Ease of Inspection</td> <td>1</td> <td>2</td> </tr> <tr> <td style="text-align: left;">Maintenance Cost</td> <td>1</td> <td>2</td> </tr> <tr> <td style="text-align: left;">Fabrication Cost</td> <td>1</td> <td>2</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings		A	B	Fatigue Performance	2	1	Ease of Inspection	1	2	Maintenance Cost	1	2	Fabrication Cost	1	2
Detail Attributes	Design Detail Rankings																		
	A	B																	
Fatigue Performance	2	1																	
Ease of Inspection	1	2																	
Maintenance Cost	1	2																	
Fabrication Cost	1	2																	
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between transverse web and the flange of corrugations. Maximum misalignment is to be less than $t/3$ where t is the thinner of the plates to be aligned and misalignment is the overhang of the thinner thickness. - Fillet welding for connections of transverse bulkhead to topside tank and upper stool shelf plating. - Ensure start and stop of welding is far away as practicable from the critical corners. 																			
<p>Ref: Lloyd's Register</p>																			

<p>Vessel Type: Bulk Carriers Critical Area: Deck Critical Detail: Intersection of Deck Plating and Longitudinal Hatch Coaming End Bracket Detail Description:</p>	  Detail 61 <small>Rev. 001-260199</small>																	
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE: Detail Design A - Extend Longitudinal Hatch Coaming with Soft Toe</p>  																	
<p>CRITICAL DETAIL:</p> 	<p>Detail Design B - Face Flange with Soft Toe</p>   																	
<p>Section A-A</p> 	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2">Detail Attributes</th> <th colspan="2">Design Detail Rankings</th> </tr> <tr> <th>A</th> <th>B</th> </tr> </thead> <tbody> <tr> <td>Fatigue Performance</td> <td>2</td> <td>1</td> </tr> <tr> <td>Ease of Inspection</td> <td>1</td> <td>2</td> </tr> <tr> <td>Maintenance Cost</td> <td>1</td> <td>2</td> </tr> <tr> <td>Fabrication Cost</td> <td>1</td> <td>2</td> </tr> </tbody> </table>	Detail Attributes	Design Detail Rankings		A	B	Fatigue Performance	2	1	Ease of Inspection	1	2	Maintenance Cost	1	2	Fabrication Cost	1	2
Detail Attributes	Design Detail Rankings																	
	A	B																
Fatigue Performance	2	1																
Ease of Inspection	1	2																
Maintenance Cost	1	2																
Fabrication Cost	1	2																
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between hatch coaming end bracket and supporting structure is to be maintained. - Maximum misalignment is to be less than $t/3$ where t is the thinner of the webs to be aligned and misalignment is the overhang of the thinner thickness. - Full penetration welding is to be used for a distance of $0.15H_c$ from the bracket toe end ensuring start and stop of welding is as far as practicable from the toes of brackets or corners. - A wrap around weld, free from undercut or notches, around the toe connection of the bracket to deck plating. 																		
<p>Ref: Lloyd's Register</p>																		

<p>Vessel Type: Bulk Carriers</p> <p>Critical Area: Deck</p> <p>Critical Detail: Intersection of Deck Transverse Stiffener and Deck Longitudinal</p> <p>Detail Description:</p>	 <p>Ship Structure Committee</p> 	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">62</h1> <p style="font-size: 10px;">Rev. 001-260199</p>																	
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Heel</p>  <p>Detail Design B - Soft Toe and Soft Heel</p> 																		
<p>CRITICAL DETAIL:</p>  <p>Section A-A</p> 	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2" style="background-color: #cccccc;">Detail Attributes</th> <th colspan="2" style="background-color: #cccccc;">Design Detail Rankings</th> </tr> <tr> <th style="background-color: #cccccc;">A</th> <th style="background-color: #cccccc;">B</th> </tr> </thead> <tbody> <tr> <td style="text-align: left;">Fatigue Performance</td> <td>2</td> <td>1</td> </tr> <tr> <td style="text-align: left;">Ease of Inspection</td> <td>1</td> <td>1</td> </tr> <tr> <td style="text-align: left;">Maintenance Cost</td> <td>1</td> <td>2</td> </tr> <tr> <td style="text-align: left;">Fabrication Cost</td> <td>1</td> <td>2</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings		A	B	Fatigue Performance	2	1	Ease of Inspection	1	1	Maintenance Cost	1	2	Fabrication Cost	1	2
Detail Attributes	Design Detail Rankings																		
	A	B																	
Fatigue Performance	2	1																	
Ease of Inspection	1	1																	
Maintenance Cost	1	2																	
Fabrication Cost	1	2																	
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between longitudinal stiffener and topside transverse ring web stiffener is to be maintained. - Ensure start and stop of welding is as far away as practicable from the heel and toe. - A wrap around weld, free from undercut or notches, around the toe and heel connections of the stiffener to the longitudinal. 																			
<p>Ref: Lloyd's Register</p>																			

<p>Vessel Type: Bulk Carriers Critical Area: Deck Critical Detail: Intersection of Deck Hatch Corners/End Beams and Topside Tank Transverse Ring Web Detail Description:</p>	 Ship Structure Committee  FLEET TECHNOLOGY LTD	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">63</h1> <p style="font-size: 10px;">Rev. 001-260199</p>																	
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p style="text-align: center;">Detail Design A - Enhanced Steel Grade and Thickness</p>  <p style="text-align: right;">2-063-1</p> <p style="text-align: center;">Hatch End Beam Detail Design</p>  <p style="text-align: right;">2-063-2</p>																		
<p>CRITICAL DETAIL:</p>  <p style="text-align: center;">Section A-A</p>  <p style="text-align: right;">2-63</p>	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2" style="background-color: #cccccc;">Detail Attributes</th> <th colspan="2" style="background-color: #cccccc;">Design Detail Rankings</th> </tr> <tr> <th style="background-color: #cccccc;">A</th> <th style="background-color: #cccccc;">Original Detail</th> </tr> </thead> <tbody> <tr> <td style="text-align: left;">Fatigue Performance</td> <td>1</td> <td>2</td> </tr> <tr> <td style="text-align: left;">Ease of Inspection</td> <td>1</td> <td>2</td> </tr> <tr> <td style="text-align: left;">Maintenance Cost</td> <td>2</td> <td>1</td> </tr> <tr> <td style="text-align: left;">Fabrication Cost</td> <td>2</td> <td>1</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings		A	Original Detail	Fatigue Performance	1	2	Ease of Inspection	1	2	Maintenance Cost	2	1	Fabrication Cost	2	1
Detail Attributes	Design Detail Rankings																		
	A	Original Detail																	
Fatigue Performance	1	2																	
Ease of Inspection	1	2																	
Maintenance Cost	2	1																	
Fabrication Cost	2	1																	
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between hatch end beam and support in the topside tank is to be maintained. - Maximum misalignment is to be less than $t/3$ where t is the thinner of the webs to be aligned and misalignment is the overhang of the thinner thickness. - Ensure start and stop of welding is as far away as practicable from toes of brackets or corners. - A wrap around weld, free from undercut or notches, around the bracket to upper deck. <p>Ref: Lloyd's Register</p>																			

B.5.3 Container Ship Structure Details

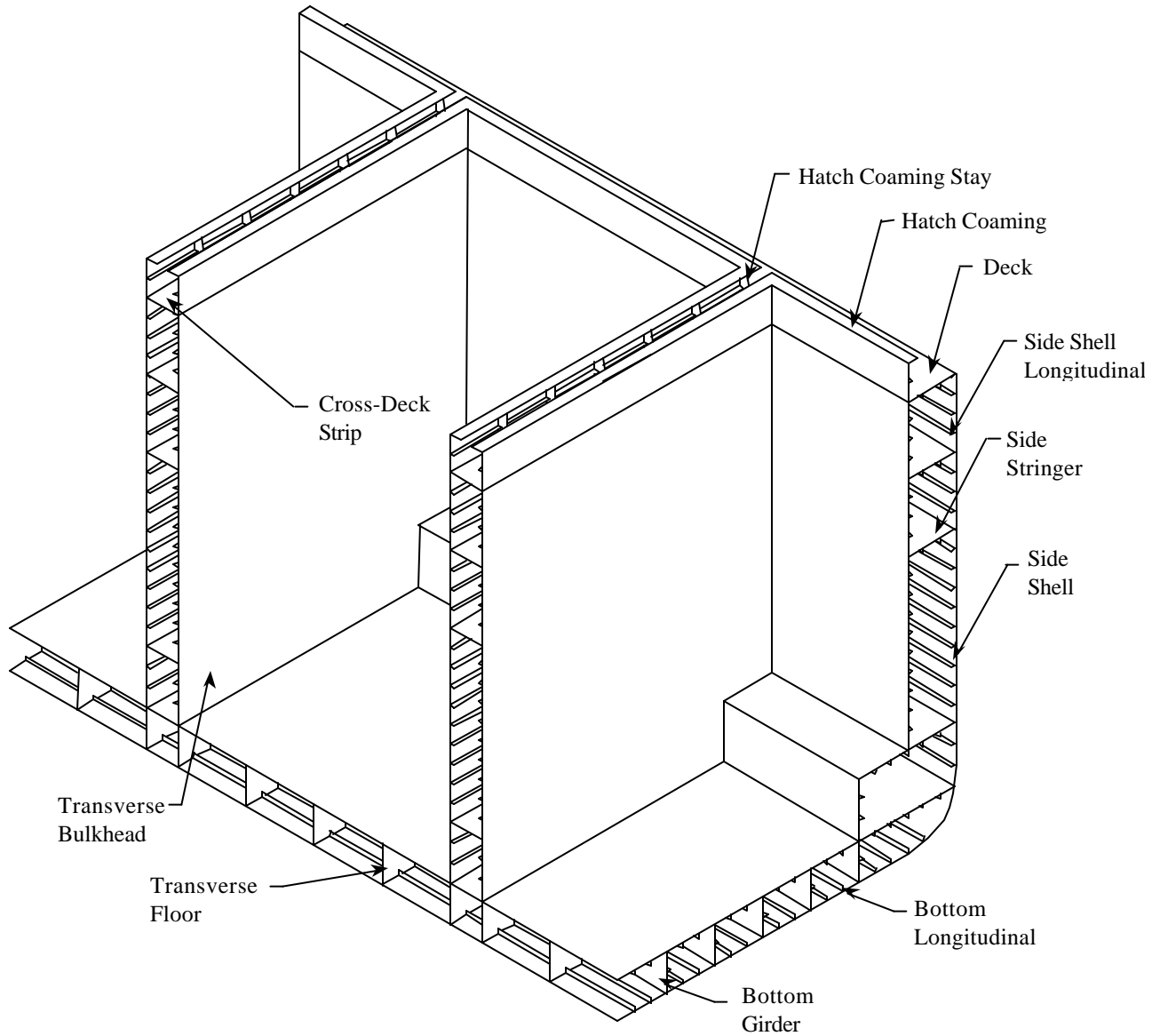


Figure B.5.3 Typical Midship Section Nomenclature for Container Ships

Table B.5.3 (a): Critical Structural Details for Double Bottom



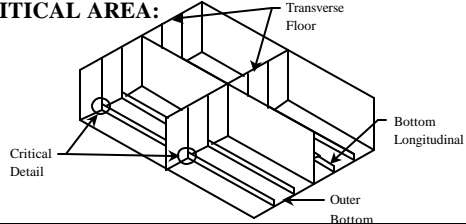
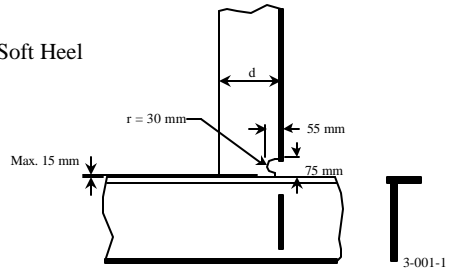
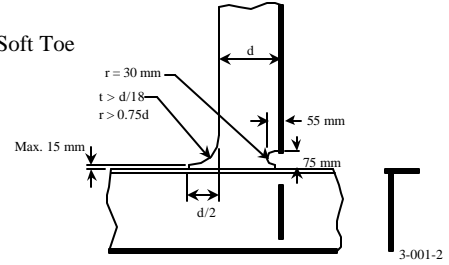
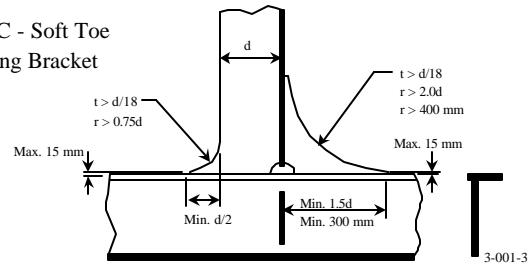
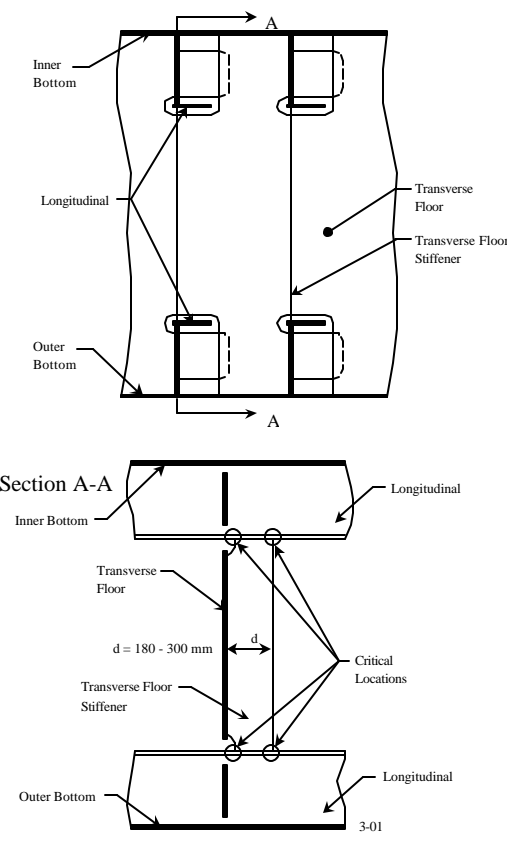
Critical Structural Details		Detail Type#	Detail Figure
Inner/Outer Bottom Longitudinals	Transverse Floor Stiffener - Asymmetric Longitudinal - Symmetric Longitudinal	1 2	
	Transverse Bulkhead	3	
Bottom Girders	Transverse Bulkhead	Section B.7	



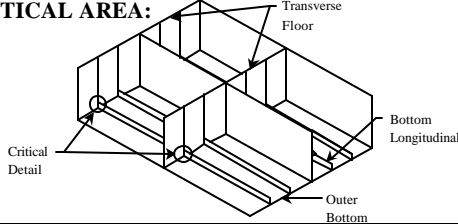
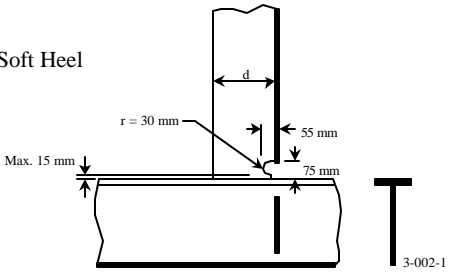
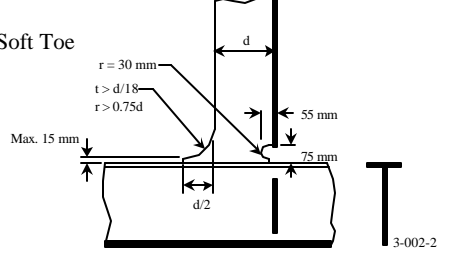
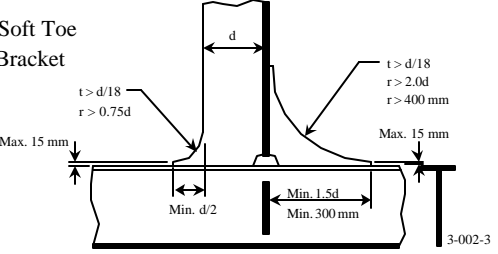
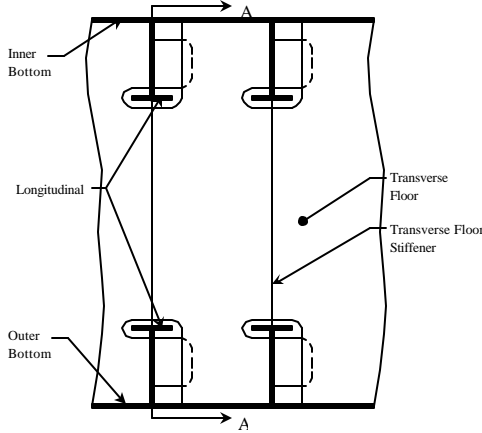
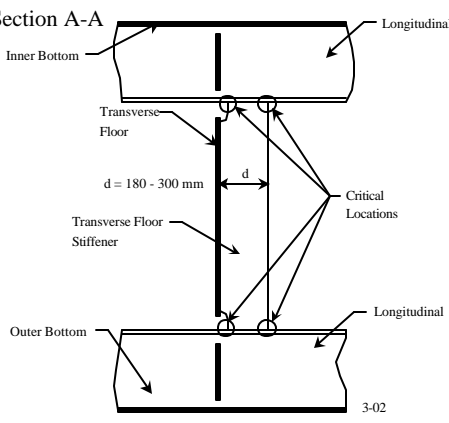
Table B.5.3 (b): Critical Structural Details for Double Side



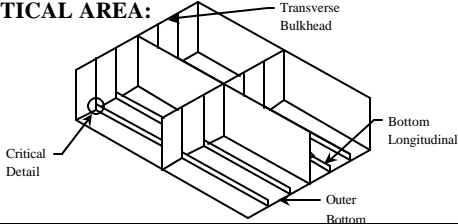
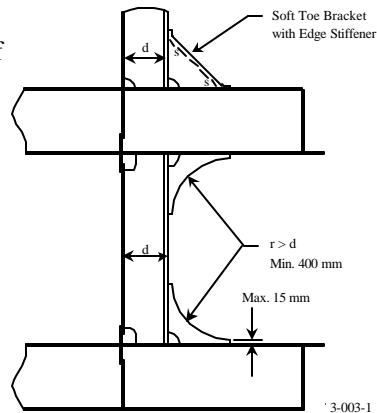
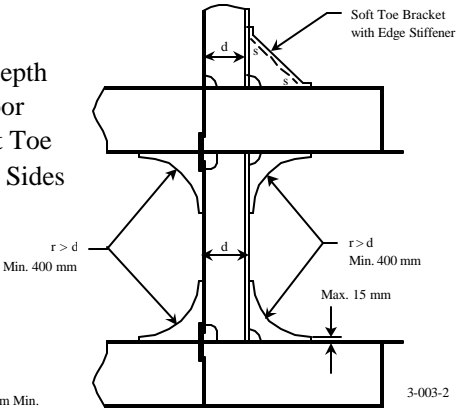
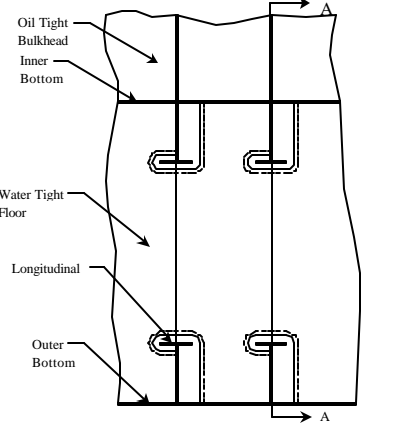
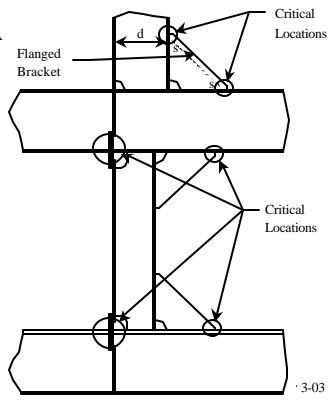
Critical Structural Details		Detail #	Detail Figure
Side Shell Longitudinals	Transverse Web Frame Stiffener - Asymmetric Longitudinal - Symmetric Longitudinal	21 22	
	Transverse Bulkhead	26	
Side Stringers	Transverse Web Frames	Section B.7	
	Transverse Bulkhead	Section B.7	
Longitudinal Bulkhead Plating	Transverse Web Frames	Section B.7	



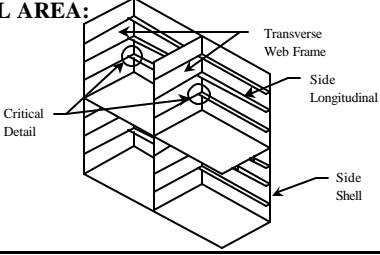
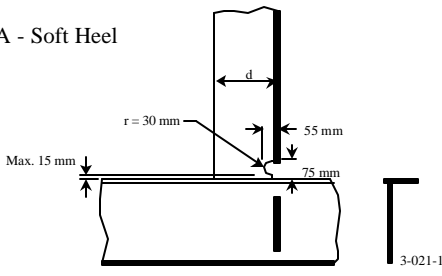
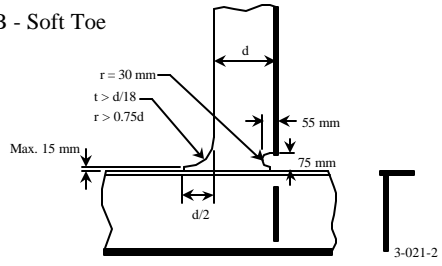
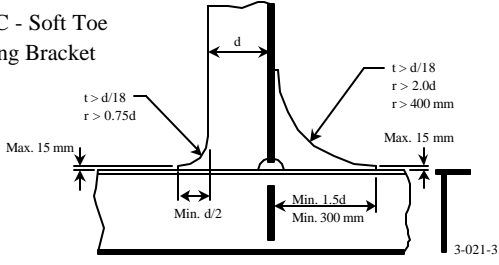
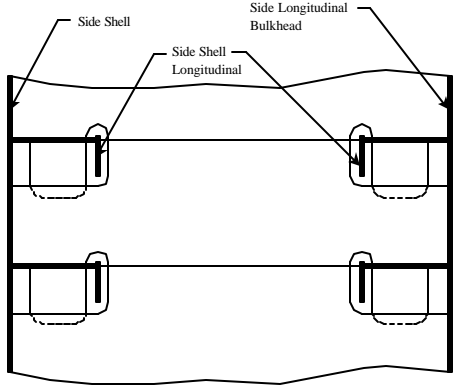
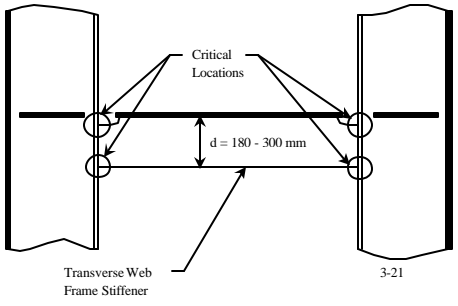
Table B.5.3 (c): Critical Structural Details for Deck and Hatch



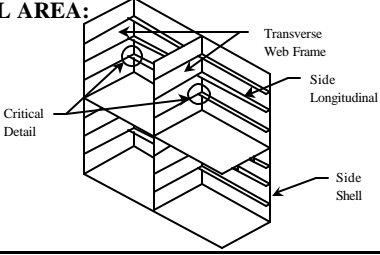
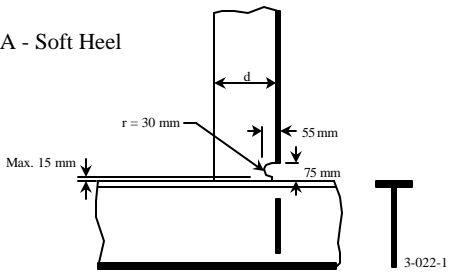
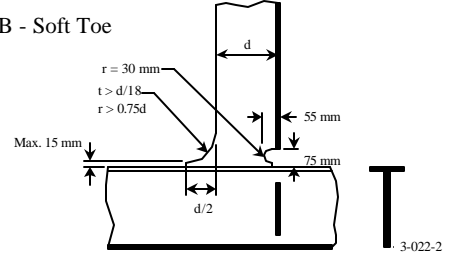
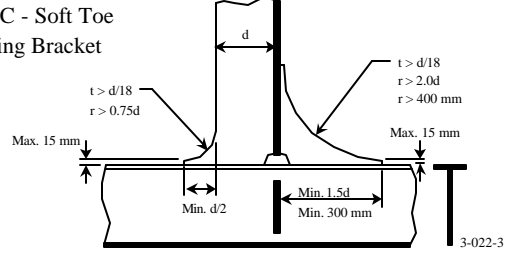
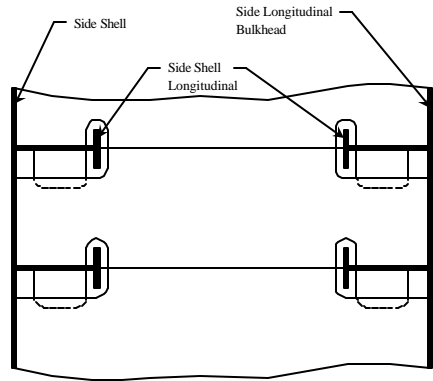
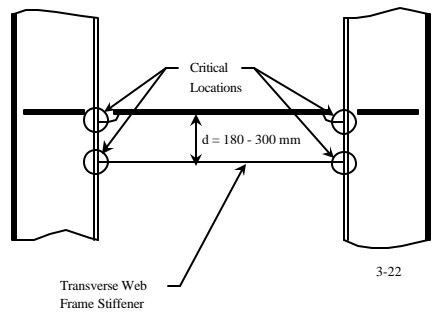
Critical Structural Details		Detail #	Detail Figure
Deck Plating	Transverse Web Frame	Section B.7	
	Cross-Deck Strips	Section B.7	
	Longitudinal Coaming Stay	Section B.7	
Hatch	Corners	Section B.7	
Hatch Coaming	Corners	Section B.7	
	Coaming Stay	Section B.7	



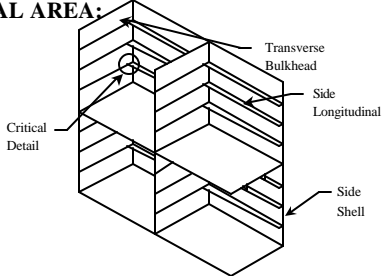
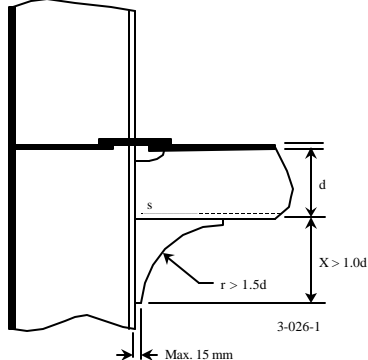
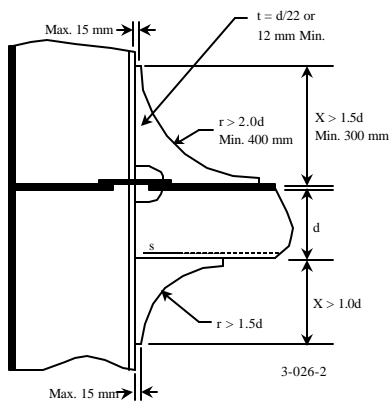
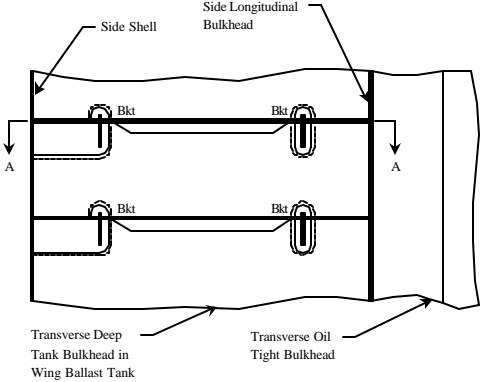
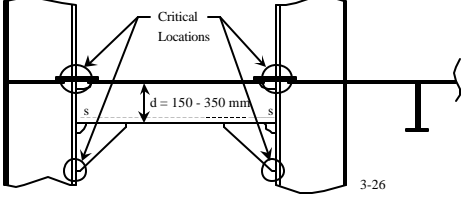
<p>Vessel Type: Container Ships Critical Area: Double Bottom Critical Detail: Intersection of Inner/Outer Bottom Longitudinal and Transverse Floor Stiffener Detail Description: Asymmetric Longitudinal</p>	 	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">1</h1> <p style="font-size: 10px;">Rev. 001-260199</p>																							
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Heel</p>  <p style="text-align: right;">3-001-1</p> <p>Detail Design B - Soft Toe and Soft Heel</p>  <p style="text-align: right;">3-001-2</p> <p>Detail Design C - Soft Toe and Soft Backing Bracket</p>  <p style="text-align: right;">3-001-3</p>																								
<p>CRITICAL DETAIL:</p>  <p style="text-align: right;">3-01</p>	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2" style="background-color: #cccccc;">Detail Indices</th> <th colspan="3" style="background-color: #cccccc;">Detail Design Good Practice</th> </tr> <tr> <th style="background-color: #cccccc;">A</th> <th style="background-color: #cccccc;">B</th> <th style="background-color: #cccccc;">C</th> </tr> </thead> <tbody> <tr> <td style="background-color: #cccccc;">Fatigue Performance</td> <td>3</td> <td>2</td> <td>1</td> </tr> <tr> <td style="background-color: #cccccc;">Ease of Inspection</td> <td>1</td> <td>1</td> <td>2</td> </tr> <tr> <td style="background-color: #cccccc;">Maintenance Cost</td> <td>1</td> <td>2</td> <td>3</td> </tr> <tr> <td style="background-color: #cccccc;">Fabrication Cost</td> <td>1</td> <td>2</td> <td>3</td> </tr> </tbody> </table>		Detail Indices	Detail Design Good Practice			A	B	C	Fatigue Performance	3	2	1	Ease of Inspection	1	1	2	Maintenance Cost	1	2	3	Fabrication Cost	1	2	3
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Maintenance Cost	1	2	3																						
Fabrication Cost	1	2	3																						
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between the transverse floor and the backing bracket is to be maintained. - A wrap around weld, free of undercut or notches, around the heel and toe connections of the floor stiffener and backing bracket. 																									

<p>Vessel Type: Container Ships Critical Area: Double Bottom Critical Detail: Intersection of Inner/Outer Bottom Longitudinal and Transverse Floor Stiffener Detail Description: Symmetric Longitudinal</p>	 	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">2</h1> <p style="font-size: 10px;">Rev. 001-260199</p>																							
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Heel</p>  <p style="text-align: right;">3-002-1</p> <p>Detail Design B - Soft Toe and Soft Heel</p>  <p style="text-align: right;">3-002-2</p> <p>Detail Design C - Soft Toe and Soft Backing Bracket</p>  <p style="text-align: right;">3-002-3</p>																								
<p>CRITICAL DETAIL:</p>  <p>Section A-A</p>  <p style="text-align: right;">3-02</p>																									
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Maintenance Cost	1	2	3																						
Fabrication Cost	1	2	3																						
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between the transverse floor and the backing bracket is to be maintained. - A wrap around weld, free of undercut or notches, around the heel and toe connections of the floor stiffener and backing bracket. 																									

<p>Vessel Type: Container Ships Critical Area: Double Bottom Critical Detail: Intersection of Inner/Outer Bottom Longitudinal and Transverse Bulkhead Detail Description:</p>	 	<p>Detail</p> <h1 style="font-size: 2em;">3</h1> <p>Rev. 001-260199</p>																	
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Increased Web Depth of Watertight Floor Stiffener and Soft Toe Brackets on Back Side</p>  <p style="text-align: right;">3-003-1</p> <p>Detail Design B - Increased Web Depth of Watertight Floor Stiffener and Soft Toe Brackets on Both Sides</p>  <p style="text-align: right;">3-003-2</p> <p><small>Note: - Higher tensile steel bracket - Bracket thickness = $d/50$ or 12 mm Min.</small></p>																		
<p>CRITICAL DETAIL:</p>  <p style="text-align: center;">Section A-A</p>  <p style="text-align: right;">3-03</p>																			
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between the floor stiffener, soft toe bracket and the web of the longitudinal is to be maintained. - A wrap around weld, free of undercut or notches, around the floor stiffener heel and the heel and toe of the bracket connection to the longitudinal. 																			
<table border="1" style="margin: auto; border-collapse: collapse;"> <thead> <tr> <th rowspan="2">Detail Attributes</th> <th colspan="2">Design Detail Rankings</th> </tr> <tr> <th>A</th> <th>B</th> </tr> </thead> <tbody> <tr> <td>Fatigue Performance</td> <td style="text-align: center;">2</td> <td style="text-align: center;">1</td> </tr> <tr> <td>Ease of Inspection</td> <td style="text-align: center;">1</td> <td style="text-align: center;">2</td> </tr> <tr> <td>Maintenance Cost</td> <td style="text-align: center;">1</td> <td style="text-align: center;">2</td> </tr> <tr> <td>Fabrication Cost</td> <td style="text-align: center;">1</td> <td style="text-align: center;">2</td> </tr> </tbody> </table>			Detail Attributes	Design Detail Rankings		A	B	Fatigue Performance	2	1	Ease of Inspection	1	2	Maintenance Cost	1	2	Fabrication Cost	1	2
Detail Attributes	Design Detail Rankings																		
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Fatigue Performance	2	1																	
Ease of Inspection	1	2																	
Maintenance Cost	1	2																	
Fabrication Cost	1	2																	

<p>Vessel Type: Container Ships Critical Area: Double Side Critical Detail: Intersection of Side Shell Longitudinal and Transverse Web Frame Stiffener Detail Description: Asymmetric Longitudinal</p>	 	<p>Detail 21</p> <p style="font-size: 0.8em;">Rev. 001-260199</p>																							
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Heel</p>  <p>Detail Design B - Soft Toe and Soft Heel</p>  <p>Detail Design C - Soft Toe and Soft Backing Bracket</p> 																								
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Fabrication Cost	1	2	3																						
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between the transverse web frame stiffener, the backing bracket and the web of the longitudinal is to be maintained. - A wrap around weld, free of undercut or notches, around the heel and toe connections of the stiffeners and the backing bracket to the longitudinal face plate. 																									

<p>Vessel Type: Container Ships Critical Area: Double Side Critical Detail: Intersection of Side Shell Longitudinal and Transverse Web Frame Stiffener Detail Description: Symmetric Longitudinal</p>	  Detail 22 <small>Rev. 001-260199</small>																							
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Heel</p>  <p>Detail Design B - Soft Toe and Soft Heel</p>  <p>Detail Design C - Soft Toe and Soft Backing Bracket</p> 																							
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<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between the transverse web frame stiffener, the backing bracket and the web of the longitudinal is to be maintained. - A wrap around weld, free of undercut or notches, around the heel and toe connections of the stiffeners and the backing bracket to the longitudinal face plate. 																								

<p>Vessel Type: Container Ships Critical Area: Double Side Critical Detail: Intersection of Side Shell Longitudinal and Transverse Bulkhead Horizontal Stiffener Detail Description:</p>	 	<p>Detail</p> <h1 style="font-size: 2em;">26</h1> <p>Rev. 001-020299</p>																	
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Soft Toe</p>  <p>3-026-1</p> <p>Detail Design B - Soft Toe and Soft Backing Bracket</p>  <p>3-026-2</p>																		
<p>CRITICAL DETAIL:</p>  <p>Section A-A</p>  <p>3-26</p>	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2">Detail Indices</th> <th colspan="2">Detail Design Good Practice</th> </tr> <tr> <th>A</th> <th>B</th> </tr> </thead> <tbody> <tr> <td>Fatigue Performance</td> <td>2</td> <td>1</td> </tr> <tr> <td>Ease of Inspection</td> <td>1</td> <td>2</td> </tr> <tr> <td>Maintenance Cost</td> <td>1</td> <td>2</td> </tr> <tr> <td>Fabrication Cost</td> <td>1</td> <td>2</td> </tr> </tbody> </table>		Detail Indices	Detail Design Good Practice		A	B	Fatigue Performance	2	1	Ease of Inspection	1	2	Maintenance Cost	1	2	Fabrication Cost	1	2
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Fatigue Performance	2	1																	
Ease of Inspection	1	2																	
Maintenance Cost	1	2																	
Fabrication Cost	1	2																	
<p>COMMENTS:</p> <ul style="list-style-type: none"> - Alignment between the web of the bulkhead horizontal stiffener, the soft toe brackets and the web of the side longitudinal is to be maintained. - A wrap around weld, free of undercut or notches, around the heel connections of bulkhead stiffener and toe connection of brackets. <p>Ref: Lloyd's Register</p>																			

B.5.4 Warship Structure Details

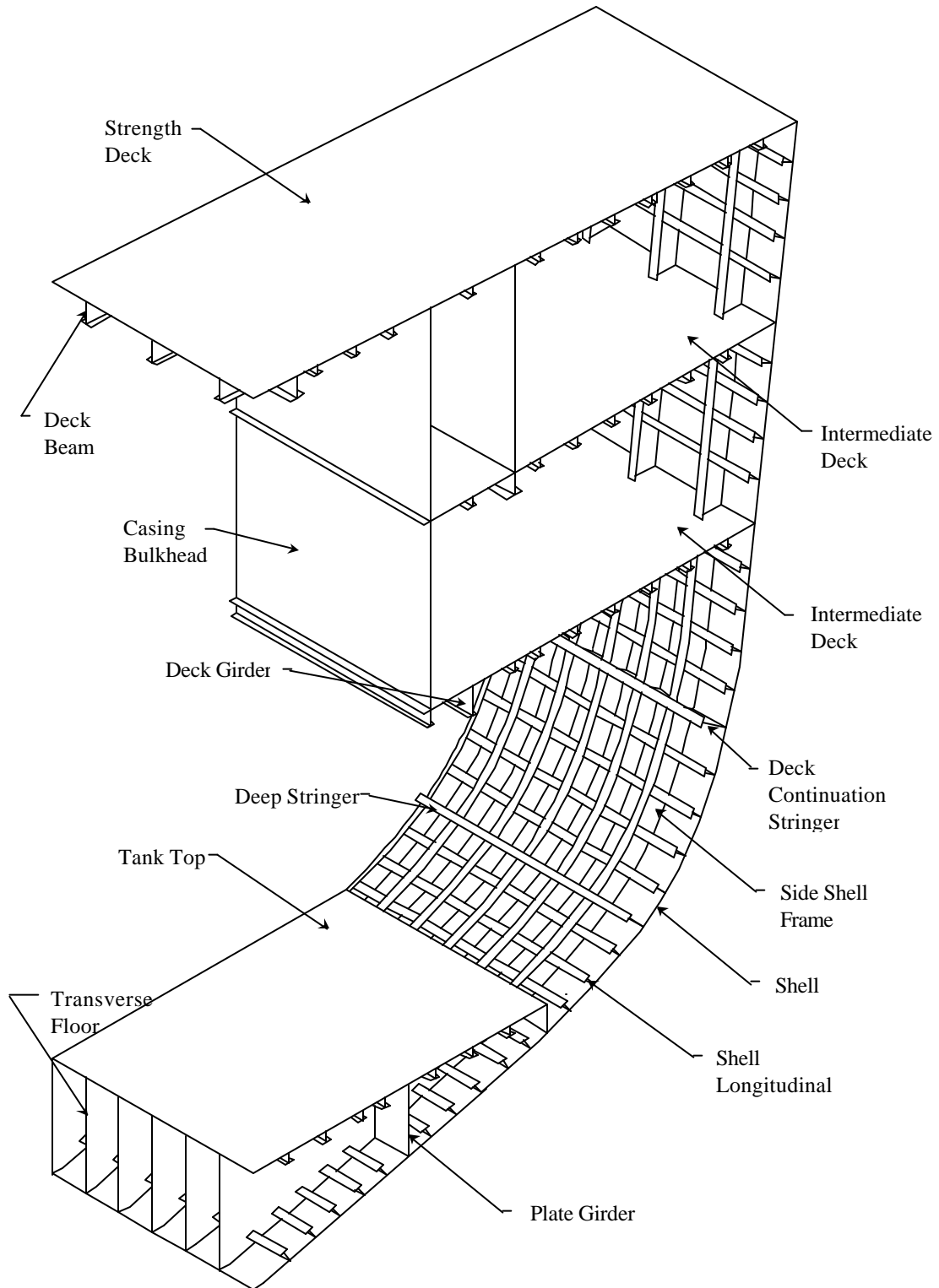


Figure B.5.4: Typical Midship Section Nomenclature for War Ships

Table B.5.4 (a): Critical Structural Details for Bottom



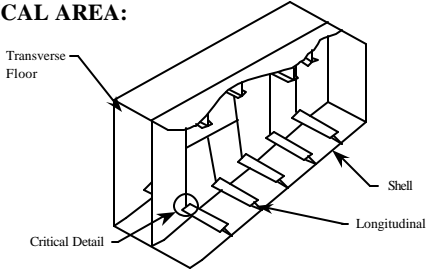
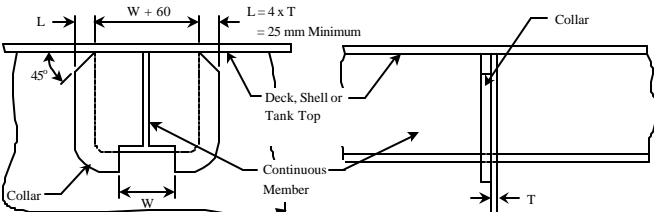
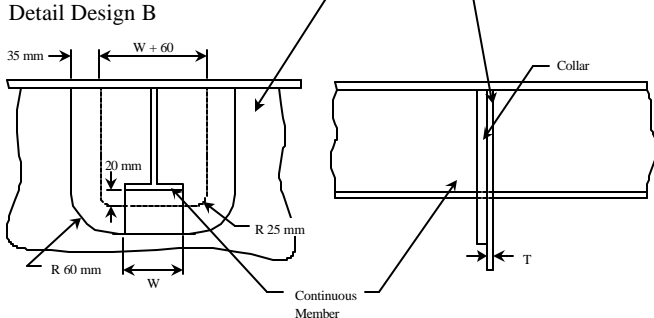
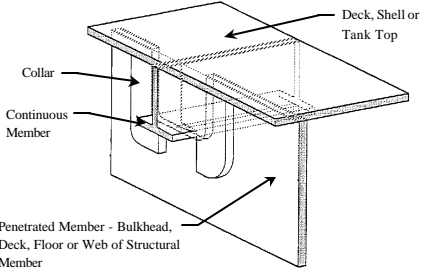
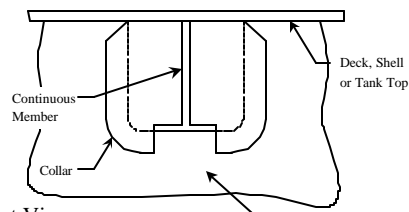
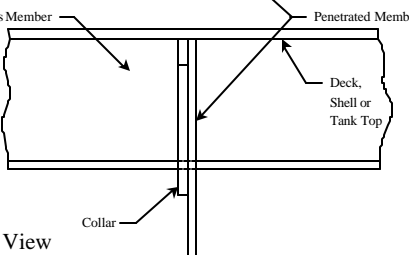
Critical Structural Details		Detail Type #	Detail Figure
Bottom Longitudinals	Transverse Floor	1	
	- Water-Tight Collar	2	
	- Non Water-Tight Collar with Lug Compensation Pieces	3	
	Transverse Floor Stiffener	4	
Transverse Floor Stiffener	Deck Plating	5	
	- Ending of Flat Bar Stiffener	6	
	- Ending of Fabricated Tee Stiffener	7	
	- Ending of Rolled Tee Stiffener	8	
	Transverse Floor Stiffener	9	
Transverse Floor Cut-out	- Flat Bar Compensation Piece	9	



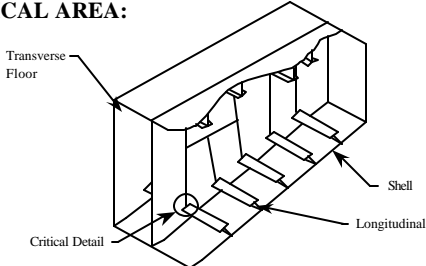
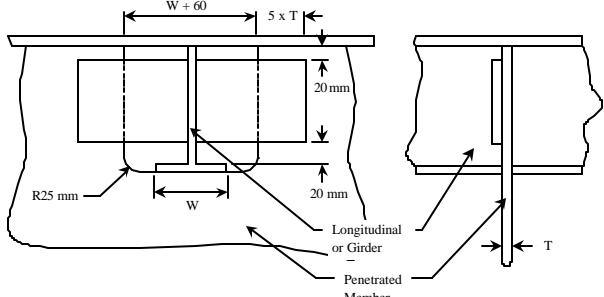
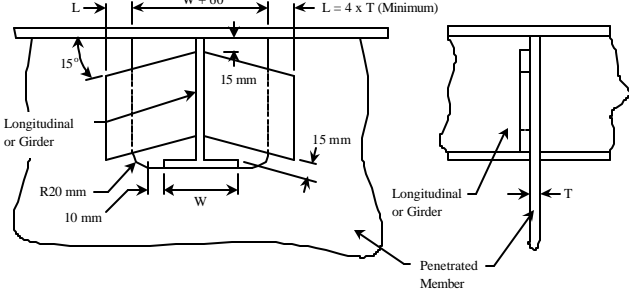
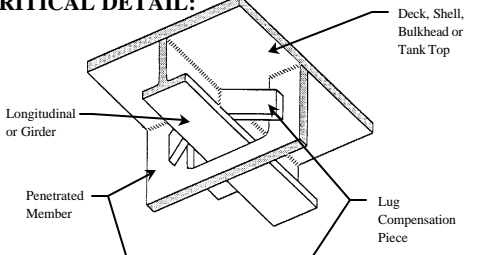
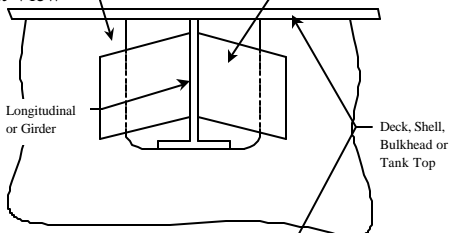
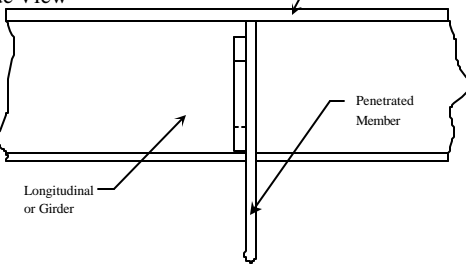
Table B.5.4 (b): Critical Structural Details for Side Shell



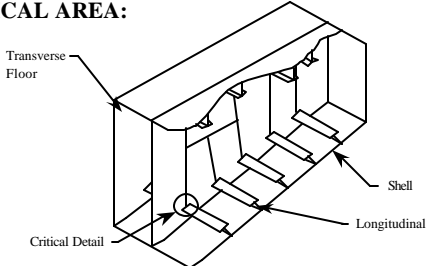
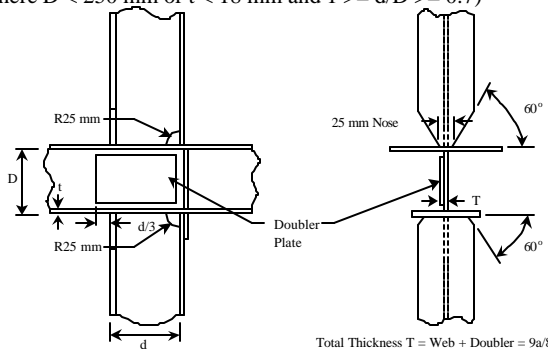
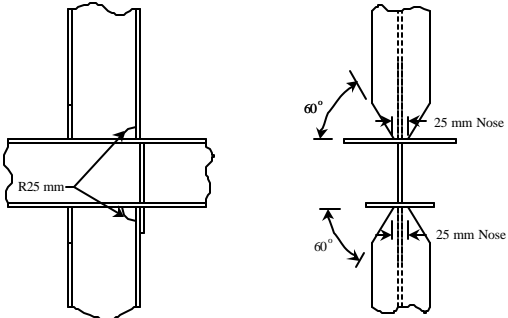
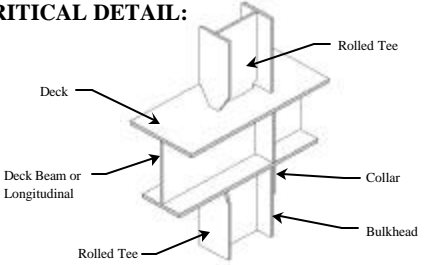
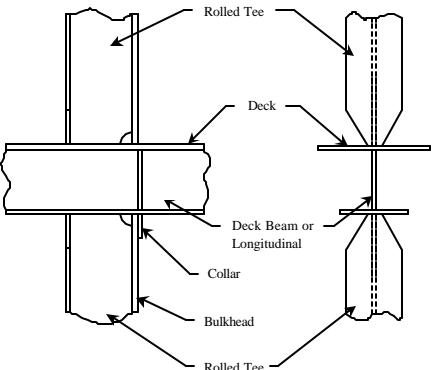
Critical Structural Details		Detail Type #	Detail Figure
Side Shell Frame	Deck Plating	10	
	- Water-Tight Collar		
	Side Shell Frame	11	
	- Rolled Tee to Fabricated Tee of Similar or Differing Depths		
	Web Frame or Bulkhead	12	
	- End Connection of Side Shell Frame	13	
	Deck Beam	14	
	- End Connection of Deck Beam	15	
- End Connection of Side Shell Frame at Deck			
Deep Stringer	16		
- Egg-Boxing of Rolled Tees of Differing Depths	15		
- Egg-Boxing of Rolled Tees of Similar Depths	16		



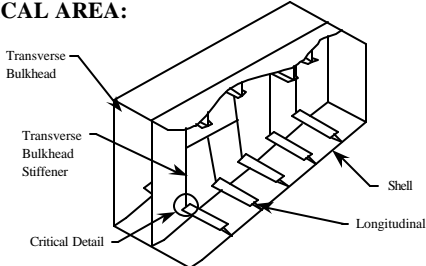
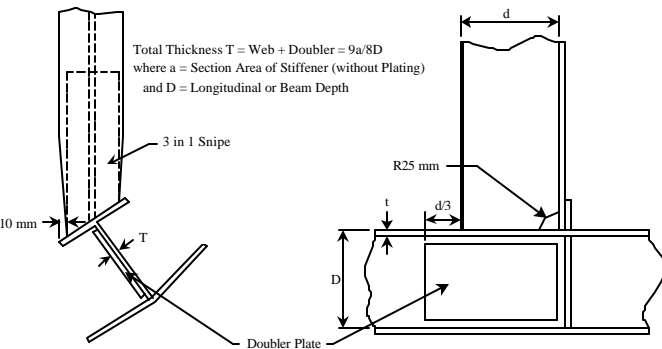
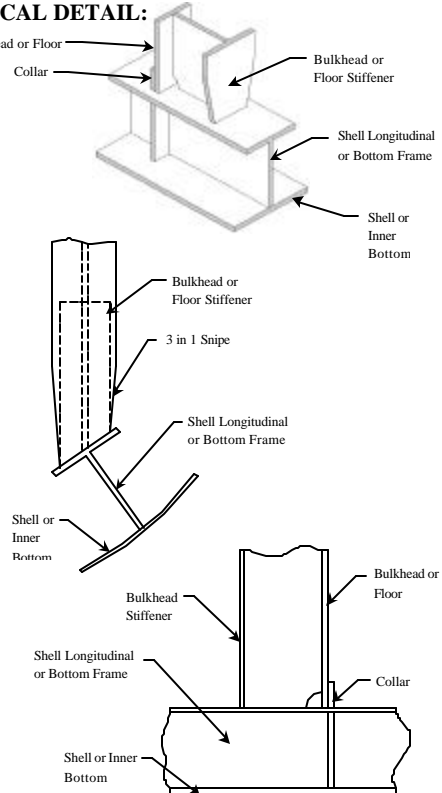
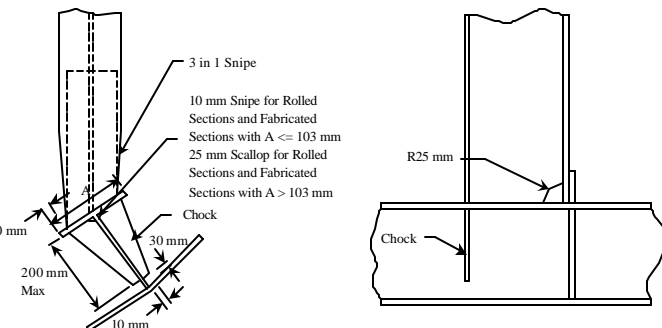
Table B.5.4 (c): Critical Structural Details for Deck



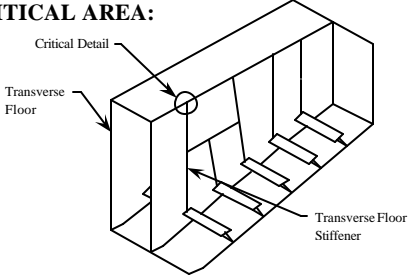
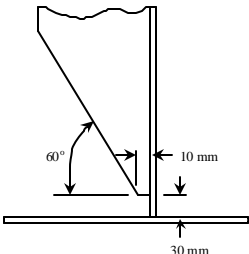
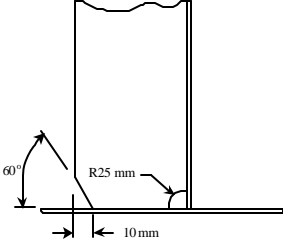
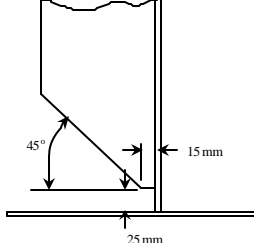
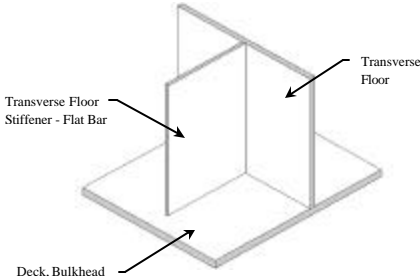
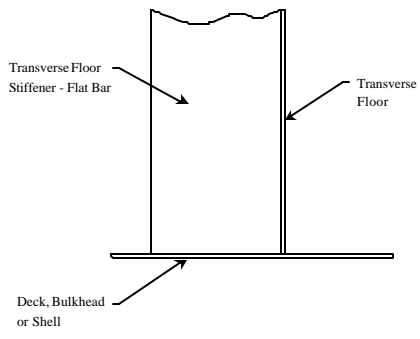
Critical Structural Details		Detail Type #	Detail Figure
Deck Beams	Deck Beams - Rolled Tee to Fabricated Tee of Similar or Differing Depths	17	
	- Rolled or Fabricated Tee to Rolled or Fabricated Tee of Similar or Differing Flange Widths	18	
	- Rolled or Fabricated Tee to Fabricated Tee of Differing Depths	19	
	Deck Longitudinal Girders - Tripping Bracket Detail	20	
	- Tee End Connection to Deeper Tee	21	
- Tee End Connection to Similar Depth Tee	22		
Deck Longitudinals - Non Water-Tight Collar with Lug Compensation Pieces	23		
Casing Bulkhead - End Connection of Deck Beam in way way of Deck Opening	24		
Casing Bulkhead - End Connection of Deck Beam at Girder Below Casing Bulkhead	25		



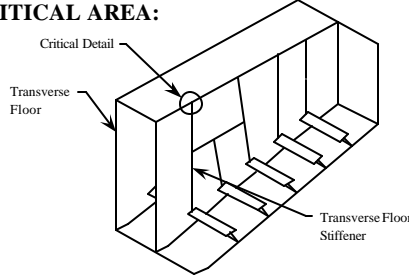
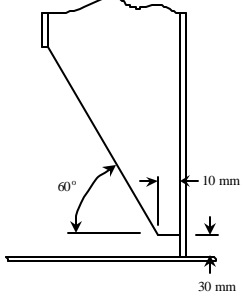
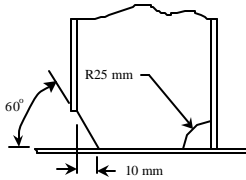
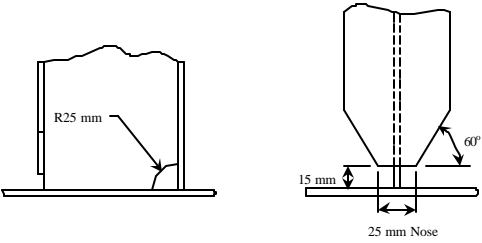
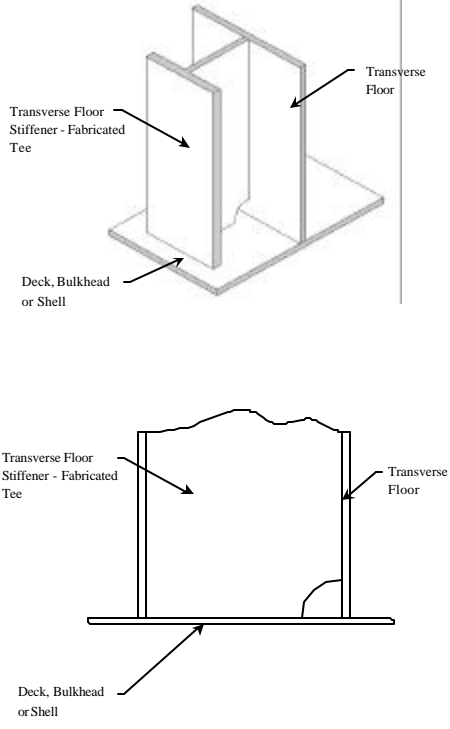
<p>Vessel Type: War Ships Critical Area: Bottom Critical Detail: Intersection of Bottom Longitudinal and Transverse Floor Detail Description: Water-Tight Collar</p>	 	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">1</h1> <p style="font-size: 10px;">Rev. 001-260199</p>																	
<p>CRITICAL AREA:</p>  <p>Labels: Transverse Floor, Shell, Longitudinal, Critical Detail</p>	<p>DETAIL DESIGN GOOD PRACTICE:</p> <div style="display: flex; flex-direction: column; align-items: center;"> <div style="margin-bottom: 20px;"> <p>Detail Design A</p>  </div> <div> <p>Detail Design B</p>  </div> </div>																		
<p>CRITICAL DETAIL:</p>  <p>Labels: Deck, Shell or Tank Top, Collar, Continuous Member, Penetrated Member - Bulkhead, Deck, Floor or Web of Structural Member</p>  <p>Labels: Continuous Member, Collar, Deck, Shell or Tank Top</p> <p>Front View</p>  <p>Labels: Continuous Member, Penetrated Member, Deck, Shell or Tank Top, Collar</p> <p>Side View</p>	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr style="background-color: #cccccc;"> <th rowspan="2">Detail Attributes</th> <th colspan="2">Design Detail Rankings</th> </tr> <tr style="background-color: #cccccc;"> <th>A</th> <th>B</th> </tr> </thead> <tbody> <tr> <td style="text-align: left;">Fatigue Performance</td> <td>2</td> <td>1</td> </tr> <tr> <td style="text-align: left;">Ease of Inspection</td> <td>1</td> <td>1</td> </tr> <tr> <td style="text-align: left;">Maintenance Cost</td> <td>1</td> <td>1</td> </tr> <tr> <td style="text-align: left;">Fabrication Cost</td> <td>1</td> <td>2</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings		A	B	Fatigue Performance	2	1	Ease of Inspection	1	1	Maintenance Cost	1	1	Fabrication Cost	1	2
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<p>COMMENTS:</p>																			



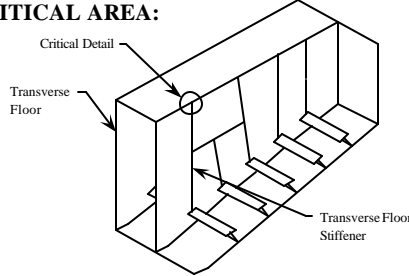
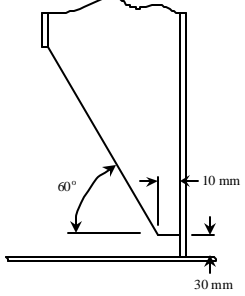
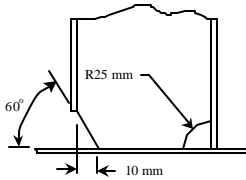
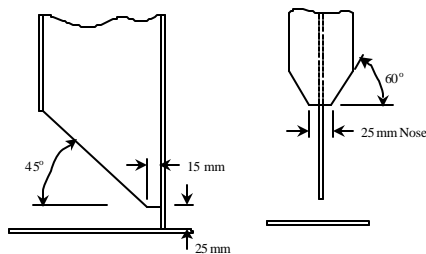
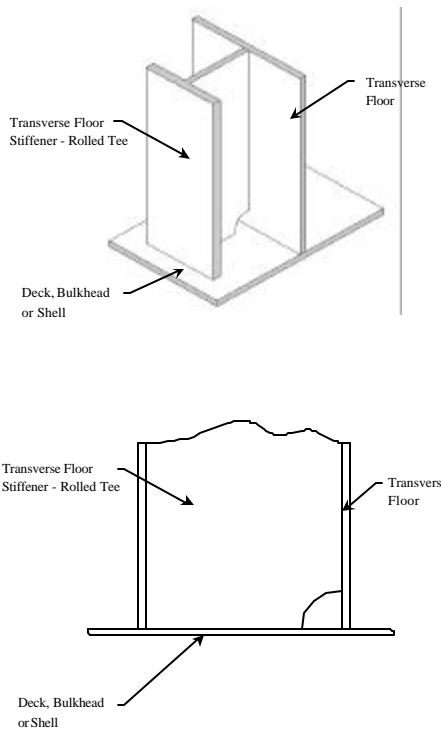
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<p>CRITICAL AREA:</p>  <p>Labels: Transverse Floor, Shell, Longitudinal, Critical Detail</p>	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A</p>  <p>Detail Design B</p> 																		
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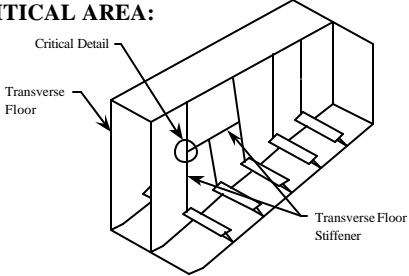
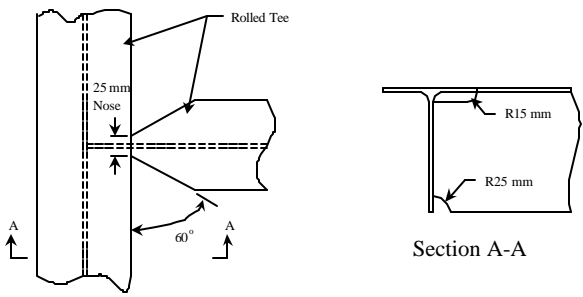
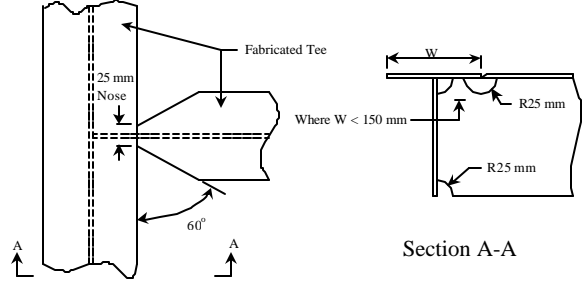
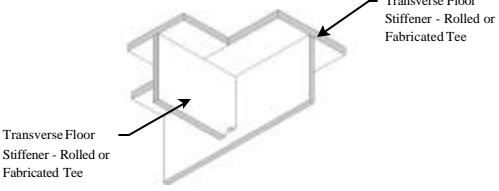
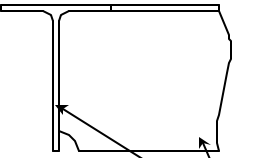
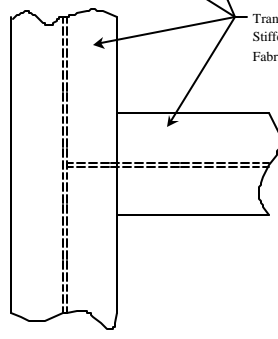
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<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - With Doubler Plate (where $D < 250 \text{ mm}$ or $t < 18 \text{ mm}$ and $1 \geq d/D \geq 0.7$)</p>  <p style="font-size: 0.8em;">Total Thickness $T = \text{Web} + \text{Doubler} = 9a/8D$ where $a = \text{Section Area of Stiffener (without Plating)}$ and $D = \text{Longitudinal or Beam Depth}$</p> <p>Detail Design B - Without Doubler Plate</p> 																		
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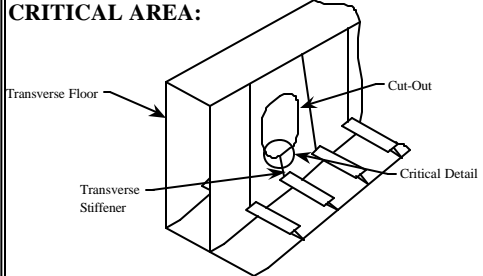
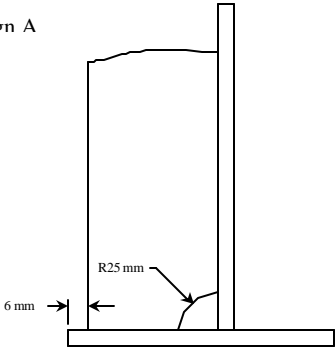
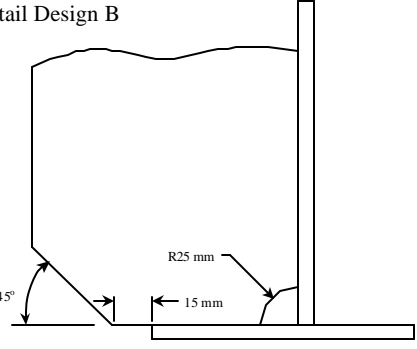
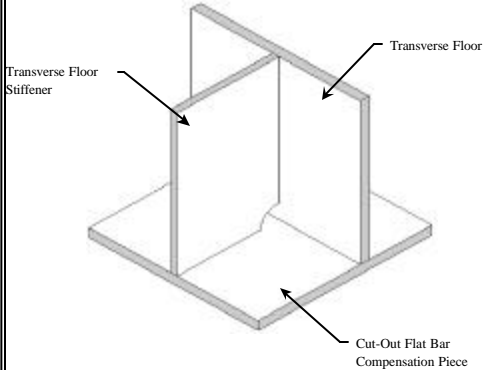
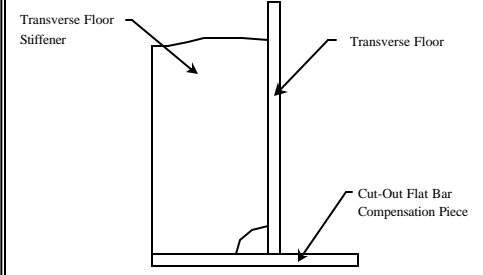
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<p>CRITICAL DETAIL:</p> 	<p>Detail Design B - With Chock</p>  <p>10 mm Snipe for Rolled Sections and Fabricated Sections with $A \leq 103$ mm 25 mm Scallop for Rolled Sections and Fabricated Sections with $A > 103$ mm</p>																					
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<p>Vessel Type: War Ships Critical Area: Bottom Critical Detail: Connection of Transverse Floor Stiffener and Deck Plating Detail Description: Ending of Flat Bar Stiffener</p>	 	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">5</h1> <p style="font-size: 8px;">Rev. 001-260199</p>																							
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<p>Vessel Type: War Ships Critical Area: Bottom Critical Detail: Connection of Transverse Floor Stiffener and Deck Plating Detail Description: Ending of Fabricated Tee Stiffener</p>	 Ship Structure Committee  FTL FLEET TECHNOLOGY LTD	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">6</h1> <p style="font-size: 10px;">Rev. 001-260199</p>																							
<p>CRITICAL AREA:</p>  <p>Labels: Critical Detail, Transverse Floor, Transverse Floor Stiffener</p>	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A</p>  <p>Detail Design B</p>  <p>Detail Design C</p> 																								
<p>CRITICAL DETAIL:</p>  <p>Labels: Transverse Floor Stiffener - Fabricated Tee, Transverse Floor, Deck, Bulkhead or Shell</p>	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2" style="background-color: #cccccc;">Detail Attributes</th> <th colspan="3" style="background-color: #cccccc;">Design Detail Rankings</th> </tr> <tr> <th style="background-color: #cccccc;">A</th> <th style="background-color: #cccccc;">B</th> <th style="background-color: #cccccc;">C</th> </tr> </thead> <tbody> <tr> <td style="background-color: #cccccc;">Fatigue Performance</td> <td>1</td> <td>3</td> <td>2</td> </tr> <tr> <td style="background-color: #cccccc;">Ease of Inspection</td> <td>1</td> <td>2</td> <td>2</td> </tr> <tr> <td style="background-color: #cccccc;">Maintenance Cost</td> <td>1</td> <td>2</td> <td>2</td> </tr> <tr> <td style="background-color: #cccccc;">Fabrication Cost</td> <td>1</td> <td>2</td> <td>3</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings			A	B	C	Fatigue Performance	1	3	2	Ease of Inspection	1	2	2	Maintenance Cost	1	2	2	Fabrication Cost	1	2	3
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<p>Vessel Type: War Ships Critical Area: Bottom Critical Detail: Connection of Transverse Floor Stiffener and Deck Plating Detail Description: Ending of Rolled Tee Stiffener</p>	 Ship Structure Committee  FTL FLEET TECHNOLOGY LTD	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">7</h1> <p style="font-size: 8px;">Rev. 001-260199</p>																							
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<p>Vessel Type: War Ships Critical Area: Bottom Critical Detail: Connection of Transverse Floor Stiffener and Transverse Floor Stiffener Detail Description:</p>	 	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">8</h1> <p style="font-size: 8px;">Rev. 001-260199</p>																	
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<p>COMMENTS:</p>																			

<p>Vessel Type: War Ships Critical Area: Bottom Critical Detail: Connection of Transverse Floor Stiffener and Transverse Floor Cut-Out Detail Description: Flat Bar Compensation Piece</p>	 Ship Structure Committee  Rev. 001-260199	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">9</h1>																	
<p>CRITICAL AREA:</p>  <p>Labels: Transverse Floor, Transverse Stiffener, Cut-Out, Critical Detail</p>	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A</p>  <p>Labels: 6 mm, R25 mm</p> <p>Detail Design B</p>  <p>Labels: 45°, 15 mm, R25 mm</p>																		
<p>CRITICAL DETAIL:</p>  <p>Labels: Transverse Floor Stiffener, Transverse Floor, Cut-Out Flat Bar Compensation Piece</p>  <p>Labels: Transverse Floor Stiffener, Transverse Floor, Cut-Out Flat Bar Compensation Piece</p>	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2">Detail Attributes</th> <th colspan="2">Design Detail Rankings</th> </tr> <tr> <th>A</th> <th>B</th> </tr> </thead> <tbody> <tr> <td>Fatigue Performance</td> <td>1</td> <td>1</td> </tr> <tr> <td>Ease of Inspection</td> <td>1</td> <td>1</td> </tr> <tr> <td>Maintenance Cost</td> <td>1</td> <td>1</td> </tr> <tr> <td>Fabrication Cost</td> <td>1</td> <td>1</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings		A	B	Fatigue Performance	1	1	Ease of Inspection	1	1	Maintenance Cost	1	1	Fabrication Cost	1	1
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Ease of Inspection	1	1																	
Maintenance Cost	1	1																	
Fabrication Cost	1	1																	
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Vessel Type: War Ships
Critical Area: Side
Critical Detail: Intersection of Side Shell Frame and Deck Plating

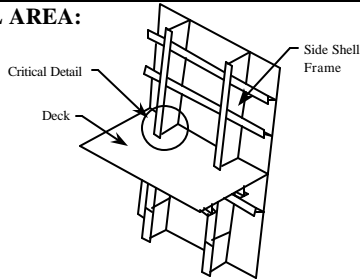


Detail
10

Rev. 001-260199

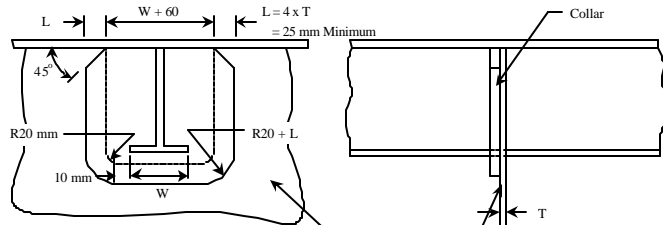
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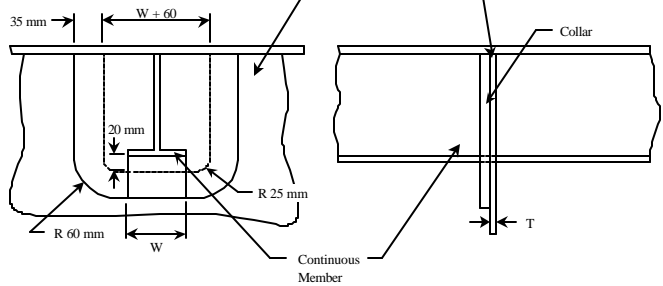


DETAIL DESIGN GOOD PRACTICE:

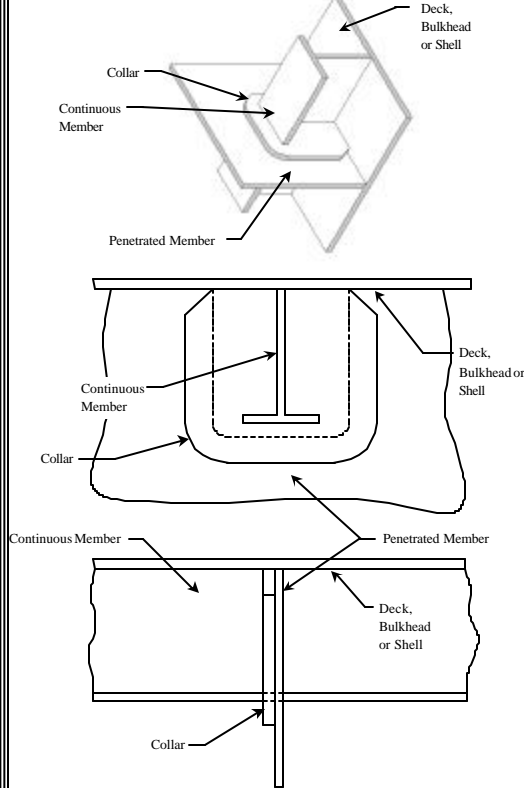
Detail Design A



Detail Design B



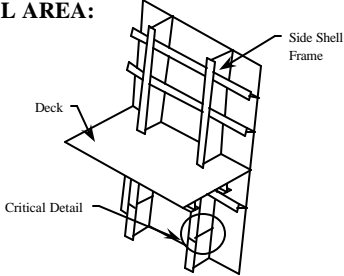
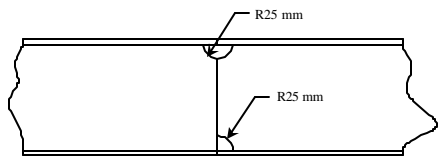
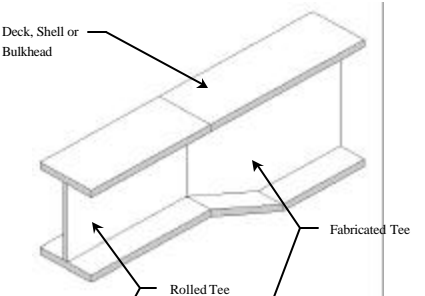
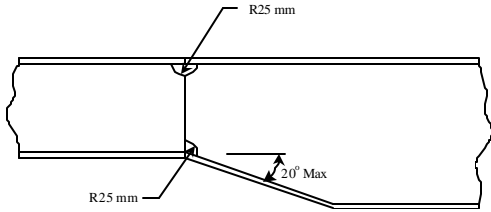




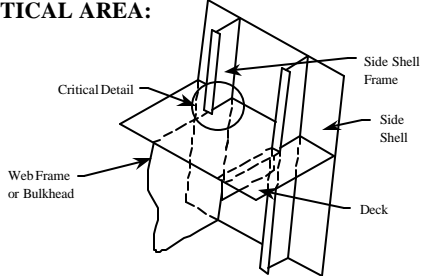
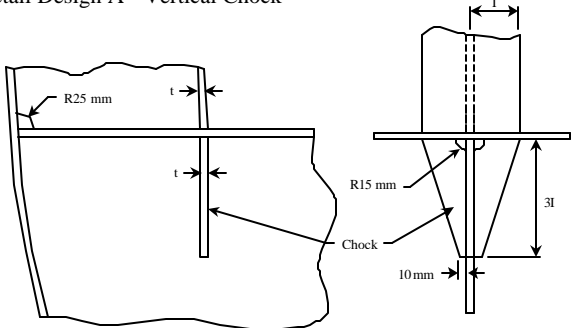
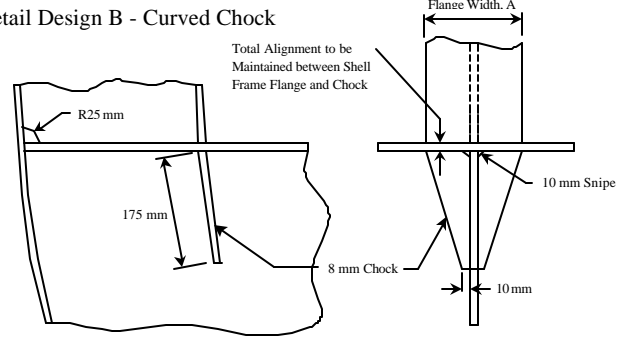
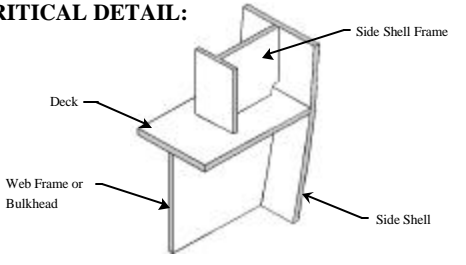
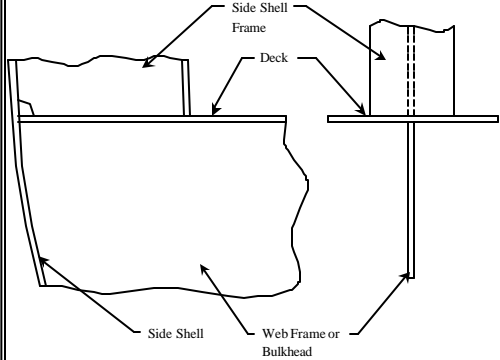
CRITICAL DETAIL:



Detail Attributes	Design Detail Rankings	
	A	B
Fatigue Performance	2	1
Ease of Inspection	1	1
Maintenance Cost	1	1
Fabrication Cost	1	2

COMMENTS:

<p>Vessel Type: War Ships Critical Area: Side Critical Detail: Connection of Side Shell Frame and Side Shell Frame Detail Description: Rolled Tee to Fabricated Tee of Similar or Differing Depths</p>	 Ship Structure Committee  FTL FLEET TECHNOLOGY LTD	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">11</h1> <p style="font-size: 8px;">Rev. 001-260199</p>												
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design - Similar Depth Tees</p> 													
<p>CRITICAL DETAIL:</p> 	<p>Detail Design - Differing Depth Tees</p> 													
<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th style="background-color: #cccccc;">Detail Attributes</th> <th style="background-color: #cccccc;">Design Detail Rankings</th> </tr> <tr> <th colspan="2" style="background-color: #cccccc;">A</th> </tr> </thead> <tbody> <tr> <td style="background-color: #cccccc;">Fatigue Performance</td> <td>1</td> </tr> <tr> <td style="background-color: #cccccc;">Ease of Inspection</td> <td>1</td> </tr> <tr> <td style="background-color: #cccccc;">Maintenance Cost</td> <td>1</td> </tr> <tr> <td style="background-color: #cccccc;">Fabrication Cost</td> <td>1</td> </tr> </tbody> </table>			Detail Attributes	Design Detail Rankings	A		Fatigue Performance	1	Ease of Inspection	1	Maintenance Cost	1	Fabrication Cost	1
Detail Attributes	Design Detail Rankings													
A														
Fatigue Performance	1													
Ease of Inspection	1													
Maintenance Cost	1													
Fabrication Cost	1													
<p>COMMENTS:</p>														

<p>Vessel Type: War Ships Critical Area: Side Critical Detail: Intersection of Side Shell Frame and Web Frame or Bulkhead Detail Description: End Connection of Side Shell Frame</p>	 	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">12</h1> <p style="font-size: 8px;">Rev. 001-020299</p>																	
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Vertical Chock</p>  <p>Detail Design B - Curved Chock</p> 																		
<p>CRITICAL DETAIL:</p>  	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2" style="background-color: #cccccc;">Detail Attributes</th> <th colspan="2" style="background-color: #cccccc;">Design Detail Rankings</th> </tr> <tr> <th style="background-color: #cccccc;">A</th> <th style="background-color: #cccccc;">B</th> </tr> </thead> <tbody> <tr> <td style="text-align: left;">Fatigue Performance</td> <td>2</td> <td>1</td> </tr> <tr> <td style="text-align: left;">Ease of Inspection</td> <td>1</td> <td>1</td> </tr> <tr> <td style="text-align: left;">Maintenance Cost</td> <td>1</td> <td>1</td> </tr> <tr> <td style="text-align: left;">Fabrication Cost</td> <td>1</td> <td>1</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings		A	B	Fatigue Performance	2	1	Ease of Inspection	1	1	Maintenance Cost	1	1	Fabrication Cost	1	1
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Fatigue Performance	2	1																	
Ease of Inspection	1	1																	
Maintenance Cost	1	1																	
Fabrication Cost	1	1																	
<p>COMMENTS:</p>																			

Vessel Type: War Ships
Critical Area: Side
Critical Detail: Intersection of Side Shell Frame and Deck Beam

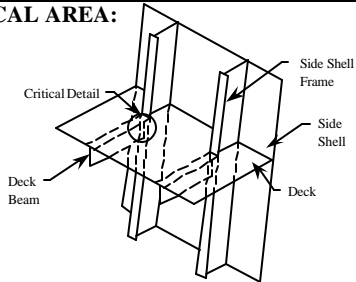


Detail
13

Rev. 001-260199

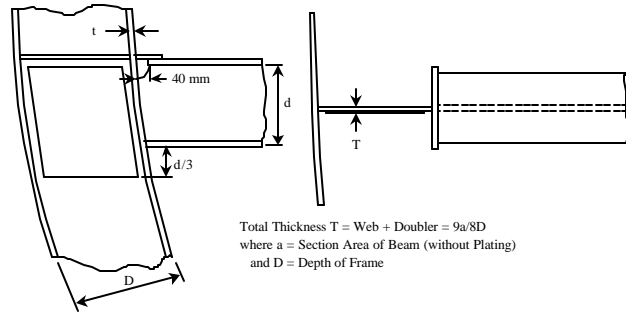
Detail Description: End Connection of Deck Beam

CRITICAL AREA:

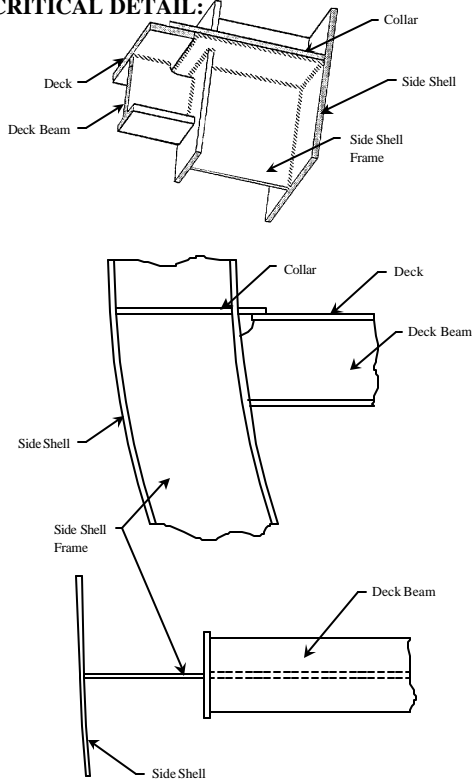


DETAIL DESIGN GOOD PRACTICE:

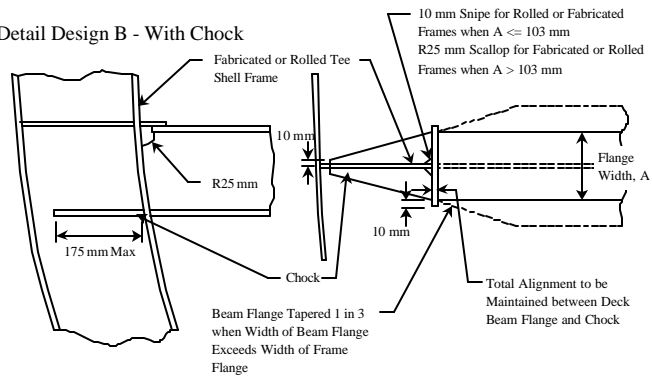
Detail Design A - With Doubler Plate
 (where $D < 250$ mm or $t < 18$ mm and $l \geq d/D \geq 0.7$)



CRITICAL DETAIL:





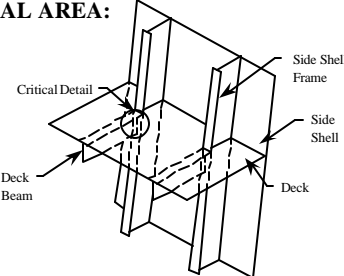
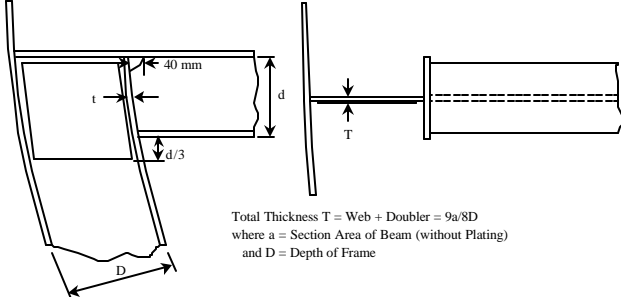
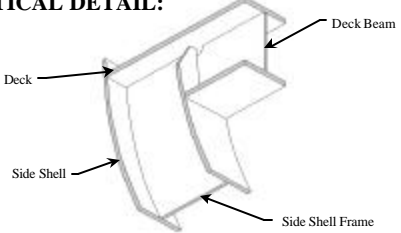
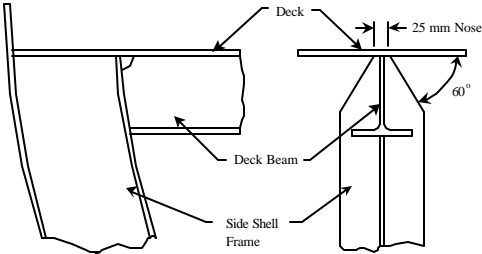
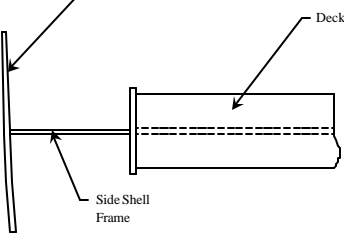
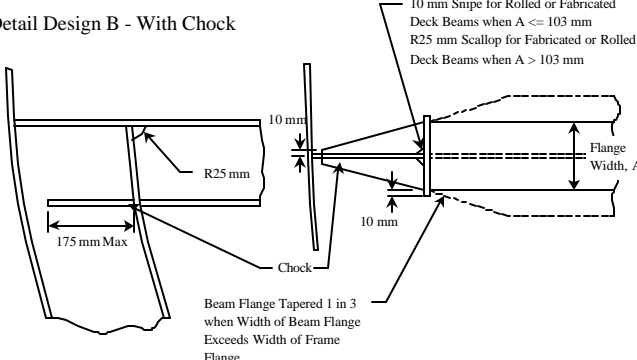
Detail Design B - With Chock



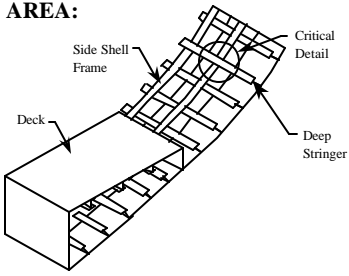
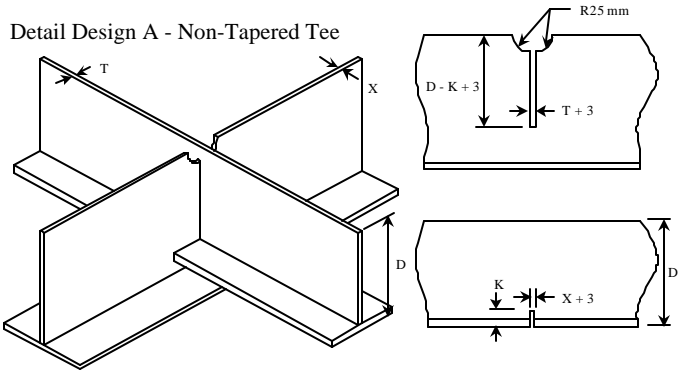
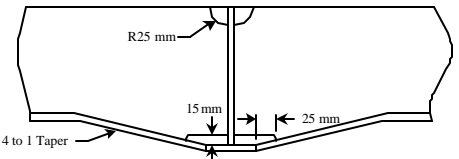
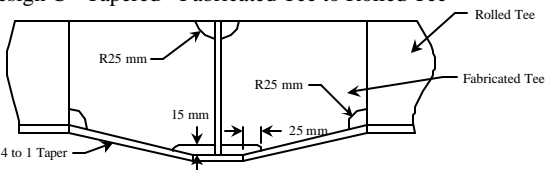
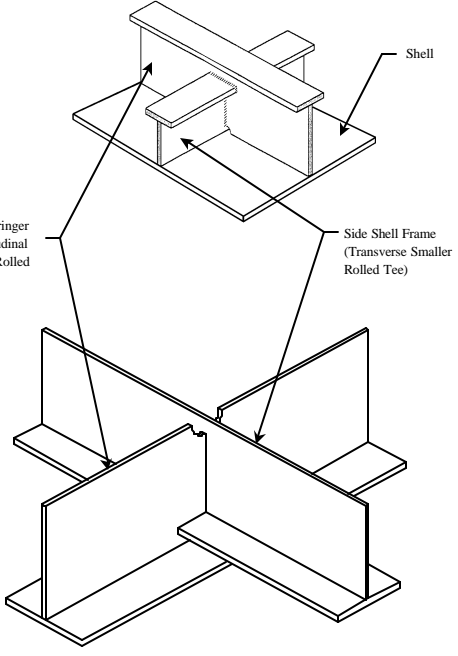




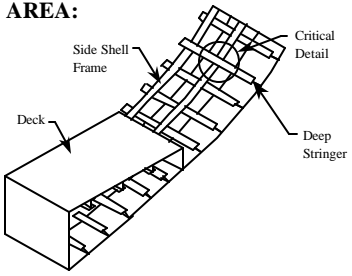
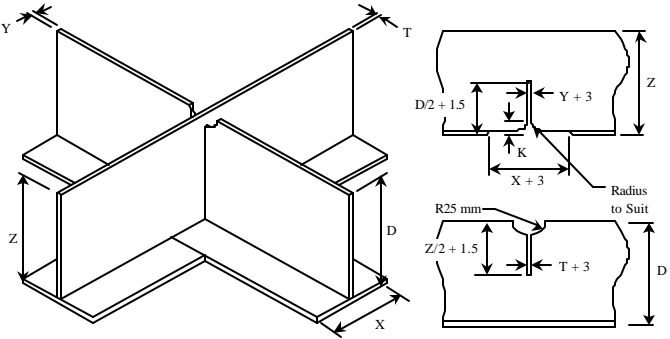
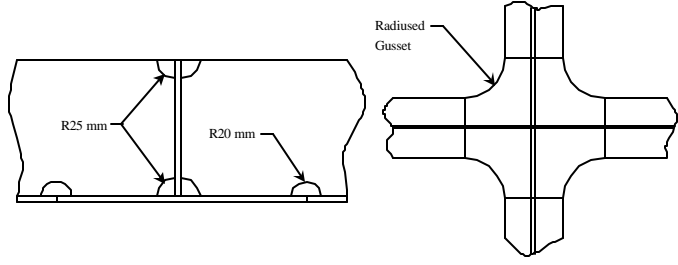
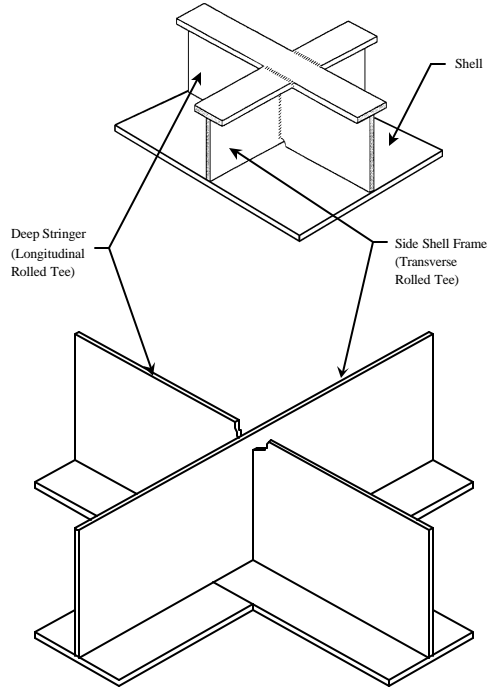
Detail Attributes	Design Detail Rankings	
	A	B
Fatigue Performance	2	1
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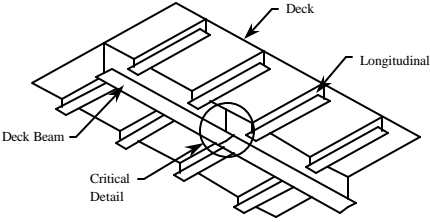
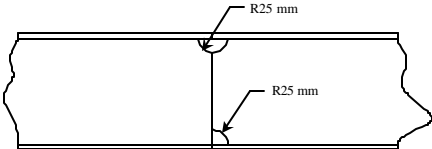
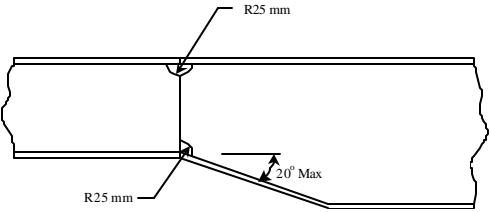
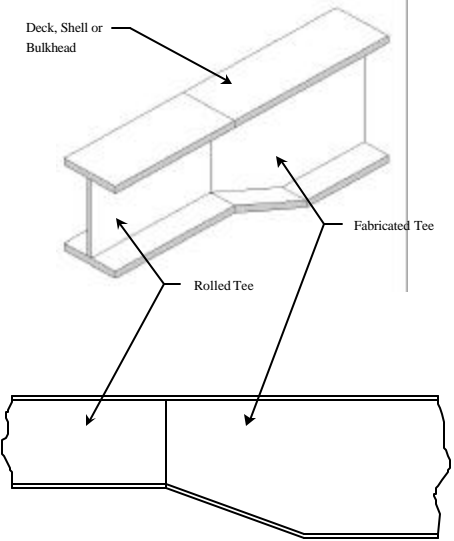
COMMENTS:



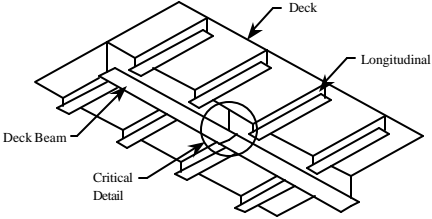
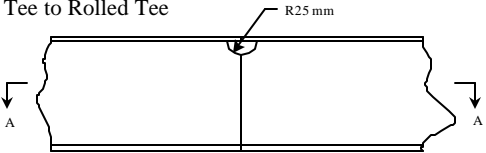
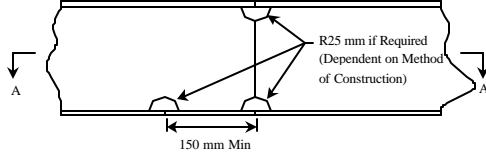
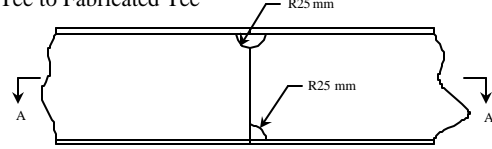
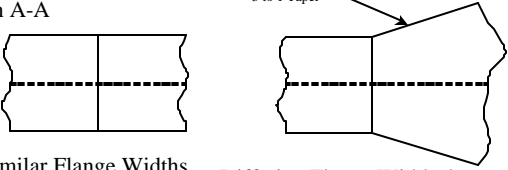
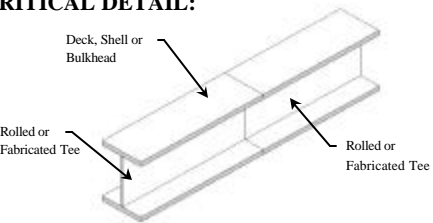

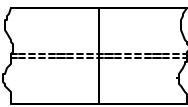
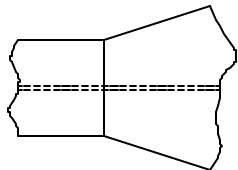
- Doubler plate not required where $d/D < 0.7$. Margin around doubler is 10 - 13 mm.



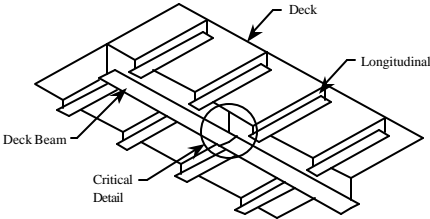
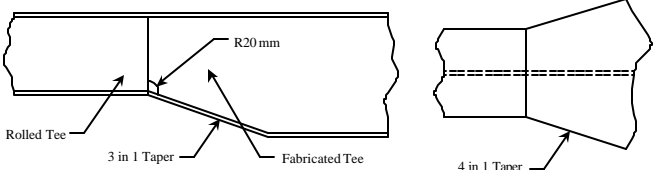
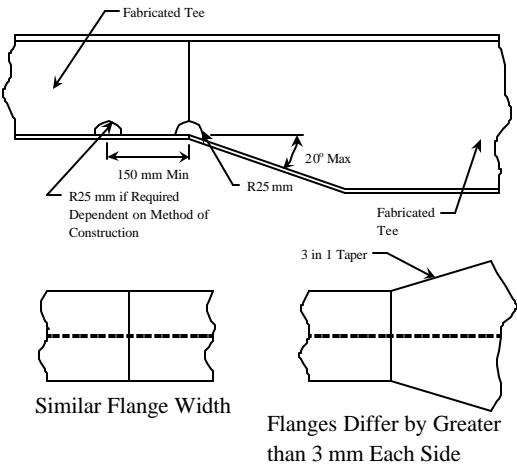
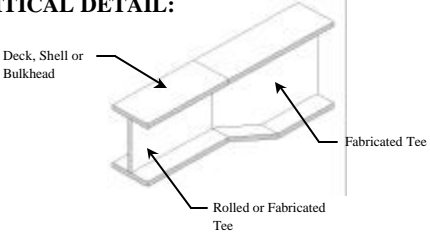
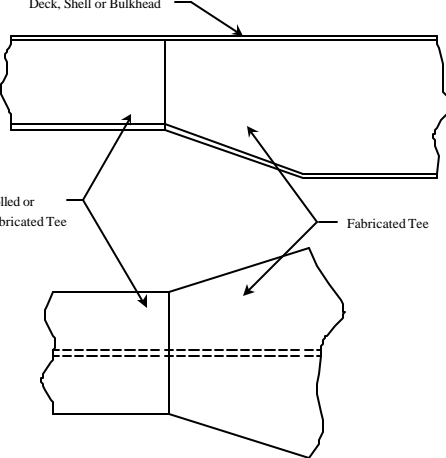
<p>Vessel Type: War Ships Critical Area: Side Critical Detail: Intersection of Side Shell Frame and Deck Beam</p> <p>Detail Description: End Connection of Side Shell Frame at Deck</p>	 	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">14</h1> <p style="font-size: 10px;">Rev. 001-260199</p>																	
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - With Doubler Plate (where $D < 250 \text{ mm}$ or $t < 18 \text{ mm}$ and $l \geq d/D \geq 0.7$)</p>  <p style="font-size: 8px;">Total Thickness $T = \text{Web} + \text{Doubler} = 9a/8D$ where $a = \text{Section Area of Beam (without Plating)}$ and $D = \text{Depth of Frame}$</p>																		
<p>CRITICAL DETAIL:</p>   	<p>Detail Design B - With Chock</p>  <p style="font-size: 8px;">10 mm Snipe for Rolled or Fabricated Deck Beams when $A \leq 103 \text{ mm}$ R25 mm Scallop for Fabricated or Rolled Deck Beams when $A > 103 \text{ mm}$</p> <p style="font-size: 8px;">Beam Flange Tapered 1 in 3 when Width of Beam Flange Exceeds Width of Frame Flange</p>																		
<p>COMMENTS:</p> <p>- Doubler plate not required where $d/D < 0.7$. Margin around doubler is 10 - 13 mm.</p>																			
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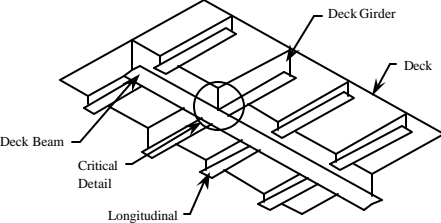
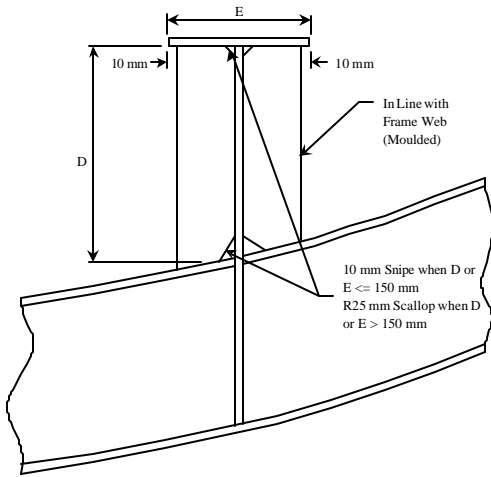
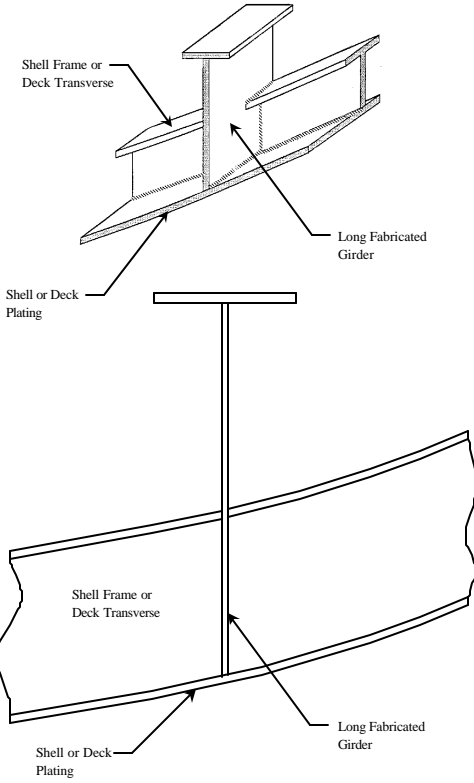
<p>Vessel Type: War Ships Critical Area: Side Critical Detail: Connection of Side Shell Frame and Deep Stringer</p> <p>Detail Description: Egg-Boxing of Rolled Tees of Differing Depths</p>	 	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">15</h1> <p style="font-size: 8px;">Rev. 001-260199</p>																							
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A - Non-Tapered Tee</p>  <p>Detail Design B - Tapered - Fabricated Tee to Fabricated Tee</p>  <p>Detail Design C - Tapered - Fabricated Tee to Rolled Tee</p> 																								
<p>CRITICAL DETAIL:</p> 	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th rowspan="2" style="background-color: #cccccc;">Detail Attributes</th> <th colspan="3" style="background-color: #cccccc;">Design Detail Rankings</th> </tr> <tr> <th style="background-color: #cccccc;">A</th> <th style="background-color: #cccccc;">B</th> <th style="background-color: #cccccc;">C</th> </tr> </thead> <tbody> <tr> <td style="background-color: #cccccc;">Fatigue Performance</td> <td>1</td> <td>1</td> <td>1</td> </tr> <tr> <td style="background-color: #cccccc;">Ease of Inspection</td> <td>1</td> <td>1</td> <td>1</td> </tr> <tr> <td style="background-color: #cccccc;">Maintenance Cost</td> <td>1</td> <td>1</td> <td>1</td> </tr> <tr> <td style="background-color: #cccccc;">Fabrication Cost</td> <td>1</td> <td>1</td> <td>1</td> </tr> </tbody> </table>		Detail Attributes	Design Detail Rankings			A	B	C	Fatigue Performance	1	1	1	Ease of Inspection	1	1	1	Maintenance Cost	1	1	1	Fabrication Cost	1	1	1
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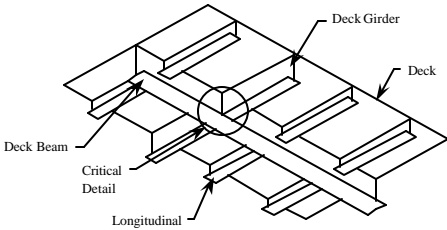
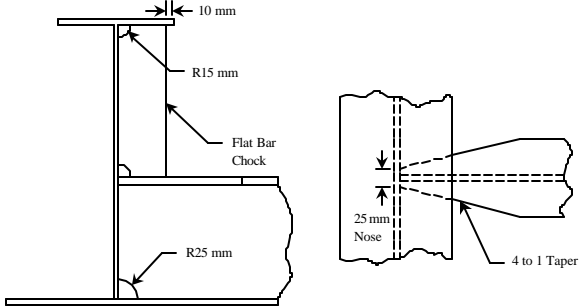
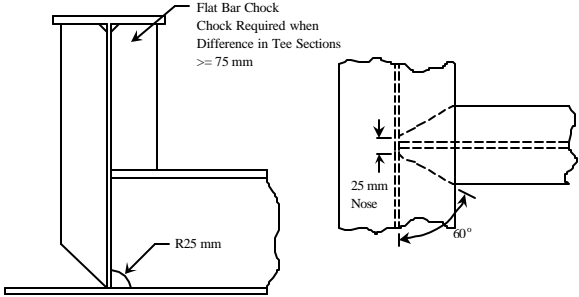
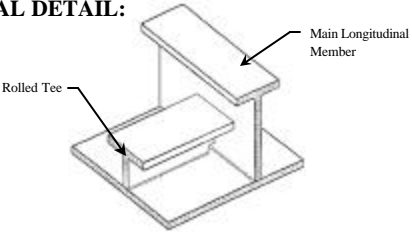
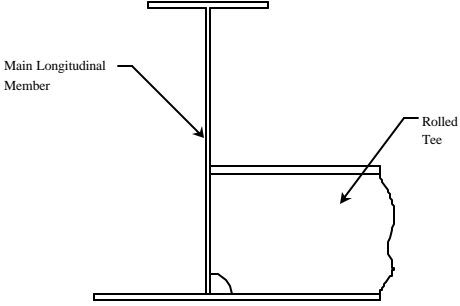
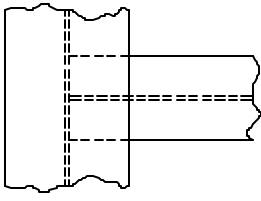
<p>Vessel Type: War Ships Critical Area: Side Critical Detail: Connection of Side Shell Frame and Deep Stringer</p> <p>Detail Description: Egg-Boxing of Rolled Tees of Similar Depths</p>	 Ship Structure Committee 	<p>Detail</p> <h1 style="font-size: 48px; margin: 0;">16</h1> <p style="font-size: 10px;">Rev. 001-260199</p>																				
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> <p>Detail Design A</p>  <p>Detail Design B - Radiused Gusset</p> 																					
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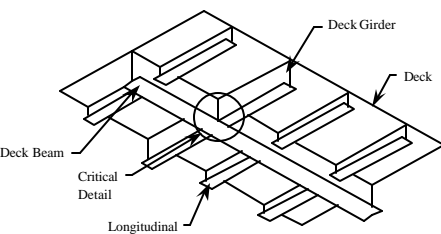
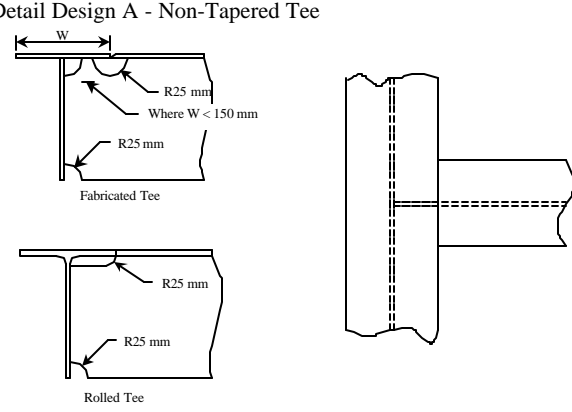
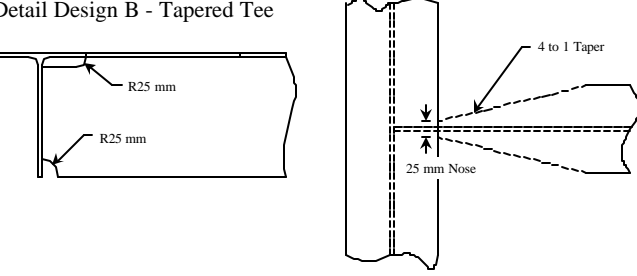
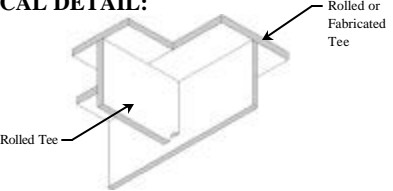
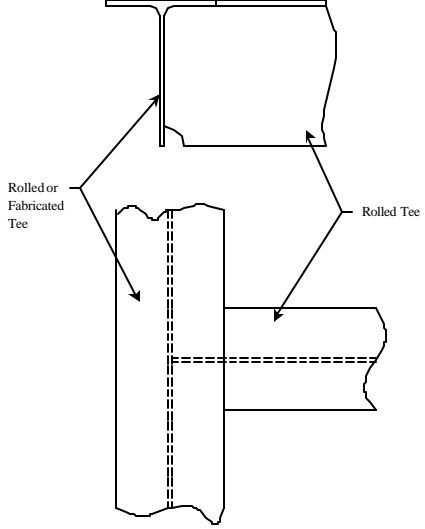
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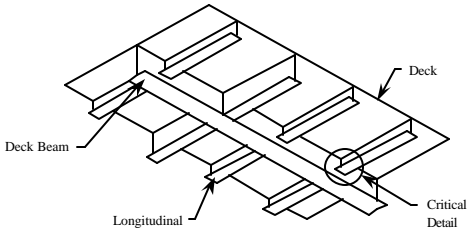
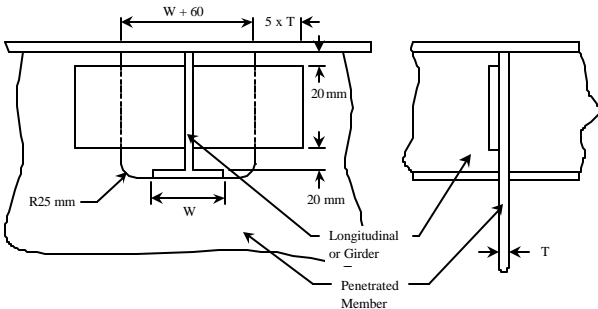
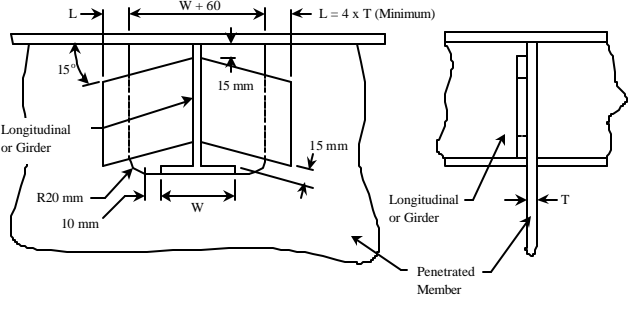
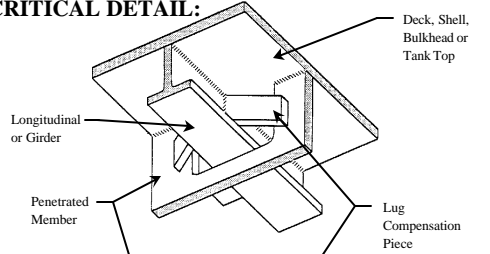
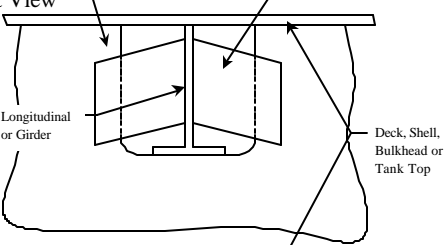
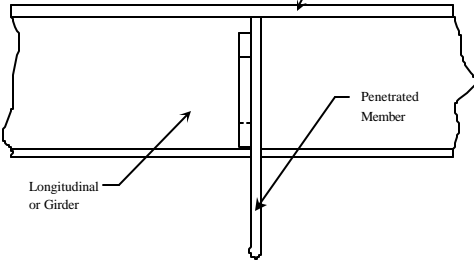
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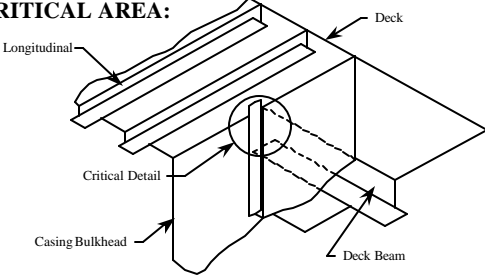
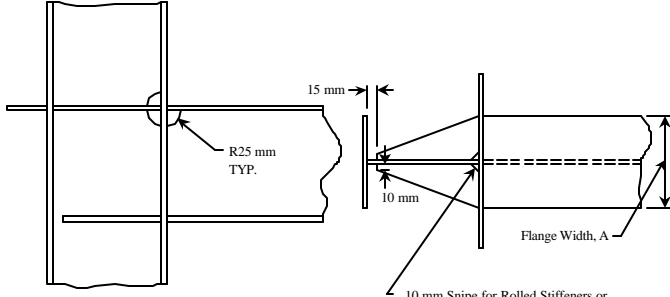
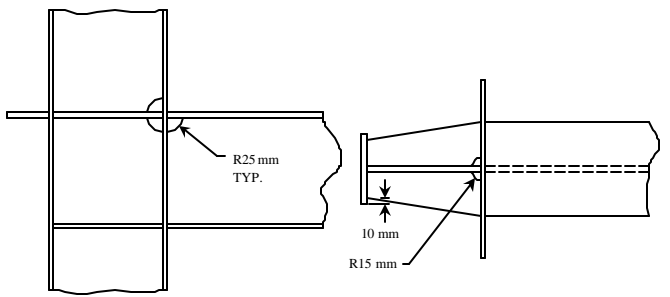
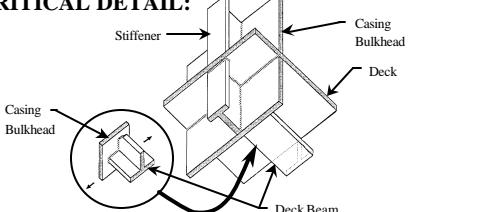
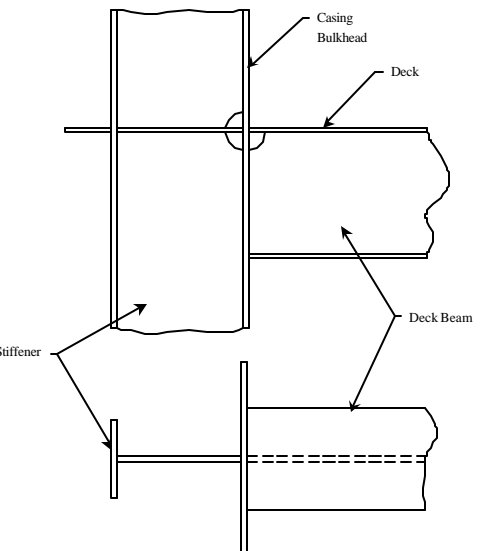
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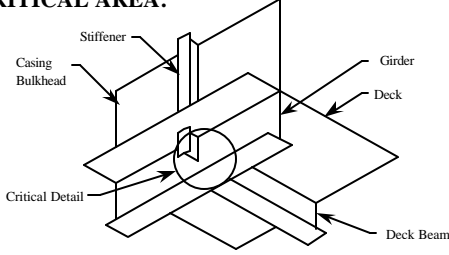
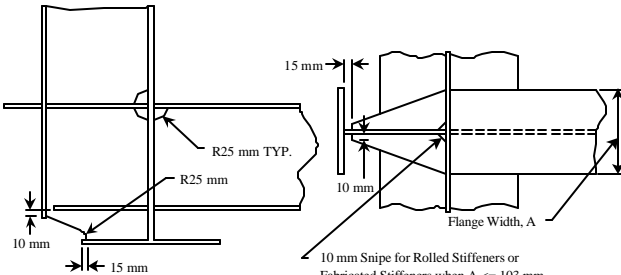
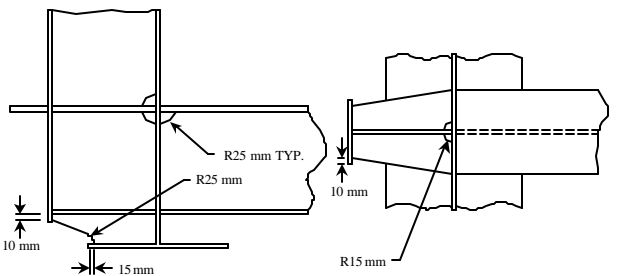
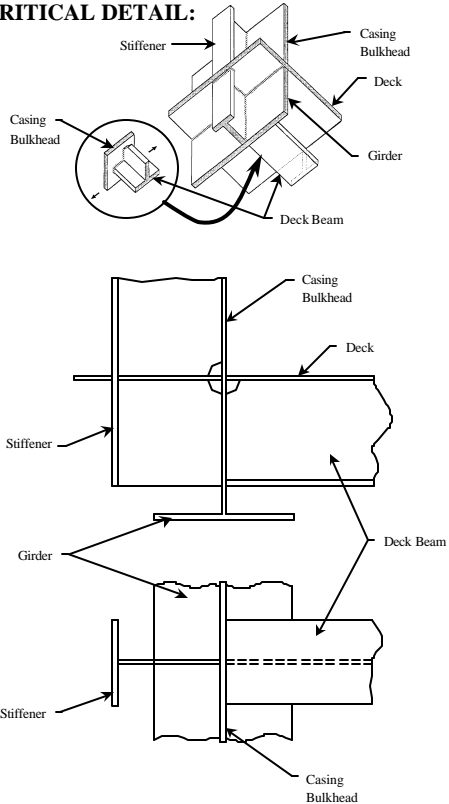
<p>Vessel Type: War Ships Critical Area: Deck Critical Detail: Connection of Deck Beam and Deck Longitudinal Girder Detail Description: Tripping Bracket Detail</p>	 Ship Structure Committee  FTL FLEET TECHNOLOGY LTD	<p>Detail</p> <h1 style="font-size: 2em; margin: 0;">20</h1> <p style="font-size: 0.8em;">Rev. 001-260199</p>												
<p>CRITICAL AREA:</p> 	<p>DETAIL DESIGN GOOD PRACTICE:</p> 													
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<p>Vessel Type: War Ships Critical Area: Deck Critical Detail: Connection of Deck Beam and Deck Longitudinal Girder Detail Description: Tee End Connection to Deeper Tee</p>	 	<p>Detail 21 <small>Rev. 001-260199</small></p>																	
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<p>Vessel Type: War Ships Critical Area: Deck Critical Detail: Connection of Deck Beam and Casing Bulkhead</p> <p>Detail Description: End Connection of Deck Beam in way of Deck Opening</p>	 	<p>Detail</p> <h1 style="font-size: 2em; margin: 0;">24</h1> <p style="font-size: 0.8em;">Rev. 001-260199</p>																	
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B.6 “GOOD” DESIGN, CONSTRUCTION AND WELDING PRACTICES TO ENHANCE FATIGUE PERFORMANCE

B.6.1 Introduction

As stated in Section B.2, structural details that are non-critical need only be subject to good design and fabrication practice to eliminate fatigue cracking. This section provides an overview of such good practice. The reader is referred to standard ship design and construction references [e.g., Ref. B.9, B.10] for further reading.

Fatigue cracks in steel ship structures are commonly initiated at weld toes. These cracks are primarily due to the presence of an initial crack-like defect, a notch or flaw that is also subject to stress concentrations due to the local weld, and the surrounding structural geometry. The presence of welding and assembly residual stresses further promotes the initiation of fatigue cracks. Accordingly, steps should be taken at both the design and construction stages to minimize these influences, such measures being beneficial in enhancing the fatigue performance of the affected welds and therefore, of the ship structure.

B.6.2 Detail Design

Because the structural designer responsible for the fatigue performance of a structure may not be directly involved in the generation of the final fabrication structural drawings, the higher level, or guidance, drawings must indicate clearly what the assumed or required fabrication tolerances are. Failure to transmit the important fatigue design data to the fabricator will greatly reduce the value of much of the process described herein.

An overall goal of good design practice is to minimize stress concentrations. Stress concentrations occur at the intersection of **primary structural members** (deck, shell and longitudinal girders) with other primary structural members, and with secondary structural members (decks, bulkheads and their stiffeners, etc.). Primary members should be arranged to ensure effective continuity of strength, and abrupt changes of depth or section should be avoided. Where members abut on both sides of a bulkhead or other members, proper alignment should be ensured. Members should have sufficient lateral stability and web stiffening and the structure should be arranged to minimize “hard-spots” and other sources of stress concentrations. Members should have sufficient end fixity, through end brackets or equivalent structure, in order to provide sufficient restraint against rotation and displacement and to provide an effective load transfer mechanism.

Secondary members are generally connected at their ends. Where a longitudinal strength member is cut at a primary support and brackets provide the strength continuity, the bracket scantlings are to be such that the section modulus and effective cross-sectional area are not less than those of the strength member.

Where **openings** are to be used, a generous **radius** should be provided. For large hatch openings, a corner radius equal to 1/24 of the hatch width, with a minimum radius of 300 mm, should be used. If the radius must be minimized in order to reduce lost space, appropriate compensation measures must be taken. Elliptical profiles are also being used more frequently for hatch corners, large

drain holes, etc., as they provide a more favourable stress flow pattern than radial cuts. For small openings, a radius of 150 mm is generally used.

Since there are always stress concentrations around large discontinuities (doors or hatchways), efforts should be made to avoid lower fatigue design category weld joints (insert plates, fillet welds, drain holes, etc.) in the vicinity of large openings.

For **openings** in the webs of stiffeners, the depth of the opening should not exceed 25% of the web depth. In addition, the opening should ideally be located such that the edges are not less than 35-40% of the web depth from the face plate. The length of the opening should not exceed the web depth or 60% of the secondary member spacing, whichever is greater, and the ends of the openings should be equidistant from the corners of cut-outs for secondary members. Cut-outs for the passage of secondary members should be designed to minimize stress concentrations. The breadth of the cut-out should be kept as small as practicable and the top edge should be rounded, or the corner radii made as large as possible. The direct connection of the penetrated web plating, or the scantlings of lugs or collars, should be sufficient for the transmission of lateral loads from the secondary member.

The exceptions to the above Guidelines are the cut-outs, or “lightening holes”, in double bottom floor plates and webs where the wide plate web with large cut-out is an improved alternative to simple stiffeners on the two adjacent structures (e.g., inner and outer shell).

Insert plates, typically used as reinforcement at the corner of a large opening, should be incorporated into the deck or shell plating. Thick insert plates should be avoided due to their restraint against weld shrinkage that leads to weld cracks. If necessary, transition strakes should be inserted to smooth the change in thickness. Doubler plates should be avoided.

Another form of **restraint and discontinuity** develops when a rigid member abruptly terminates at the midspan of a flexible plate panel. This creates a hard spot (stress concentration) in the plate panel, and it is recommended that the terminating rigid member be extended and tapered to the end of the plate panel.

For the **intersection of two planes** (i.e., longitudinal bulkhead and deck) where a primary stress is transferred from one plane to the other, long connection brackets are recommended to ensure a smooth transfer of stress between the planes.

The **toes of brackets**, etc., should not land on unstiffened plate panels and the toes should be concave or tapered.

The use of **scallops** should be avoided where possible. However, scallops are regularly used where a groove weld of a stiffener or girder is made after the members have been assembled in place. Scallops are not recommended for stiffening members, girders or bilge keels in way of completed shell or deck butts; rather it is recommended that the weld reinforcement be removed where crossed by the stiffener, girder or bilge keel.

B.6.3 Construction and Welding Practice

Sound construction and welding practice encompasses various aspects such as control of assembly and fabrication tolerances, selection of joint design, good workmanship, optimum welding sequence, etc. The welded **cruciform joint** with load carrying fillet welds is frequently required in ship fabrication. The fatigue life of such a joint can be drastically reduced by the presence of **misalignment** between the load carrying members which are welded to the continuous member of the joint.

The fit-up tolerances typically limit the maximum allowable misalignment, M_e , to $t_s/3$ (max. 5 mm), where t is the continuous plate thickness and M is measured from the centrelines of the intersecting plates (see Figure B.6.3.1). The decrease in fatigue life is proportional to M and therefore, it is highly desirable to minimize the misalignment.

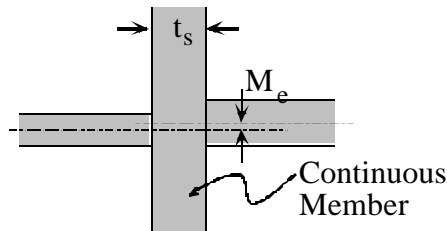


Figure B.6.3.1: Maximum Welded Cruciform Joint Misalignment

Misalignment between two groove-welded members is also detrimental to fatigue life. Codes and Standards have recognized this and generally limit the maximum misalignment to 10% of the thickness of the thinner member, but no more than 3 mm. For the same reason, where the difference in plate thickness is greater than 3 mm, the thicker plate should be tapered (not exceeding 1 in 3) or bevelled to form the welded joint. Where the difference is less than 3 mm, the transition may be achieved within the width of the weld.

Lap connections are typically not used to connect plates that may be subjected to high tensile or compressive loading. When lap connections are used, the width of the overlap is not to exceed four times nor be less than three times the thickness of the thinner plate.

Fillet welds are typically used for T-connections and should be on both sides of the abutting plate. Where the connection is highly stressed, deep penetration or full penetration welds may be required.

As far as **weld joint design** is concerned, it is well recognised that fatigue cracks are relatively easily initiated in transversely loaded partial penetration groove welds and therefore, full penetration groove welds must be specified when cyclic loading is present. Similarly, transversely loaded full penetration groove welds made from one side on to a steel backing strip have a far lower fatigue performance compared to similar welds made from both sides. However, the ad hoc use of a backing strip to address the problems associated with too large a root gap during assembly may adversely affect the fatigue strength of the welded joint.

When it comes to member **fit-up for fillet welds**, it is customary to aim for as small a gap as possible between the two members. For gaps exceeding 1.6 mm, the required fillet leg length is increased to compensate for the increased gap. At the same time, there is some evidence that a gap of 1.5 to 2 mm influences the weld residual stresses in such a way that fatigue performance is enhanced, provided that the joint restraint is low, i.e., the attaching member is free to move in response to weld shrinkage [Ref. B.11]. Under such circumstances, a 1.5 mm gap is ideal.

Weld shape is another important factor influencing the fatigue performance of the welded joint. In fillet welds, a slightly convex or slightly concave weld profile is desired. Excessive convexity is detrimental from a fatigue point of view and is limited in fabrication codes.

In the case of **butt joints**, weld **reinforcement** and **undercut** are important considerations. Excessive weld reinforcement increases the stress concentration at the weld toe and therefore the maximum allowable reinforcement in fabrication codes is typically 3.2 mm with the weld smoothly blending into the base metal. Undercut, which is defined as a groove or notch formed in the base metal adjacent to the weld toe, can seriously reduce the fatigue life of the welded joint. Formed as a result of incorrect selection of welding parameters or welding consumables, or lack of welder skill, the fabrication codes generally specify smaller allowable undercut for dynamically loaded structures. For example, Reference B.12 allows a maximum undercut of 0.25 mm in critical members when the weld is transverse to the applied stress. Otherwise, the maximum allowed undercut is 0.8 mm (1/32"). In comparison, for statically loaded structures, the undercut is generally limited to a depth of 1 mm, and for short lengths, the allowable undercut may be as high as 1.6 mm.

Longitudinal welds generally have good fatigue performance. However, the presence of **starts and stops** which are not ground out, or **breaks in the backing strip** (unwelded joint in backing strip) in the case of welds made from one side, significantly reduce the fatigue strength of the weld.

Where stiffening members are continuously **fillet welded across completely finished butt or seam welds**, the butt welds are to be made flush. Similarly for the butt welds in the webs of stiffening members, the butt weld is to be completed and generally made flush with the stiffening member before the fillet weld is made.

Other **welding flaws** such as hydrogen-induced cold cracks in the heat affected zone, solidification cracks in the weld metal, incomplete root penetration in welds made from one side but without a backing strip, etc., are other likely sites for fatigue crack initiation. Embedded flaws such as slag inclusions and porosity are relatively benign for fatigue unless present in excessive amounts.

B.6.4 Residual Stresses

Residual stresses in steel ships can be considered to be composed of two components: **short range** weld thermal residual stresses and **long range** assembly and restraint stresses. The short range weld thermal residual stresses result in high levels of tensile stresses that approach the yield strength of the base metal. These residual stresses, which are always present and are difficult to control or modify, can reduce the fatigue strength of the welded joint. However, through careful attention to fit-up tolerances and welding sequences, the long range assembly and restraint stresses can be reduced. If the members to be welded need to be jacked or bent in order to achieve the correct alignment, bending stresses will be introduced into the structure and these can contribute to fatigue crack initiation. Once cracking is initiated, these stresses tend to relax. In terms of welding sequence, it should be so arranged that as welding progresses, the members retain as much freedom to move as possible.

Reference B.9 provides recommendations for welding sequence for butts and seams, with and without internal framing, and for large sub-assembled plate panels. Another approach is to commence welding in the deck and bottom regions and progress towards the neutral axis of the ship. The restraint stresses will then be high near the neutral axis where the applied stresses are lower.

B.6.5 Weld Toe Dressing Treatments

Finally, it should be added, that in certain situations it may not be possible to avoid relatively inferior fatigue design details, or the structural detail may involve high stress concentrations. In these situations, fatigue crack initiation is deemed unavoidable at the design stage. In such circumstances, one could consider **weld toe dressing treatments** such as hammer or shot peening, toe grinding, TIG remelting, etc. Further information on the benefits and costs of such techniques are covered in Reference B.13.

B.8 REFERENCES

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PART C - FATIGUE STRENGTH ASSESSMENT

C.1 INTRODUCTION

C.1.1 Background

The analysis procedure involved in the estimation of the fatigue life or probability of fatigue failure over the design life of a ship may be described in terms of the following **four basic steps**:

- (1) **Establish the long-term statistical distribution of cyclic loads on a ship over its design life;**
- (2) **Determine the corresponding long-term statistical distribution of the notch stress range ($D_{S_{notch}}$) at the anticipated crack initiation site;**
- (3) **Calculate the fatigue damage inflicted by these loading cycles;**
- (4) **Perform a reliability analysis to predict the probability of fatigue based failure.**

Classification Societies, amongst others, have developed a variety of approaches to assess the fatigue life of ship structural details by expanding on this basic four step process. Discussion of the fatigue design approaches adopted by the major class societies is presented in Section C.2.3 and Appendix A.

C.1.2 Objective

The assessment procedures set out in this Guide are intended to be used by engineers and naval architects in ship structural design to reduce the likelihood of premature fatigue failure and to assist in planning through-life inspection strategies by identifying the relative fatigue resistance of different elements of the ship.

The objective of Part C of the Guide - *Fatigue Strength Assessment* - is to describe the procedures for predicting the fatigue life of ship structural details that are subjected primarily to wave-induced loads.

C.1.3 Scope

While the four basic steps in fatigue analysis are easily understood, the analytical techniques required can be relatively complex. Two alternative fatigue assessment procedures, categorized as Level 2 and Level 3 (a simplified, and a direct spectral approach respectively), are outlined in a step-by-step fashion in Sections C.2.1 and C.2.2, respectively, and a flow chart for these procedures is presented in Figure C.2.1.

The intended applications and limitations of these procedures are discussed in Section C.2.3, while detailed guidance for the calculation of input loads, structural response and resulting fatigue damage are given in Sections C.3 to C.6. A reliability procedure for estimating the probability of failure associated with predicted fatigue lives is given Section C.7.

The processes presented are exhaustive, but they are the most practical available at the time of the production of this manual. It should be recognized that some of the procedures remain at an early stage of development.

C.2 FATIGUE ANALYSIS PROCEDURES

C.2.1 Overview

The Level 2 and Level 3 approaches are presented in parallel in Sections C.3 through C.6, each section dealing with one of the four basic steps in the analytical process. While the two Levels contain the same four basic steps as stated in C.1.1, they differ in the details of their execution. To illustrate the differences in the two parallel procedures, Figure C.2.2.1 expands the four basic steps into eight more detailed blocks. In each block of the fatigue analysis process flow chart (Figure C.2.2.1) reference is made to the section within this report (Part C) in which a technical description of the procedure is presented.

The analytical process shown in Figure C.2.2.1 includes two cross-over points between the simplified and spectral analysis approaches. These cross-over points indicate, for instance, that the user may use the full analysis approach to determine the long term statistical distribution of wave loads and cross over to use the simplified approach to determine the corresponding statistical distribution of notch stress ranges. Although the steps in the two approaches use data and develop results of a similar form, the level of conservatism, and possibly accuracy, may not be consistent.

C.2.2 Commentary

C.2.2.1 Intended Applications

The Level 2 procedure is best suited to the assessment of standard structural details in ships, whereas the Level 3 approach is intended for: (i) assessing structural details outside the calibration-base of the simplified procedure (e.g., novel structural configurations, unusual wave environments), and, (ii) confirming and refining the fatigue performance of critical structural details at the end of the detail design stage.

In any design where fatigue is expected to be a significant issue, the Level 1 approach of using good fatigue-resistant details should be used as a matter of course. In the event that a combination of Level 1, 2, and/or 3 approaches does not provide an adequate expected fatigue life in the detail design stage, then a more general redesign of the structure may be required to lower the overall (or local nominal) stress level.

C.2.2.2 Design Life

The design life for structural details in damage tolerant structure should be set to the service life of the ship (typically 20 years).

If damage tolerance cannot be achieved or is impractical for the structure surrounding a structural detail, the design life of the detail should be many times the service life of the ship to ensure that fatigue cracks do not initiate in service. In other words, use a safe life design philosophy.

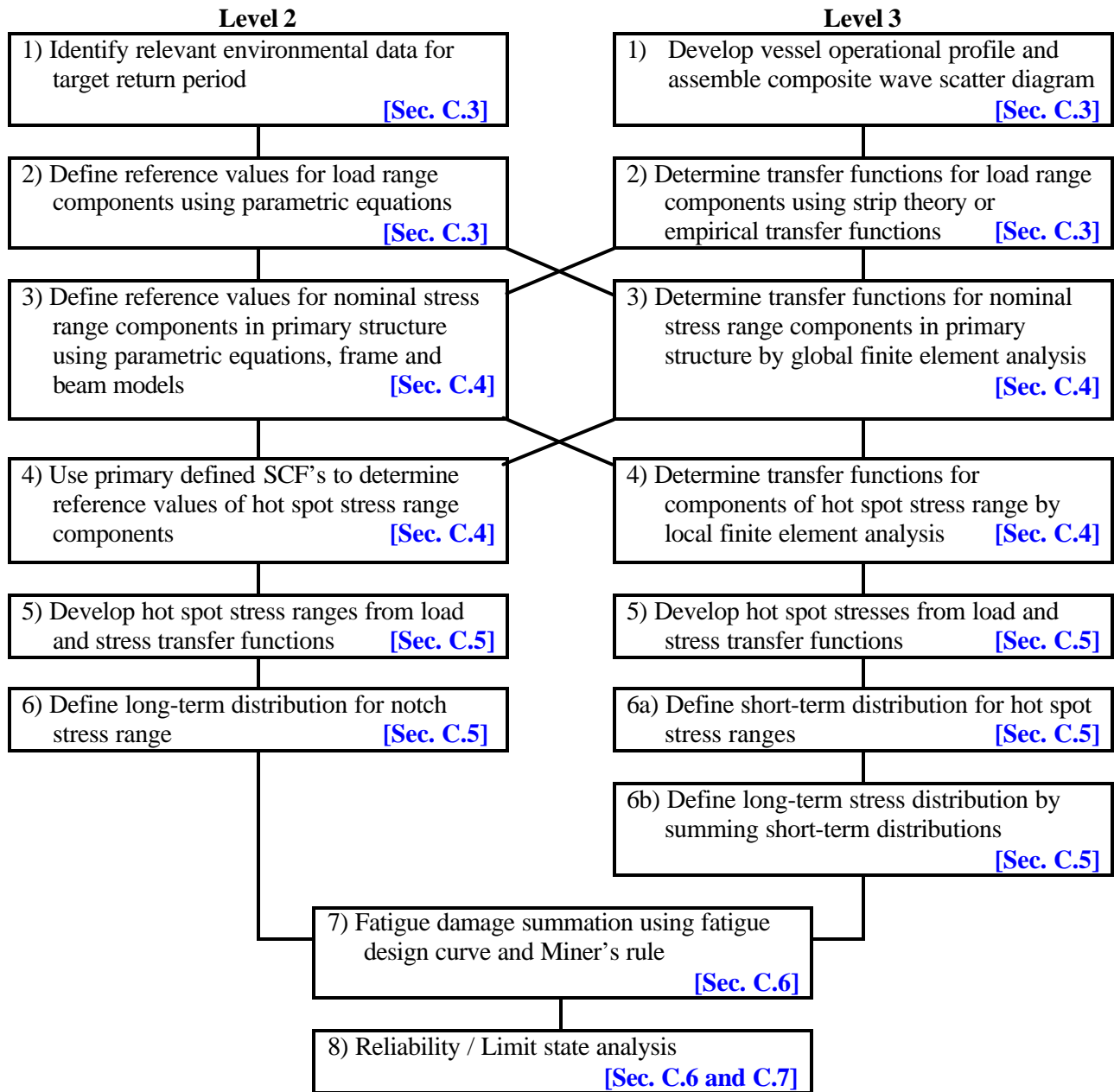


Figure C.2.2.1: Fatigue Strength Analysis Flow Chart: Levels 2 and 3 Methods

The safety margins in the fatigue design curves presented in Section C.5 are consistent with the safety margins in fatigue design curves for welded joints in bridges and offshore structures. Experience has demonstrated that these margins are generally sufficient to ensure a finite but acceptably low probability of failure during the service lives of bridges and offshore structures, where failure is defined as the initiation of a through-thickness crack several inches long. Fatigue cracks that do initiate will tend to do so towards the end of the service life, and the damage tolerance of the surrounding structure will ensure that these cracks can be detected and repaired before they pose a threat to structural integrity.

In the case of ships, the greater uncertainty in the estimation of loads is countered by the greater redundancy of ship structures. Therefore, the safety margins in the fatigue design curves presented in Section C.5 should be adequate for structural details in damage tolerant ship structure. The acceptable probability of failure should be agreed upon between the ship owner and ship designer, and should be based on previous experience whenever possible.

C.2.2.3 Non-linearities

Neither the Level 2 procedure nor the Level 3 procedure presented herein account for non-linearities in the load and structural responses of a ship. Some of the parametric equations presented in Sections C.3 and C.4 for predicting reference loads and stresses are based on Classification Society rules, while an implicit assumption in the spectral analysis is that the structural and load responses of the ship are linear. Certain non-linearities can be neglected in a fatigue damage assessment. For example, the well-known non-linear effect of hull shape on hull girder bending stresses occurring especially with large wave heights and pronounced bow flare has only a small effect on fatigue life. The major part of fatigue damage is caused by the large number of smaller stress cycles for which the linear assumption is adequate. Other non-linearities, however, can have a significant influence on fatigue damage. For example, the time history of the side shell pressure at a location near the still-water line that is temporarily emerged due to ship motions can be highly non-linear because negative pressure or suction cannot occur. Linearization and time domain techniques have to be used to account for these non-linearities.

C.2.3 Design Codes and Criteria

In the course of a commercial design it will be necessary to obtain Classification Society approval and therefore it is prudent to follow the fatigue design procedures specified by the specific Society, [e.g., References C.1 to C.3], or to have the procedure used approved by the appropriate Society. The procedures for fatigue design, which are currently promulgated by the Classification Societies, vary somewhat in scope and applicability. In order to assist in determining the relative merits of these codes a discussion and detailed comparison of their treatment of fatigue is provided in Appendix A.

C.3 LOAD ANALYSIS

C.3.1 Determination of Environmental Loads

C.3.1.1 Introduction

This section provides guidelines for determining the fatigue loading that a ship structure will experience throughout its lifetime or a particular period of interest. It is assumed, as elsewhere herein, that this Guide is being used by a qualified and practicing ship designer or structural engineer, and therefore this Guide does not provide comprehensive direction on how to determine the loads on a ship structure. Rather it provides guidance on how the loading is applied to the fatigue analysis problem.

The ship designer is assumed to be already working with loads to determine the vessel's design strength, and size the structural elements and scantlings, and with knowledge of the operational area and possibly the operating profile for the vessel. This Guide will show how the information needs to be formulated and used for fatigue assessment. The fundamental difference is that fatigue design has to be based on an explicitly statistical representation of the complete loading environment, while much of the rest of the design only considers single extreme values.

C.3.1.2 Definition of Loads

For fatigue life analysis, loads must be defined in terms that allow derivation of *stress ranges*. This may require definition of a load component in hog and sag, or as maximum compressive and tensile load, so that a *range* can be defined. As for overall ship design, loads can be categorized as global or local. Reviewing these loads, the important global loads are:

- a) stillwater loads;
- b) wave loads;
 - low frequency steady-state, response largely quasi-static
 - high frequency transient (wave impact or slamming), dynamic.
 - high frequency steady-state (springing)

while the main local loads are:

- c) hydrostatic pressure loads ;
- d) pressure loads due to waves, as in b);
- e) inertia loads from cargo or fluids (sloshing) induced by ship motion;
- f) functional loads, from machinery and deck equipment;
- g) ice loads.

Any loads that result in significant variation in stress levels are potentially relevant for fatigue assessment. Fatigue damage is seen as cumulative, and thus all loading sources potentially contribute. However, since a vessel will typically be exposed to between 3×10^7 and 1×10^8 wave cycles in its life, unless another load source compares (within an order of magnitude) in either magnitude or number of cycles with the wave loading at a particular location, it is unlikely to affect the cumulative load history. Thus, sources such as machinery vibration loads (unless locally relevant) or ice loads may be ignored. Ref. C.4 provides additional discussion on this subject.

The relative significance of each type of load depends, among other things, upon the ship type, the payload, structural configuration and location of the structure. Tables A.3.1 to A.3.5 (pages A.3-A.4) provided earlier, give guidance in identifying the important loads for different ship and detail types. Further discussion of the treatment of some of the most important loads is given below.

(a) *Stillwater Loads*

Stillwater and hydrostatic pressure loads are not a direct factor in fatigue design as they typically only vary with loading conditions, and thus do not occur with sufficient frequency to cause significant fatigue damage. As fatigue damage accumulation is reduced when part or all of a stress cycle is in compression, it can be argued that the stillwater condition contributes in reducing or increasing the risk of fatigue. However, in general, stillwater (plus wave) compressive loads will not be high enough to reverse the generally tensile residual stresses at a weld. Thus, the assumption that wave induced bending is fully reversing about a stillwater reference is generally valid.

In cases where the stillwater hull girder load is significant, such that parts of the hull do remain in compression throughout, special techniques to treat this offset may be justified. Ref. C.5 provides a discussion of the impact of mean stress on fatigue.

(b) *Wave Loads*

The dominant load on most ships arises from the response to waves. The primary mechanism for wave loading is through hull girder bending. The loading is cyclic with periods of the order of several seconds; the response is essentially quasi-static.

In severe sea states, dynamic phenomena such as slamming may occur which result in transient impact loads. The response to this type of load is characterized by frequencies that are considerably higher than those associated with normal wave loading. As the response is transient, analysis of these loads is difficult using standard spectral techniques. However, it is generally assumed that the number of slams in the ship's lifetime will be small enough to neglect in fatigue analysis.

The phenomenon of “springing” in which the hull girder responds in flexure on a continuous basis, at frequencies at or close to the natural frequency(ies) of the hull, obviously have an impact on fatigue. This phenomenon is seen on occasions in, for example, Lakers [Ref. C.6] but is much less common than transient (slamming) effects.

(c) *Local External and Internal Pressure Loads*

Waves may also impose significant loads on local structural elements. The primary examples are the variation in static pressure on the side shell due to ship motions (roll, pitch, heave) and dynamic pressure loads near the waterline due to wave impact.

Internally, ship motions result in inertial (acceleration) forces on liquid or solid cargoes and deck equipment. In addition, “sloshing” of tank liquids can occur under specific conditions of tank level, fluid density, and vessel response. “Sloshing” loads can be severe, resulting in damage. However, they are normally neglected in fatigue design.

C.3.1.3 Definition of Operational Profile

The starting point for the fatigue analysis, (as with ship structural design overall) is the operational profile for the vessel. This will determine the wave climate that will be experienced. The definition of the operational profile may be as simple as stating a general area of operation (e.g., North Atlantic) and an endurance or service speed. At the other extreme, a full operational profile may state how much time the vessel will spend in various areas of the world, and at what times in the year, as well as the distribution of its speed and headings to permit the exposure to wave data to be determined,.

C.3.1.3.1 Level 2 – Method

Since the Level 2 methods for calculation of loads often rely on prescriptive parametric equations derived for specific vessel types, the amount of information required for defining the operational profile is minimal.

Since the basic approach in the Level 2 method is to develop a reference load (e.g., a design value) for use in defining the long-term statistical distribution (a Weibull representation being the “standard” approach), the information required on operational profile is limited to that required to define the parameters used to develop the reference loads. This *may* include all or some of the following:

- Ship loading conditions, e.g., loaded departure, loaded arrival, ballast departure, ballast arrival. This data may be used directly, and if not will be used to define parameters such as draft and displacement for use in the reference equations.
- Speed - Speed is not typically used as an explicit parameter, but is used to identify high(er) speed vessels (e.g., container ships), where allowances for dynamic effects such as keel or flare slamming should be considered.
- Area of operation - while a specific area of operation is not typically cited, if the anticipated operations are inshore or sheltered waters, then a reduction factor may be applicable in the formula for the reference load. There are explicit examples, such as the rules for Great Lakes vessels, where reductions in strength requirements are associated with the geographic area of operations.

There are implicit assumptions in the prescriptive rules which the designer should be aware of. For example, in developing extreme loads for some local structure (for example midbody framing), primary loads may be based on head seas for vertical bending moment, whereas rules for side shell pressure will have been developed assuming a beam sea, and a prescribed minimum roll angle [C.7]. Combining the loads by simple or weighted summation may not always justify the same spectral representation to be used. However, in practice the assumption that this can be done is reasonable, and is acceptable for the Level 2 method

Another such assumption relates to the implied wave conditions used to develop the design loads. This is discussed further in Section C.3.1.4.1.

C.3.1.3.2 Level 3 – Method

A schematic representation of the Level 3 method is provided in Figure C.3.1.3.1, which is taken from Ref. C.8.

In the Level 3 - Direct Calculation - Method, the loads are determined from a detailed knowledge of the ship's operational profile. In using direct methods for calculating extreme loads, considerable simplifications of the operational profile are usually accepted. For fatigue calculations, it may be necessary to examine the data in more detail to ensure a reasonably accurate representation of spectrum shape.

The operational profile information required for a detailed calculation includes:

- The projected route of the vessel described in terms of areas of operation and the % time spent in these areas;
- Vessel loading conditions or missions and relative time spent in each mode; (loading conditions are appropriate for commercial vessels, while the mission may be more appropriate for military or patrol vessels);
- Distribution of time spent at each heading relative to the predominant sea direction;
- Vessel average speed ranges and relative amount of time spent at each speed in a particular sea state or wave height.

This data can then be combined with a statistical representation of the wave climate for each area to provide a complete picture of the vessel's "sea operational profile", as described in the next section.

In order to make the calculations feasible, each of the parameters is discretized in some manner. For example, the route can be divided into Marsden Zones [Ref. C.9] (or zones of latitude and longitude transited by the vessel) and the time spent in these zones. Loading can be treated in terms of standard conditions. Relative heading can be simplified into head, bow, beam, quartering, and following seas; and speed can be treated as sets of speed ranges.

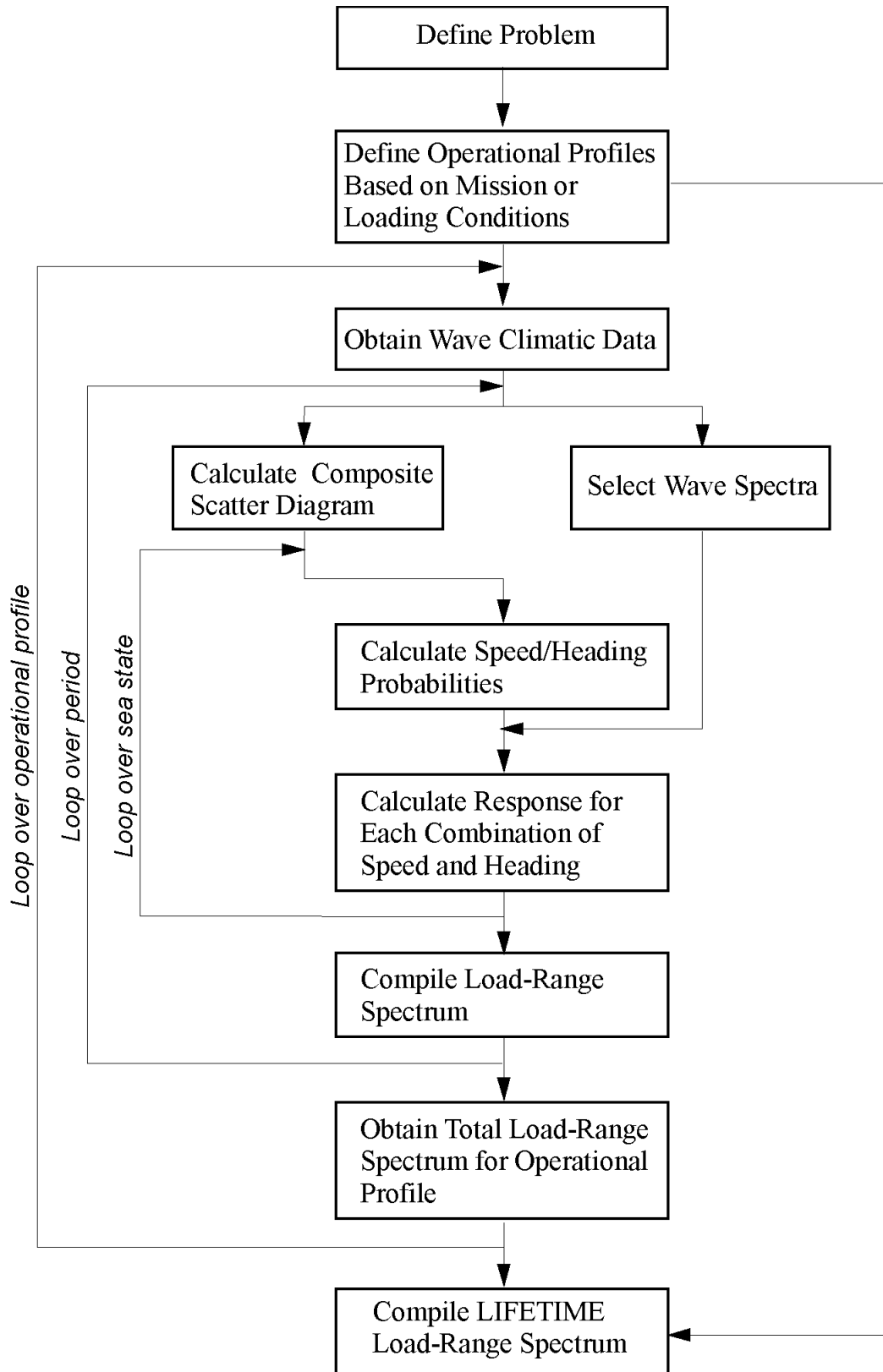


Figure C.3.1.3.1: Schematic of Level 3 - Direct Calculation - Method

When a new design will follow the same operational profile as an existing ship, the existing ship's operations may be studied and characterized from operational logs, [e.g., Ref. C.8]. For new designs, operational profiles can be generated from the operators' plans. The level of discretization of operational and/or environmental data should correspond to the certainty in the operational profile information.

The process of developing a detailed operational profile requires the development of input joint probability diagrams, including ship speed versus Sea State (or wave height) (Table C.3.1.3.1) and then ship relative heading versus Sea State (or wave height) (Table C.3.1.3.2). These are obtained either from historical data (as in the examples) or perhaps from operating directions for the vessels (particularly speed in given sea states).

When an operational profile is developed in the absence of historical data, speed, sea state and heading are often assumed to be independent quantities. This may not always be the case, as in severe sea states, the practice is to reduce speed and to orient the ship in preferred directions. However, since the bulk of fatigue damage arises from the exposure to moderate conditions, and because the amount of time spent in these severe sea states is not as significant as that spent in more moderate conditions, the assumption of independence is reasonable, and avoids extremely complex computations that are not justified. When the profile is developed from existing ships' logs, it will obviously reflect current practice, which may or may not be modified by other features of the new design.

The impact of on-board weather routing and forecasting systems in reducing the exposure of the vessel to extreme conditions, may also "skew" the observed or theoretical operational profile. Such measures cannot be taken into account in the design process at present. For a more detailed discussion of these issues see Ref. C.8.

The next data set required is the distribution of time spent in each geographical area (Table C.3.1.3.3). In order to construct the lifetime operational profile, time spent in port should also be included. A ship which spends 50% of its time alongside will obviously see fewer wave encounters per year than one which is more or less continuously at sea.

These three distributions are combined with the wave data into a table of joint probabilities of speed, heading, wave height, and wave period for the specific profile. The process for producing this final joint distribution is described in the following section.

**Table C.3.1.3.1: Sample Operational Profile
Joint Probability of Speed and Wave Height**

SPEED (knots)	Significant Wave Height (m)							SUM
	1	2	3	4	5	6	7	
0-6	0.0028	0.0000	0.0009	0.0000	0.0000	0.0010	0.0007	0.0054
6-10	0.0056	0.0031	0.0033	0.0082	0.0086	0.0261	0.0010	0.0559
10-14	0.0129	0.0219	0.0503	0.0378	0.0449	0.0896	0.0019	0.2593
14-18	0.0900	0.1253	0.1322	0.1007	0.1338	0.0968	0.0005	0.6793
SUM	0.1113	0.1503	0.1866	0.1467	0.1873	0.2135	0.0041	1.0000

**Table C.3.1.3.2: Sample Operational Profile:
Joint Probability of Heading and Wave Height**

Heading	Significant Wave Height (m)							SUM
	1	2	3	4	5	6	7	
Head Seas	0.0084	0.0114	0.0167	0.0128	0.0160	0.0221	0.0005	0.0879
Strbd. Bow	0.0234	0.0326	0.0380	0.0295	0.0383	0.0375	0.0005	0.1998
Strbd. Beam	0.0283	0.0375	0.0439	0.0355	0.0454	0.0497	0.0009	0.2413
Strbd.Quart.	0.0421	0.0586	0.0729	0.0565	0.0724	0.0800	0.0013	0.3840
Following	0.0098	0.0136	0.0164	0.0129	0.0165	0.0176	0.0003	0.0871
SUM	0.1120	0.1537	0.1880	0.1472	0.1886	0.2070	0.0035	1.0000

Table C.3.1.3.3: Geographical Distribution of Time

Marsden Zone	Total %
15	5%
16	30%
17	10%
23	9%
24	28%
25	18%
SUM	100

C.3.1.4 Definition of Wave Climate

C.3.1.4.1 Level 2 – Method

In general, the wave climate is not explicitly defined for the Level 2 approach, rather a “design wave” condition is either defined or implied by the form of the load equation. The probability of the design wave occurring in the lifetime of the ship is inherent in the various load equations specified by the Classification Societies, and as agreed by IACS, and is understood to be 10^{-8} encounters per 20-year life.

Modifications may be required to the load formulae to correct for ship size to ensure equal probability of occurrence. In the case of larger vessels, linear extrapolation of the prescriptive formula would result in an unrealistically large wave height, and thus upper bounds have been introduced to the formulae based on vessel dimensions.

Most major Classification Societies have replaced prescriptive rules by direct calculations for large vessels (e.g., $L > 190\text{m}$). This requires the development of at least part of the data described here for use with the Level 3 approach. In the case of smaller vessels, including many naval ships, the concern with prescriptive rules, again most typically for longitudinal strength, is that the resulting “design” load will occur at a higher frequency of occurrence than is consistent with a spectral analysis. Whenever direct calculations of extreme loads are made, care should be taken to ensure that the underlying assumptions are understood and properly accounted for in the fatigue analysis

C.3.1.4.2 Level 3 - Method

C.3.1.4.2(a) Sources of Wave Climate Data

The wave climate experienced by ships varies considerably depending on the area of operation. Wave data is available for most parts of the world including oceans and large bodies of water such as the Great Lakes. Perhaps the most comprehensive compilation of wave data is published by British Maritime Technology [Ref. C.9]. Regional wave data is also available, an example being for Canadian Waters - including the Great Lakes [Ref. C.10]. Wave data sites are also available on the Internet for example, at, www.meds.dfo.ca maintained by the Canadian MEDS (Marine Environmental Data Services), and www.nodc.noaa.gov maintained by the U.S. National Oceanic and Atmospheric Administration (NOAA).

Wave climate atlases typically divide the world's oceans into blocks or areas. One such system is that of the “Marsden Zones” used frequently in commercial and offshore applications as presented in Ref. C.9.

Wave heights may be quoted directly, usually in 1 m gradations, or alternatively as “sea states”. Ships’ log data often use sea states due to their more direct relationship with observable physical phenomena. Some warship design methods define wave parameters as NATO sea states, and the ship design performance requirements may be expressed in these terms. [Ref. C.11]. When sea state data is used, it will generally have to be transformed into wave height values for use in load calculations. This transformation is described in Section 3.1.4.2(f).

The degree to which the wave climate can be defined depends on how well the route is defined. In cases where the wave climate is unknown, it is advisable to employ “standard” wave climate data for areas known to experience severe weather. A good choice in this circumstance is to use data gathered in the North Atlantic, not only because it experiences severe weather compared with many other areas, but also because the quality of data for this area of the world's oceans is very high as it is particularly well instrumented. Hence, the use of the familiar, “Winter North Atlantic” ocean definition.

The data sources provide long-term wave height and possibly period data, usually expressed in terms of, for example, the frequency of occurrence of a significant wave height and/or period. For fatigue analysis, it is necessary to derive a *distribution* of the wave energy across the wave frequency band from this data.

C.3.1.4.2(b) Selection of Wave Spectral Model

In order to use the wave climate data in response modelling, it is generally necessary to select a spectral model for wave height. This is a mathematical representation of the distribution of wave energy as a function of the spectral parameters.

Much has been written about the use of appropriate sea spectra, and some references are provided. For coastal areas it is important to fit the data to a “limited fetch” spectrum such as JONSWAP [Ref. C.12], while for a fully developed sea, the Bretschneider spectra [Ref. C.13] is popular. The spectra selected must be appropriate to the data available. If the latter includes much energy at swell frequencies, then a spectrum with a greater number of parameters may be necessary.

If the vessel is being designed for a specific area for which actual sea data is available from, say, directional wave buoys, then it may be possible to use this directly. However, care should be taken to ensure that an adequate number of readings are available to provide a statistically valid prediction for the climate the ship will experience.

C.3.1.4.2(c) Encounter Frequency and Reference Axes

Wave height spectra typically refer to the wave climate at a *stationary* point in the ocean. The frequency of waves that the ship experiences differs from the frequency a stationary observer would experience. The former is usually referred to as the “encounter frequency”. The spectrum being used in the analysis needs to be modified to account for this fact. In general, the ship direction will not be aligned with the direction of the waves. The encounter frequency that accounts for the effects of speed and heading is given by:

$$\omega_e = \omega \left(1 - \frac{V}{g} \cos \theta \right) \quad (\text{C.3.1})$$

(See Nomenclature for variable definitions).

The expression for the wave height spectrum also needs to be modified to account for the transformation of the axes system from a fixed point to one that is translating with the ship. The modified wave spectrum is given by:

$$S_{\eta}(\omega_e) = S_{\eta}(\omega) \frac{1}{1 - (2\omega V/g) \cos \theta} \quad (\text{C.3.2})$$

If a software package is used to generate motions and loads, these corrections will normally be generated automatically.

C.3.1.4.2(d) Representation of Confused Seas with Two-Parameter Spectra

In addition, if a two-parameter spectrum is employed, it is often the practice to account for the “short-crestedness” of the seas. This provides a means of accounting for the variation in wave energy with direction within a single- moded spectrum. As a two-parameter spectrum does not explicitly consider wave direction (only height and frequency), the alternative to assuming uni-directional seas (all wave energy concentrated on one axis) is to apply a cosine-squared spreading function as follows:

$$S_{\eta}(\eta_e, \eta') = S_n(\eta_e) \times \frac{2}{\pi} \cos^2 \eta' \quad (\text{C.3.3})$$

Again, most software packages will allow for either uni-directional or 'corrected' representations of the spectrum.

C.3.1.4.2(e) Wave Scatter Diagrams

Wave climate data for both directional and non-directional seas are usually expressed in terms of “wave scatter diagrams” which express the relative frequency of occurrence of certain combinations of wave height and modal period. Hence, using statistical terminology, the diagram is the joint probability density function for wave height and period. A typical non-directional wave scatter diagram (in this case for a composite area covering the North Atlantic) is shown in Table C.3.1.4.1.

The direct calculation method is based on the creation of a composite wave scatter diagram derived from the reference wave climates that comprise the proposed shipping route or operating area. Mathematically, the composite scatter diagram is defined as:

$$(H_S, T_Z)_{\text{composite}} = \sum_{i=1}^{N'} \mu_i (H_S, T_Z)_i \quad (\text{C.3.4})$$

where H_S and T_Z are the significant wave height and zero crossing period respectively, μ_i is the proportion of time spent in the i th area (Marsden zone), and N' is the total number of areas along the route. In practice, the composite wave scatter diagram is created by:

- (1) multiplying the relative frequency values in wave scatter diagrams for each geographical area by their respective factor μ_i ;
- (2) adding all of the modified relative frequency values for common height and period to develop a single weighted scatter diagram.

Table C.3.1.4.1: Scatter Diagram for North-Atlantic for Use in Fatigue Computations

T _z (sec)	H _s (m) for 1000 wave encounters												total
	0.5	1.5	2.5	3.5	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	
13.50	0	0	0	0	0	1	1	0	0	0	0	0	2
12.50	0	0	1	2	2	2	2	1	1	1	0	0	12
11.50	0	1	4	6	7	5	4	2	2	1	1	0	33
10.50	0	4	14	21	18	13	8	4	3	2	1	1	89
9.50	1	16	43	48	34	20	11	5	3	1	1	0	183
8.50	4	47	80	65	38	18	8	4	2	1	0	0	268
7.50	13	75	79	46	21	9	3	1	1	0	0	0	248
6.50	21	54	33	14	5	2	1	0	0	0	0	0	130
5.50	14	13	5	1	0	0	0	0	0	0	0	0	33
4.50	3	0	0	0	0	0	0	0	0	0	0	0	3
3.50	0	0	0	0	0	0	0	0	0	0	0	0	0
2.50	0	0	0	0	0	0	0	0	0	0	0	0	0
Total	56	210	259	203	125	70	38	17	12	6	3	1	1000

While the ideal is to compile a composite wave scatter diagram that reflects the intended route of the ship, this information is not necessarily available. In this situation, it may be necessary and appropriate to use average pre-compiled data. While Table C.3.1.4.1 is intended for use for routes in the North Atlantic, this is significantly more severe than the “world average” wave climate which is shown in Table C.3.1.4.2. The designer has the option of using a more severe set of conditions to ensure a level of conservatism in the fatigue analysis.

Table C.3.1.4.2: Scatter Diagram for World Wide Trade for Use in Fatigue Computations

T _z (sec)	H _s (m) for 1000 encounters												total
	0.5	1.5	2.5	3.5	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	
13.50	0	0	0	0	0	0	0	0	0	0	0	0	0
12.50	0	0	1	1	1	1	0	0	0	0	0	0	4
11.50	0	1	2	3	3	2	1	1	1	0	0	0	14
10.50	0	3	9	11	8	5	2	1	1	0	0	0	40
9.50	1	13	27	24	15	8	4	2	1	1	0	0	96
8.50	4	39	57	38	19	8	3	1	1	0	0	0	170
7.50	13	80	76	37	14	5	2	1	0	0	0	0	228
6.50	32	99	61	21	6	2	1	0	0	0	0	0	222
5.50	44	70	28	7	2	0	0	0	0	0	0	0	151
4.50	32	22	6	1	0	0	0	0	0	0	0	0	61
3.50	8	2	0	0	0	0	0	0	0	0	0	0	10
2.50	1	0	0	0	0	0	0	0	0	0	0	0	1
Total	135	329	267	143	68	31	13	6	4	1	0	0	997

The composite scatter diagram then forms the basis for representing the wave climate associated with the design operational profile, giving the relative proportion of the wave encounters which will have the given height and period (significant wave height and mean crossing period) given by each cell in the scatter diagram.

C.3.1.4.2(f) *Sea States versus Wave Height*

As noted above, wave statistics and other operational profile data may be available in terms of wave heights, sea states, or some combination of the two. If a mix of data is available, a transformation of wave heights to sea states (or vice-versa) will be required. For example, this can be achieved by assuming that significant wave heights follow a three parameter Weibull distributions, where:

$$F(H_s) = 1 - \exp \left[- \left(\frac{H_s - m}{q} \right)^h \right] \quad (C.3.5)$$

Values of q , m and h may be found and the probabilities of sea states determined from:

$$f(\text{Sea State}) = F(H_{s2}) - F(H_{s1}) \quad (C.3.6)$$

This process is demonstrated in the Examples - Section D. Similarly, the wave period data may not be expressed in the same way in a scatter diagram as in a spectral representation (periods may be peak or zero-crossing) and may need to be converted prior to use.

C.3.1.5 Determination of Wave Load Distribution

Once the ship and wave data required have been developed, they can be combined (for the Level 3 method) into a composite "sea operational profile" containing all the information needed to construct a long-term distribution of the loads

Two examples of composite wave scatter diagrams have been presented above. To illustrate the overall process, a third composite has been constructed for the ship whose operational data was presented in tables C.3.1.3.1–C.3.1.3.3. To simplify the presentation, the period data has been excluded and only the wave heights are shown.

The composite wave height probability distribution (Table C.3.1.5.1) for distribution of time in the relevant sea areas (Table C.3.1.3.3) is combined with the conditional probability of vessel speed and wave height (Table C.3.1.3.1) according to the expression:

$$f_{mc} \cdot f_V (V | H_s) = f_s \quad (C.3.7)$$

where f_s = joint probability of significant wave height and speed (Table C.3.1.5.2)

The term $f_V(V | H_s)$ is calculated from Table C.3.1.3.1 as follows:

$$f_V(V | H_s) = \text{prob}(V \text{ and } H_s) / \text{prob}(H_s) \quad (\text{C.3.8})$$

where: $\text{prob}(V \text{ and } H_s)$ is the joint probability of speed and wave height (individual entries in the Table C.3.1.3.1), and $\text{prob}(H_s)$ is the marginal probability of wave heights, shown in the bottom row of Table C.3.1.3.1 as bolded numbers. Table C.3.1.5.2 presents the results of the calculation of f_s :

Table C.3.1.5.1: Composite Distribution of Wave Height Probabilities (f_{mc})

Hs [m]	Marsden Combined
0-1	0.1131
1-2	0.2970
2-3	0.2660
3-4	0.1634
4-5	0.0849
5-6	0.0407
6-7	0.0188
7-8	0.0087
8-9	0.0041
9-10	0.0020
10-11	0.0008
11-12	0.0006

Table C.3.1.5.2: Two-Dimensional Joint Probability Distribution (f_s)

SPEED (kn.)	Significant Wave Height (m)							SUM
	1	2	3	4	5	6	7	
0-6	0.0014	0.0000	0.0014	0.0000	0.0000	0.0006	0.0062	0.0096
6-10	0.0028	0.0012	0.0053	0.0148	0.0075	0.0154	0.0082	0.0552
10-14	0.0065	0.0082	0.0800	0.0686	0.0392	0.0527	0.0164	0.2717
14-18	0.0457	0.0472	0.2103	0.1826	0.1167	0.0570	0.0041	0.6636
SUM	0.0564	0.0566	0.2970	0.2660	0.1634	0.1256	0.0349	1.0000

The third parameter, heading, is then incorporated in the analysis. The procedure is similar to that for speed, but is combined directly in the expression to give a total probability (three-dimensional probability), as calculated from the expression:

$$f_{\text{total}} = f_s f_{\theta}(\theta | H_s) \quad (\text{C.3.9})$$

where $f_{\theta}(\theta | H_s)$ is the conditional probability of heading (θ) for given wave height H_s or sea state.

The latter term is calculated from Table C.3.1.3.2 as follows:

$$f_{\theta} (\theta | H_s) = \text{prob}(\theta \text{ and } H_s) / \text{prob} (H_s) \quad (\text{C.3.10})$$

The values of f_{θ} calculated for every entry from Table C.3.1.3.2 are multiplied by each entry in Table C.3.2.2. In this example, there are four ranges of speeds x five headings = 20 results. This must be repeated for each of seven wave heights. Thus, a matrix of three-dimensional probability of simultaneous occurrence of speed (V), heading (θ) for the given wave height or sea state (H_s) in the combined operating area has $5 \times 4 \times 7 = 140$ entries. Table C.3.1.5.3 gives the results of the calculations. The values shown in Table C.3.1.5.3 are probabilities standardized by multiplying by 1000. Thus the probability of occurrence of Sea State 2 in head seas, with a vessel speed between 14-18 knots is $3.504/1000 = 0.003504$.

When (as in this case) the ship data is drawn from a relatively short period of log data, it may well be that zero joint probabilities appear to exist for certain combinations of conditions. It is then necessary to make some assessment of how to handle these. In this example, the lack of specific ship records for wave heights above 7 m should not be taken to imply that such conditions will never be encountered, as the composite wave data goes up to 12 m seas for the areas under consideration. In Table C.3.1.5.3, the joint probability data for the 7 m wave height has therefore been applied to all higher wave heights. Some such adjustment is necessary, as these conditions may contribute significantly to fatigue damage (and to ultimate strength assessments). On the other hand, it is less important to “fill in the blanks” at the lower end of the range, where the potential inaccuracies are less significant.

Selective elimination of cells in the matrix by combining their ranges and probabilities can reduce the overall computational effort required considerably. For each cell retained, a set of possible wave periods will also need to be considered, as shown in Tables 3.1.4.1 and 3.1.4.2. Each speed, heading, wave height, and period combination will generate its own response spectrum. A full analysis of this matrix could thus require over 1000 separate calculations to be undertaken and their results combined. As shown in the examples in Section D, a reduced scope will normally be quite sufficient.

Table C.3.1.5.3: Operational Profile Summary Table

SPEED (kn.)	<i>Sea State 1</i>					<i>Sea State 2</i>				
	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following
0-6	0.109	0.301	0.364	0.542	0.126	0.000	0.000	0.000	0.000	0.000
6-10	0.214	0.594	0.717	1.067	0.248	0.086	0.246	0.283	0.443	0.103
10-14	0.493	1.367	1.650	2.457	0.571	0.613	1.749	2.009	3.144	0.730
14-18	3.447	9.564	11.546	17.192	3.998	3.504	10.004	11.492	17.986	4.175
SUM	4.3	11.8	14.3	21.3	4.9	4.2	12.0	13.8	21.6	5.0

SPEED (kn.)	Sea State 3					<i>Sea State 4</i>				
	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following
0-6	0.125	0.286	0.330	0.548	0.123	0.000	0.000	0.000	0.000	0.000
6-10	0.467	1.066	1.232	2.045	0.461	1.285	2.968	3.568	5.686	1.293
10-14	7.089	16.186	18.702	31.036	6.990	5.956	13.757	16.538	26.351	5.991
14-18	18.637	42.549	49.163	81.588	18.376	15.857	36.628	44.032	70.159	15.950
SUM	26.3	60.1	69.4	115.2	26.0	23.1	53.4	64.1	102.2	23.2

SPEED (kn.)	<i>Sea State 5</i>					<i>Sea State 6</i>				
	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following
0-6	0.000	0.000	0.000	0.000	0.000	0.061	0.103	0.137	0.221	0.049
6-10	0.639	1.530	1.818	2.898	0.661	1.641	2.779	3.690	5.938	1.308
10-14	3.317	7.940	9.430	15.035	3.428	5.633	9.540	12.669	20.388	4.490
14-18	9.891	23.676	28.121	44.833	10.223	6.085	10.306	13.686	22.024	4.850
SUM	13.8	33.1	39.4	62.8	14.3	13.4	22.7	30.2	48.6	10.7

SPEED (kn.)	<i>Sea State 7</i>					SPEED (kn.)
	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following	
0-6	0.941	0.843	1.651	2.268	0.456	0-6
6-10	1.254	1.125	2.201	3.024	0.608	6-10
10-14	2.509	2.249	4.402	6.049	1.216	10-14
14-18	0.627	0.562	1.100	1.512	0.304	14-18
SUM	5.3	4.8	9.4	12.9	2.6	100

C.3.2 Determination of Load Ranges

C.3.2.1 Level 2 – Method

In the simplified (Level 2) analysis method, the loads are defined in such a way as to allow the application of a statistical method, typically based on a Weibull distribution, to determine the distribution of the load amplitudes and frequencies. For most structural details, the focus will be on the hull girder analysis, but methods for calculating other components discussed above are presented.

C.3.2.1.1 Design Load Equations

The designer will identify design load equations from the various Classification Society rules or specific sets of rules such as those available for warship design, [Ref. C.14]. Designers will also know that such rules are often somewhat empirical, cannot necessarily be interchanged, and, must be applied intelligently, with an understanding of the underlying assumptions and their expected scope of application. This information may be provided in Class notes or through papers in the open literature. The use of a set of equations from any single source will generally result in load (stress) levels which are broadly comparable for a given ship configuration. Thus, any set of load equations can provide the basis for subsequent fatigue design.

Examples of design load formulae are provided below. These formulae, (taken from Ref. C.3) are certainly not the only equations available. If a design is being undertaken with a choice of Classification Society already finalized, it is advisable to use their preferred formulae. It should be noted that Classification Society formulae for fatigue loads are not always identical to those provided for basic scantling design.

The overall approach is to define the loads in terms of the following components:

- vertical bending moment;
- horizontal bending moment;
- torsional bending moment;
- dynamic external pressure loads;
- internal pressure loads due to ship motions.

Each of these may need to be considered under a set of ship loading conditions, depending on the type of vessel under consideration. Examples of design load equations for use in Level 2 analysis are provided in Appendix B.

For naval vessels, fatigue calculations normally only take into account wave bending effects, though local machinery and propeller-induced vibrations may also be important. Most naval design standards incorporate similar parametric formulae to those shown in Appendix B for commercial vessels.

C.3.2.1.2 Determining Loads for Fatigue Analysis

Some approaches to fatigue design also use the “lifetime extreme” stress range as a reference level, but the majority (including this Guide) use a higher exceedence probability/lower value to reflect the fact that most damage accumulates at lower stress ranges. In order to convert values from one probability level to another probability level, a correction factor, f_f (using Weibull distributions) is required. f_f is given by:

$$f_f = \{(\ln p_2)/(\ln p_1)\}^{1/h} \quad (\text{C.3.11})$$

where:

- p_2 = reference probability level to which the load is to be changed (10^{-4} in this Guide)
 p_1 = probability level at which the quantity is specified (generally 10^{-8} as discussed above)
 h = Weibull shape parameter

Specifically, the factor f_f is applied to the load equations to correct them for the revised probability of occurrence.

The Weibull Shape parameter, h , is dependent on the location of the structure and may be established from long-term wave load analysis. The following values are given in Ref. C.3, for tankers and bulk carriers:

$$h_0, \text{ the basic long-term Weibull shape parameter} = 2.21 - 0.54 \log(L)$$

- | | | |
|-----|--|--|
| (1) | For deck longitudinals: | $h = h_0$ |
| (2) | For bottom longitudinals: | $h = h_0 - 0.005 T_{act}$ |
| (3) | For external pressures: | |
| | Ship side above the waterline: | $h = h_0 + 0.05((D_m - z_b)/(D_m - T_{act}))$ |
| | Ship Side at Waterline: | $h = h_0 + 0.05$ |
| | Ship Side below the waterline: | $h = h_0 + 0.05(z_b/T_{act}) - 0.005(T_{act} - z_b)$ |
| (4) | For Internal Pressures on Longitudinal and Transverse bulkheads: | |
| | | $h = h_0 + 0.05$ |

The above equations are for vessels with a typical roll period (natural roll period <14 seconds). For vessels with a long roll period, the value 0.05 in the above equations can be taken as zero.

If information is available from similar designs/operational profiles to suggest that the distribution shape parameter for any or all loadings should be changed, then alternative values can be used in the above correction factors. Similarly, different probability ratios can be introduced if desired.

The general assumption in all simplified (Level 2) approaches is that loads (stress ranges) can be combined by simple superposition (or minor modifications to this) and the same spectrum shape applies to the combined loads as to the principal components. While not strictly true, this assumption is adequate for most cases.

C.3.2.1.3 *Application of Direct Calculation Methods in the Level 2 Approach.*

Under current Class requirements, the loads for certain types of vessel must be analysed using direct calculation methods rather than specified parametric design equations such as those discussed above. This applies, for example, to tankers over 190m length. Classification Societies have developed specialized software for this, including the ABS Safehull system [Ref. C.15]; the LR ShipRight suite [Ref. C.16]; GL's Poseidon system [Ref. C.17]; and DNV's Nauticus program [Ref. C.18]. These methods have several characteristics in common, including:

- computer-based systems
- explicit calculation of extreme loads
- explicit calculation of structural resistance
- capability to assess fatigue performance

The methods incorporate a fatigue design component, generally following the process outlined under the Level 2 Method in Figure C.2.2.1, i.e., they are based on an estimate of extreme load derived, and the stress range spectrum is based on the Weibull model.

The simplifications incorporated in these methods mean that they are not generic in terms of the ship types to which the resulting software can be applied. To reduce computational demand, first-principles results are synthesized by ship type and structural configuration to simpler algorithms. The Classification Societies have developed systems primarily for larger vessels, the main ship types being tankers, bulk carriers, and container ships (this varies between Class Societies). Implicit in many of the systems is an assumption of a specific structural configuration.

The designer can employ the same basic technique using other, non-proprietary methods. In all cases, this will involve identification of a design sea condition, and the development of response amplitude operators (RAO's) for each load component, as per the procedures defined in Section C.3.2.2. Alternatively, there are published parametric equations for hull girder design moment derived from RAO's by Sikora et al. [Ref. C.19 and C.20], that are appropriate for warships and high speed commercial vessels. This approach is described later.

C.3.2.2 Level 3 – Method

C.3.2.2.1 *Basis of the Approach*

The basis for developing loads in the spectral analysis method, the core of the Level 3 approach, is the development of *transfer functions* generally referred to as Response Amplitude Operators, or RAO's. An RAO represents the response of the ship's structure to excitation by a wave of unit height, and is derived over the full range of (encounter) frequencies that will be experienced. RAO's are complex numbers with real and imaginary components that express the amplitude and phase relationship between the wave load (forcing function) and the response.

This Guide does not develop the mathematical basis for the various ship response RAO's. There are a variety of suitable references [Ref. C.21-C.23]. In general, the numerical calculations are computationally demanding and require a computer code. Empirical (model test or parametric) representations have, however, been developed for some specific ship types.

For a Level 3 fatigue analysis:

- (1) An RAO should be developed for every combination of speed and heading represented in the operational profile, as well as for every load component.
- (2) Vessel response is obtained for every individual condition (combination of speed, heading and sea condition). The RAO's are applied to each cell in the wave scatter diagram, using the wave height and period as characteristic values with the selected sea spectrum. Each resulting load response spectrum represents the *short-term* response to the conditions defined in the specific cell. As a short-term response must be obtained for every condition in the operational profile, a large number of spectral analyses is required.
- (3) The fatigue assessment is a *long-term* analysis that employs all the data given in an operational profile. Each individual response spectrum is multiplied by the probability of that combination. The end result is a statistical distribution due to wave loading. This approach is then used to predict the probability of structural failure due to progressive damage accumulation.

Regarding (1), the overall approach is simplified considerably if it is assumed that the ship responses are linear, allowing a single set of RAO's to be used to cover the full range of wave conditions defined in the wave scatter diagram. In extreme seas, the vessel response may be highly non-linear, and thus in principle, different RAO's should be used to calculate the response. However, for the purposes of fatigue analysis, the extreme value loads represent a small number of cycles in the total life, and thus the (non-conservative) error is generally acceptable.

C.3.2.2.2 Calculation of Response Amplitude Operators (RAO's)

An RAO needs to be developed for each of the load types which are considered in the fatigue analysis. These are expressed in the form:

$$H_v(T|2) = \text{RAO for vertical bending moment,}$$

Other RAO's and terms are defined in the Nomenclature.

The RAO's for hull girder response (bending moment and shear) are derived from section properties, the external forces due to hydrostatic restoring forces, the hydrodynamic response of the hull, and the incident wave excitation force.

As indicated by the parametric equations B.9, B.10 in Appendix B, the equations for external and internal pressure loadings are strongly dependent on the hydrodynamic (vessel motion) response and acceleration components. Thus, the basic RAO will be derived from the RAO for the vessel's *motion* response, and will rely on a conversion function to pressure, as outlined in Section 3.3.1.1. In addition, there may be a “dynamic” component due to the incident wave impacting on the structure externally, or internal sloshing of liquids. This load component is developed using standing wave theory, as developed in Ref. C.23 or C.24, for an incident wave on a vertical wall. As this phenomenon is specific to the higher frequency ranges that are not coincident with the peak response to static pressure variation, it has been proposed that a combined RAO is created. This is shown in the third example, Section D.

RAOs can be determined by model tests, full-scale measurements, or by computer programs. Currently, the most commonly used method for predicting ship motions and sea loads, involving the computation of RAO's, is linear strip theory. Several computer programs based on this theory are readily available (e.g., Ref. C.25).

The limitations of strip theory may be overcome with the use of computer programs which use computational fluid dynamics approaches, wave diffraction theory, hull geometry panel methods, and time-domain based codes. These programs are gaining popularity, but are more complex, and the end result may not justify the effort required to generate the data in any situation other than for research or for unusual hull forms. In the context of a fatigue analysis, the limitations associated with the use of strip theory are generally offset by the narrow range of conditions that are affected by those limitations, and by the speed of computation (given that multiple short-term analyses must be generated under the Level 3 approach).

As an alternative to numerical or physical modeling of the RAO's, parametric equations have been developed for specific vessel types and loads. For example [Ref. C.20] provides such data for hull girder bending for warships and high speed commercial craft.

This method relies on a generalized response amplitude operator for vertical bending moment at midships. For a set of ships consisting of frigates, destroyers, and high speed commercial vessels, the RAO is, after appropriate normalization, represented by a single curve, where

$$\begin{aligned} \text{The ordinate of response is: } & (\sqrt{\text{RAO}})/\rho g (L_{BP})^2 B F_1 F_2 \\ \text{The abscissa of frequency is: } & \Omega F_3 \end{aligned} \tag{C.3.12}$$

where:

$$\begin{aligned} F_1 &= |\sqrt[3]{\cos\theta}| \\ F_2 &= 1.1 \tanh(1.5 + V/g) + 0.03(V/g)^2 \\ \Omega &= \omega / (\sqrt{(2\pi g / L_{BP})}) \\ F_3 &= \sqrt{\cos\theta} \end{aligned}$$

Figure 3.2.1 is taken from Ref. C.20 and illustrates the nature of the data.

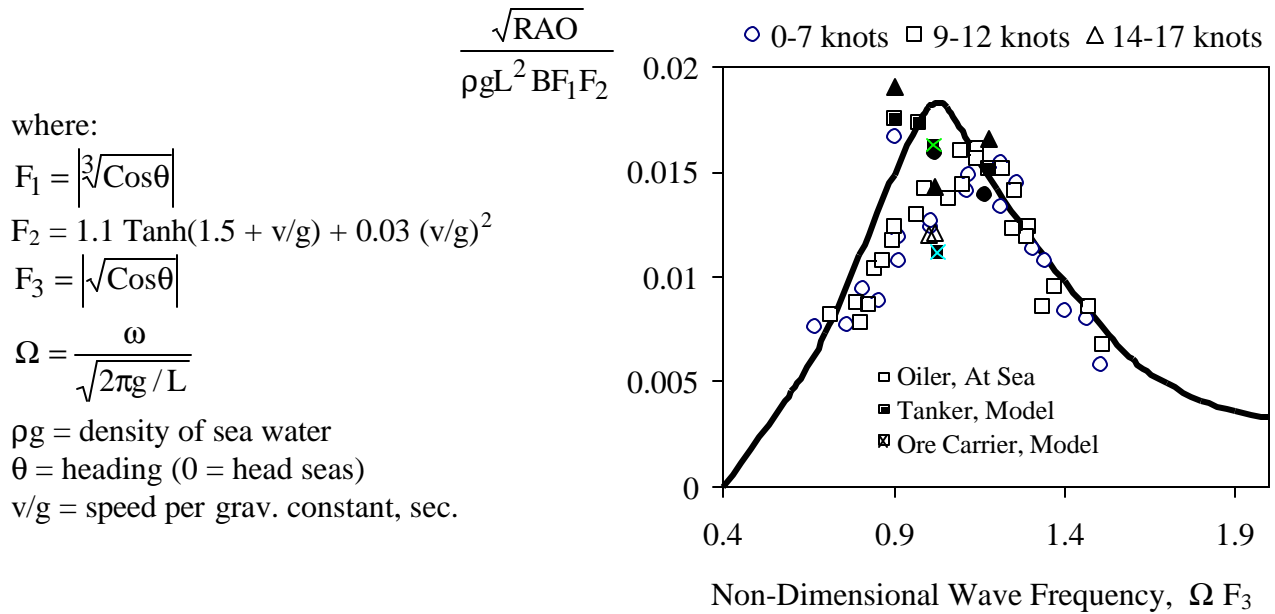


Figure 3.2.1: RAOs for Oilers, Tankers and Ore Carriers

C.3.2.2.3 Determining Load Ranges

In the Level 3 approach, the objective is again to provide a load range. Each loading combination in the operational profile must ultimately be converted from an amplitude to a range.

C.3.2.2.3(a) Wave- Induced Hull Girder Bending Moments

Vertical Bending moment is generally assumed to be fully reversing, i.e., any variation between hog and sag is not explicitly considered. This is a function of the basic linear theory. The vertical bending moment range will be twice the magnitude of the response calculated from the spectral analysis based on the single amplitude RAO for vertical bending.

The horizontal (and torsional) wave bending moment range is not fully reversing. Using an analogous approach to the Level 2 discussion, the horizontal bending range can be equated to the single amplitude horizontal bending response.

C.3.2.2.3(b) External Pressure Range

The Level 3 external pressure range is calculated from two separate transfer functions consisting of:

- (1) The range of quasi-static head at a point of interest on the side shell due to vertical motions of the vessel, is:

$$\Delta p_{st} = \rho g(z_2 - z_1) \quad (C.3.13)$$

where z_1, z_2 are the instantaneous immersions of the POI at the extreme points of motion. The prefix “ Δ ” refers to a “range” value of given parameters where z is derived from the local vertical motion RAOs for the point of interest. It is recommended that the single amplitude response be used due to intermittent emergence of the point of interest, noting that when the point of interest emerges from the water, the outside pressure falls to zero.

- (2) The dynamic pressure, p_d , at the point of interest (POI) – due to the incident wave and reflected wave in deep water given by Equation C.3.14, with terms as defined in Figure 3.2.2, and $d_w = \infty$:

$$p_d = 2 \left[\frac{1}{2} \rho g H \cos(kx) \cos(\omega t) \right] \quad (C.3.14)$$

where k is the “wave number” = ω^2/g in deep water.

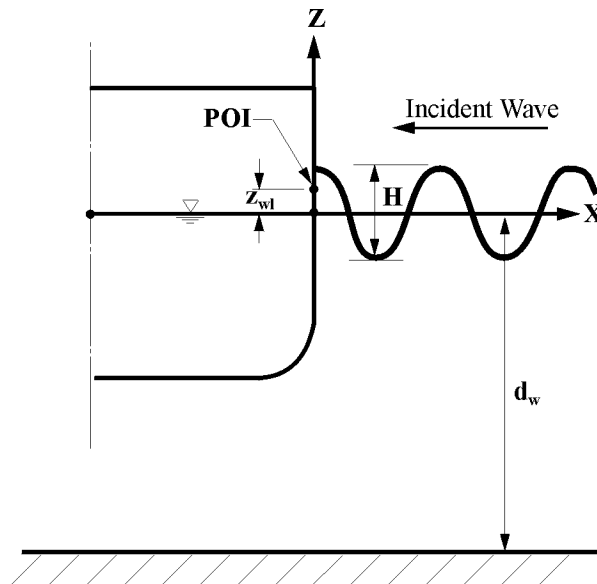


Figure 3.2.2: Dynamic Pressure on Side Shell Scenario

The resulting pressure range transfer function is calculated as the difference between the total pressure at the wave crest and wave trough:

$$\Delta p_d = p_d (\text{wave crest}) - p_d (\text{wave trough}) \quad (\text{C.3.15})$$

These transfer functions are directly calculated as ranges, as they are expressed as a pressure difference. As phasing information is generally unavailable, separate fatigue damage assessments will be conducted for each pressure component, each taken over the life of the vessel.

- (3) The combined (if phase data is available) or individual pressure component is then defined as the “effective lateral pressure” (p) in subsequent stress calculations.

C.3.2.2.3(c) Internal Pressure Loads due to Ship Motion

For reasons analogous to the external pressure calculation, the internal pressure RAO should also be developed as a range. In this case, the designer must be concerned about: (i) saturation of the tank; and, (ii) combination of quasi-static pressure variation and sloshing effects.

C.3.2.3 *High Frequency Loads - Slamming, Whipping and Springing*

In general, the magnitude of high frequency loads that result from slamming (whipping) or springing will be significantly smaller than the wave bending loads. Therefore, unless whipping or springing is expected to occur frequently, they can generally be neglected for fatigue analysis.

Springing is a steady-state phenomena, where higher mode hull responses are excited by general wave interaction. The phenomena is most closely associated with Great Lakes bulk carriers (Lakers) which feature high Length/Depth ratios [Ref. C.6].

Whipping is a transient effect resulting from bow flare or keel slamming effects and are location dependent, where slamming is of greatest concern in the forebody. In the calculation of the extreme value hull girder response, the superposition of transient loads will result in an increase of the loading over the basic wave-induced loading. This has implications for the simplified fatigue approach, and the designer must decide whether or not this component should be carried through the Weibull analysis on the basis of the frequency of occurrence of slamming over the load range (and the extent to which slamming effects may already be accounted for in empirical load formulae).

If these types of transient effects are expected to be significant contributors to fatigue damage, complex representations of ship dynamic response is needed to quantify these with reasonable precision. For a restricted range of fast ship forms (warships and others) a simplified method can be found in Ref. C.20.

Alternatively, slamming may be handled by manipulating appropriate parts of the load spectrum. In the case of both slamming and springing, it may be more appropriate to add an additional safety factor to the “standard” prediction rather than attempt the full analysis.

C.3.3 Summary

For the Level 2 Method, the load amplitudes to be used in the subsequent development of stress ranges are generally determined from well-recognised parametric equations with no explicit determination of the ship operations or the wave climate. For example the vertical bending moment, M_v , may be derived from formulae published in Classification Society rules, and modified to a more appropriate level of exceedance probability as shown herein.

Load amplitudes to be used in determining stress ranges may require adding calculated loads (such as with vertical bending – in which the load range is the addition of the sagging and hogging moments) or, as with horizontal bending, selecting the calculated single amplitude value.

Loads for side and internal structure are likely to be expressed as a pressure, for subsequent treatment in determining stresses using lateral loading analyses techniques.

For the Level 3 Method, a more rigorous examination of the ship operational profile and wave climate is used to produce statistical distributions of load under all the various operating conditions. Response Amplitude Operators (transfer functions) are developed for each load component of interest. In some cases, more than one RAO must be generated to define a load component, e.g., side shell pressure loads. Seakeeping programs may be used to determine the RAOs. The array of operating conditions (generated from the operational profile and wave scatter diagram) and the associated RAOs, then comprise components used to derive a stress range spectrum to be developed in the subsequent sections.

C.4 STRESS ANALYSIS

C.4.1 Introduction

This section provides procedures for relating the ship load distributions or spectra, developed in Section C.3, to the local response of the structural detail of interest for use in the design of fatigue resistant details. This involves developing “ship load” to “structural detail hot spot principal stress” transfer (or response) functions.

The fatigue design processes, presented in this Guide, are based on stress transfer functions or coefficients relating the local stresses to the global hull girder bending moments, external sea pressures acting on the hull, and internal pressures acting on the tank boundaries. The stress coefficients are evaluated by calculating the local field stresses at the point of interest for a unit value of each load component (e.g., vertical, horizontal and torsional bending moment loads, internal and external pressure loads). In general, this will involve conducting stress analyses for unit loads considering each type of loading individually. Strictly, the stress coefficients are a function of wave frequency. However, it appears [Ref. C.3]) that it is acceptable practice to compute stress coefficients for one particular wave frequency, and heading for that matter, and apply it to all wave frequencies and/or headings. The total stress spectrum at the location of interest can then be estimated by combining the stress coefficients and loads spectra using the methods outlined in Section C.2. The remainder of this section outlines methodologies to determine the local stresses from unit loads, hence determining the stress coefficients.

The processes covered in this section are the third and fourth steps outlined in the flow chart for the fatigue design procedures (Figure C.2.1). The two-step process, described below, may be completed via the Level 2 (simplified) or Level 3 (direct spectral analysis) routes or a combination of the two.

Level 2	Level 3
1) Define reference values for nominal stress range components in primary structure using parametric equations, frame and beam models	1) Determine transfer functions for nominal stress range components in primary structure by global finite element analysis
2) Use primary defined SCF's to determine reference values of hot spot stress range components	2) Determine transfer functions for components of hot spot stress range by local finite element analysis

The cross-over linkages, shown in Figure C.2.1, are intended to indicate that any combination of the first and second steps of the Simplified and/or Spectral Analysis approaches may be used in the design of fatigue resistant details. The ability to cross over affords the designer the freedom to use any combination of analytical detail appropriate or available in the design process.

As in the other elements of the fatigue design process, there are a number of approaches with varying degrees of complexity and accuracy that may be used. The approach selected for the stress analysis should, in general, be consistent with the complexity and accuracy applied to other elements of the design process.

C.4.2 Stress and Stress Concentration Category Definitions

In the design of ship structures, both stresses and structural elements may be classified to simplify and isolate structural elements or their response to applied loads. The following definitions of primary, secondary and tertiary stresses [Ref. C.26] will be used in this Guide:

- σ_1 - Primary - stresses due to bending, shear and torsion in the main hull girder;
- σ_2 - Secondary - stresses in a stiffened grillage due to bending and membrane effects; and,
- σ_3 - Tertiary - membrane stresses in panels between stiffeners.

The utility of separating out classes of structure or stresses is less important when numerical (finite element) analysis methods are used to analyse the response of the entire structure. The classification approach is important when stresses are superimposed to estimate the response of structural elements to the applied loads.

In general, the fatigue life of a structural detail is a function of the fluctuating stress field at the point of interest. These stresses may be treated directly, or resolved into four components as shown in Figure C.4.2.1 and described below.

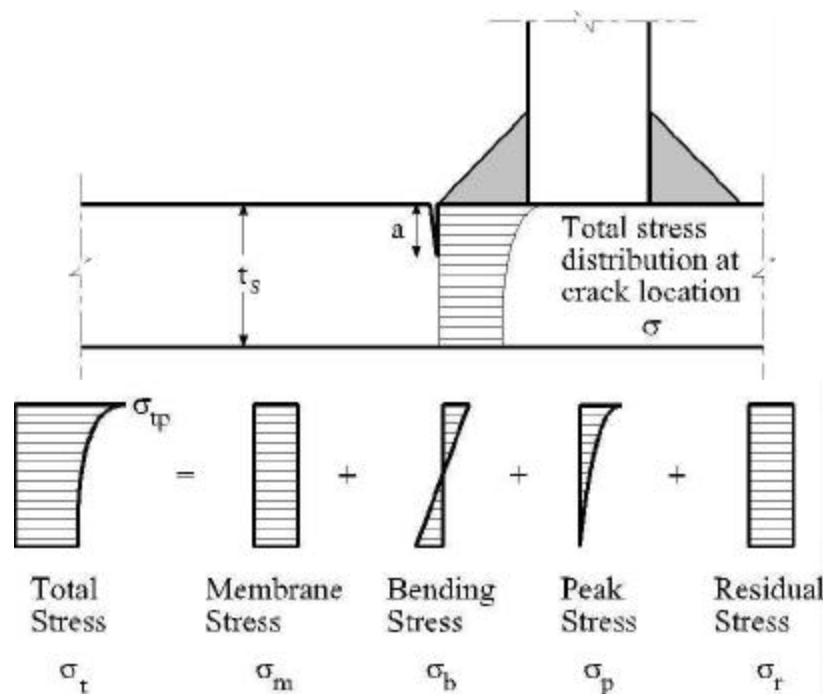


Figure C.4.2.1: Stress Components in a Welded Joint

C.4.2.1 Local Nominal Membrane and Bending Stress (σ_m and σ_b)

The local nominal membrane stress is the uniformly distributed stress that is equal to the average value of stress across the section thickness. The local bending stress is the component of nominal stress due to applied loading that varies linearly across the section thickness. The nominal stresses satisfy the simple laws of equilibrium of forces and moments from applied loads. They may be derived from simple formulae, beam element models, or coarse mesh finite element analysis (FEA) as described in Section C.4.4.2. The term “local nominal stress” is used because stress concentrations resulting from the gross shape of the structure surrounding the local detail of interest will affect the local stress field magnitude (e.g., shear lag effects) and must be included in the local nominal stresses.

C.4.2.2 Peak Stress (σ_p)

The peak stress is the component of stress due to applied loads due to stress concentrations at local discontinuities in the vicinity of the crack. The peak stress represents the highest value, usually at the surface at a notch (e.g., weld toe). Peak stresses arise from stress concentrations due to the following effects:

- (1) **Geometric Stress Concentrations (K_g)**: due to the gross geometry of the detail considered. The effect of the geometric stress concentration typically decays over distances of the order of the section thickness.
- (2) **Notch Stress Concentrations (K_w)**: due to the local geometry of the notch (e.g., weld geometry). The effect of the notch stress concentration typically decays over distances of the order 10% to 20% of the section thickness. Notch stress concentrations are not considered in the structural detail safe life fatigue design process in this Guide.
- (3) **Misalignment Stress Concentrations (K_{te} , $K_{t\alpha}$)**: due to bending stresses caused by misalignments including eccentricity tolerance (K_{te}), and angular mismatch ($K_{t\alpha}$). These are normally used for plate connections only. The effect of the misalignment stress concentrations typically decay over distances of the order of the section thickness.

C.4.2.3 Residual Stresses (σ_r)

Residual stresses are local self-equilibrating stresses that arise from fabrication and welding. In general, residual stresses are strain/displacement limited phenomena and, as such, do not contribute to plastic collapse if they relax. However, they do add to the tensile stress field in the vicinity of the crack and have to be included in the calculation of the stress intensity factor for residual strength assessments. Residual stresses may also be resolved into membrane and bending components. However, since there is only limited quantitative data on the distribution of welding residual stresses in ship structural details, it is normal practice to assume a uniform (membrane) residual stress field approaching tensile yield strength (i.e., $\sigma_r \approx \sigma_y$). Residual stresses need not be considered in the Miner’s summation fatigue design process being presented in this Guide since their effect on the mean stress level should be accounted for in the S-N curve or fatigue life data.

C.4.2.4 Total Stress

The total stress is the sum of the various stress components. The maximum value of total stress at the crack location is referred to as the **peak total stress** (S_{tp}).

C.4.2.5 Stresses in Fatigue Analysis

As mentioned in the previous sections, the stress analysis for fatigue design does not necessarily incorporate all stress concentrating effects. While this Guide is based on a “hot spot” stress approach, other rules may be based on nominal or notch stress stresses. The difference between these approaches lies in how the experimental fatigue test data is reduced in developing the design S-N curves. In order to correctly determine the stresses to be used in fatigue analyses, it is important to note the definition of the stresses used in the S-N curve being applied. Possible stress definitions include:

- Nominal stresses are those derived from beam element models or from coarse mesh FEM models. Stress concentrations resulting from the gross shape of the structure, e.g., shear lag effects, are included in the nominal stresses derived from coarse mesh FEM models.
- Geometric stresses include nominal stresses and stresses due to structural discontinuities and presence of attachments, but excludes stresses due to presence of welds. Stresses derived from fine mesh FEM models provide geometric stress data. Effects caused by fabrication imperfections (e.g., misalignment of structural components), are however normally not included in FEM analyses, and must be separately accounted for. The greatest geometric stress value at the point of interest (including fabrication imperfections), the weld toe, is commonly denoted hot spot stress and the location of the greatest geometric stress outside the region affected by the geometry of the weld is termed the hot spot.
- Notch stress is the total stress at the weld toe (hot spot location) including the geometric stress, misalignment effects and the stress due to the presence of the weld.

C.4.3 **Determination of Local Nominal Stresses**

As discussed in the previous section, local nominal stresses are those that would be calculated at the location of interest in the absence of the stress concentration due to the local structural detail and weld. The local nominal stresses include the stress concentration effects of the overall geometry of the structure surrounding the detail, but not the detail itself.

The local nominal stresses may be calculated, for unit loads, using a combination of parametric formulae for simple structural assemblies and global stress concentration factors to account for the gross geometry of the structure and the effects of misalignment. This approach is referred to in this Guide as the “Level 2” approach.

C.4.3.1 Level 2 Approach

Calculation of hull girder stresses is the simplest way of getting reasonable approximations to the stress levels in longitudinal hull girder elements and connections and may be used for a quick evaluation of stress levels in important details. Global hull girder stresses may be calculated based on gross scantlings. Local stress components should be calculated based on net scantlings, i.e., gross scantlings minus corrosion allowances. The hull girder analogy is based on simple beam theory in which:

- plane cross-sections remain plane;
- stresses remain in the elastic range and thus allow superposition;
- the beam is essentially prismatic (no openings or discontinuities); and,
- there is no interaction between bending and other response modes (e.g., transverse and longitudinal deflections or shear and torsional distortions).

Formulae for calculating hull girder stresses are included in Classification Society Rules. Alternatively, the following formulae derived from formulae presented in [Ref. C.3] may be used. The approach presented here estimates nominal stress levels based on the cumulative contributions of primary (hull girder), secondary and tertiary bending stresses, as shown in Figure C.4.3.1. Secondary and tertiary bending stresses are the result of local bending of structural members due to lateral pressure.

The simplified analysis formulae, presented in the sections which follow, relate the design loads (bending moments and pressures) to nominal stresses based on the geometry of the vessel structure, as defined in Figure C.4.3.2.

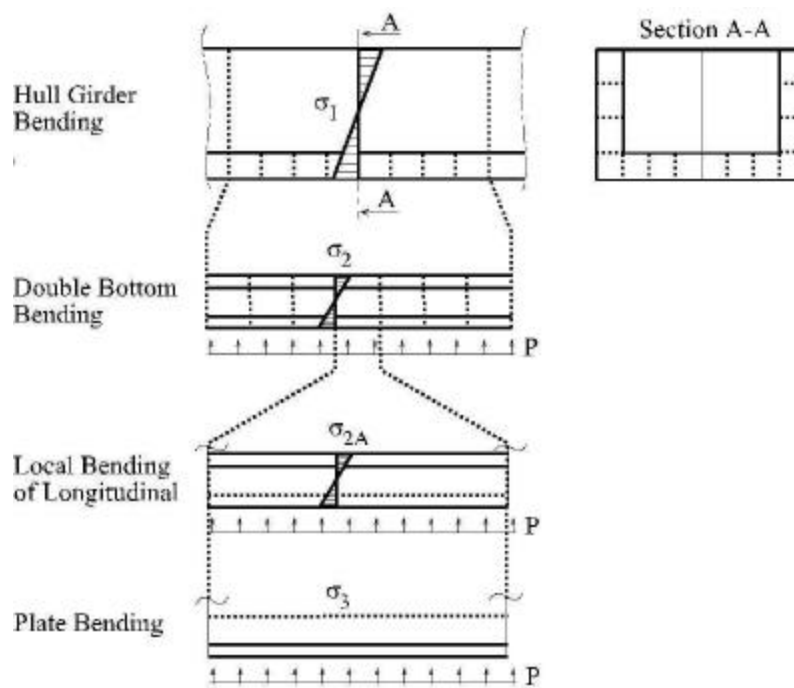
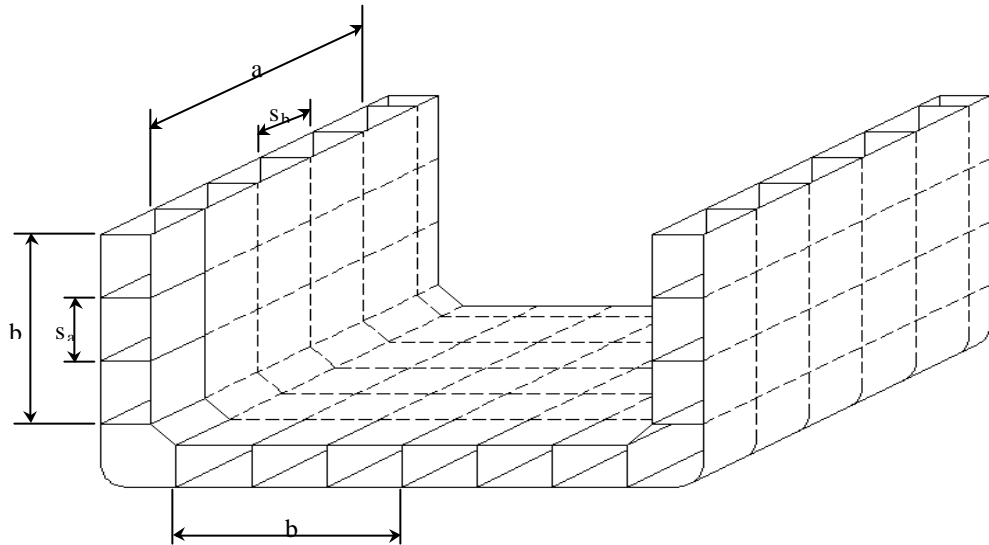
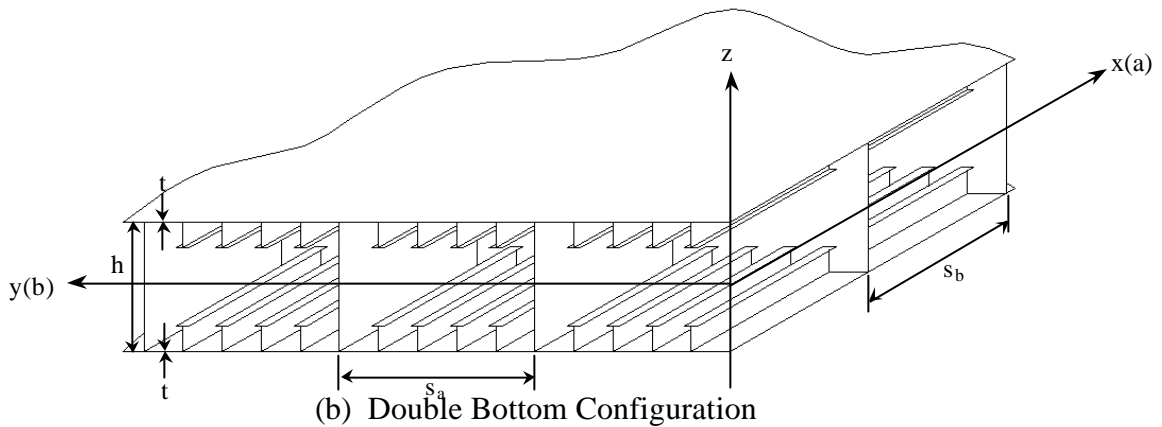


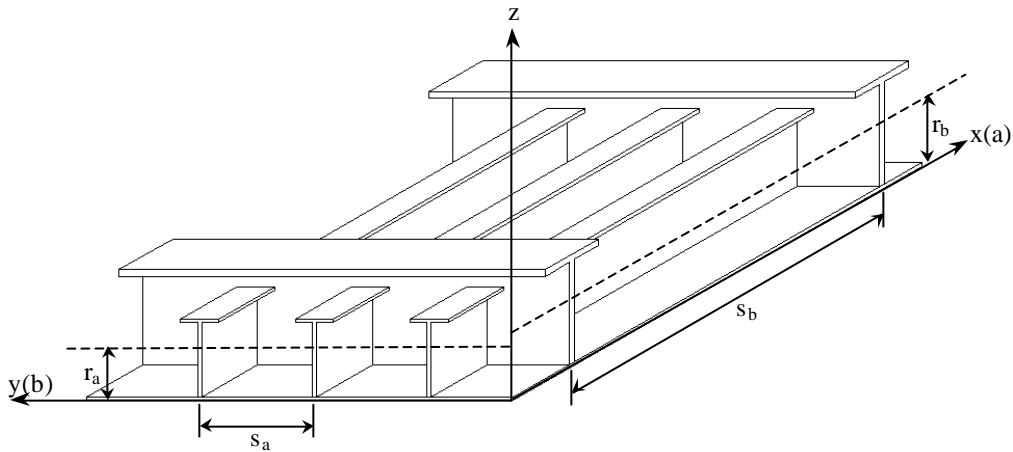
Figure C.4.3.1: Simplified Stress Analysis of Hull Girder [Ref. C.3]



(a) Double Skin Configuration



(b) Double Bottom Configuration



(c) Single Bottom Configuration

Figure C.4.3.2: Definition of Geometric Parameters for Hull Configurations
[Ref. C.3]

C.4.3.1.1 Primary Hull Girder Bending

The horizontal bending stress ($\sigma_{m,h}$) and vertical bending stress ($\sigma_{m,v}$) at a particular point in the ship hull can be combined as follows to account for phase differences between the two stresses:

$$s = \sqrt{s_{m,h}^2 + 2\epsilon s_{m,h} s_{1,v} + s_{m,v}^2} \quad (\text{C.4.1})$$

where ϵ is the stress correlation coefficient. The correlation coefficient is zero in head seas and nearly unity in beam and quartering seas because horizontal bending in these seas is mainly due to heeling of the ship. For design purposes, it is sufficient and conservative to simply add the horizontal and vertical bending stresses.

In the vertical and horizontal bending stress analysis formulae presented below the following structure should be included in the calculation of the section moment of inertia, section modulus or neutral axis location:

- Deck plating (strength decks and other effective decks);
- Shell and inner bottom plating;
- Longitudinal bulkheads and girders; and,
- Longitudinal stiffeners.

Only longitudinally effective structure should be considered in these calculations. For a structural element to be longitudinally effective it must extend over sufficient length for some portion of the longitudinal stress field to enter the structure. An approximate rule of thumb is that longitudinal stresses diffuse at a rate of 1 in 2 in a web (e.g., side shell or longitudinal bulkhead) and about 1 in 4 in flanges (e.g., deck or bottom structure), therefore, the structure must be continuous for twice or four times its width, respectively in order to be considered fully effective.

For vertical hull girder bending:

$$\sigma_{mv} = K_G M_v z_{na} / I_v \quad (\text{C.4.2})$$

and, for horizontal hull girder bending,:

$$\sigma_{mh} = K_G M_h y_{na} / I_h \quad (\text{C.4.3})$$

C.4.3.1.2 Secondary/Tertiary Stresses due to Internal and External Pressure Loads

Local secondary bending stresses are the results of bending, due to lateral pressure, of stiffened single skin or double hull cross-stiffened panels between transverse bulkheads (see Figure C.4.3.2). This approach may be applied to bottom or deck structures, sides or longitudinal bulkheads.

The preferred way of determining secondary stresses is by means of Finite Element Analysis (FEA) or by frame analysis models. Alternatively, secondary bending stresses may be estimated from parametric equations such as the following equations [Ref. C.3]. Similar equations are given in Ref. C.1. The secondary and tertiary bending stresses are presented in the following five categories:

- (a) Longitudinal Secondary Bending Stress in Double Bottom Panels;
- (b) Transverse Secondary Bending Stress in Double Bottom Panels;
- (c) Secondary Bending Stress in Single Skin Panels;
- (d) Bending Stress of Stiffeners Between Transverse Supports (e.g., Frames, Bulkheads); and,
- (e) Tertiary Bending Stress of Plates Bounded by Stiffeners.

The effects of these secondary and tertiary stresses are cumulative with the local nominal stress levels calculated from the hull girder analysis.

(a) Longitudinal Secondary Bending Stress in Double Bottom Panels

Longitudinal secondary bending stresses in double bottom panels at the intersection of transverse bulkheads may be estimated by the following formulae:

- Double Bottom Wider than Long ($b > a$): [Case 1 and 2, Table C.4.3.1]

$$\sigma_2 = (K_b p b^2 r_a) / \sqrt{(i_a i_b)} \quad \rho = (a/b) (i_a / i_b)^{1/4} \quad (\text{C.4.4})$$

- Double Bottom Longer than Wide ($a > b$): [Case 3 and 4, Table C.4.3.1]

$$\sigma_2 = (K_b p a^2 r_a) / i_a \quad \rho = (b/a) (i_b / i_a)^{1/4} \quad (\text{C.4.5})$$

where:

$$\begin{aligned} i_a &= I_a / s_a \\ i_b &= I_b / s_b \end{aligned}$$

(b) Transverse Secondary Bending Stress in Double Bottom Panels

Transverse secondary bending stresses in double bottom panels at the intersection of transverse bulkheads may be estimated by the following formulae.

- Double Bottom Longer than Wide ($a > b$): [Case 3 and 4, Table C.4.3.1]

$$\sigma_2 = (K_b p b^2 r_b) / i_b \quad \rho = (a/b) (i_b / i_a)^{1/4} \quad (\text{C.4.6})$$

- Double Bottom Wider than Long ($b > a$): [Case 1 and 2, Table C.4.3.1]

$$\sigma_2 = (K_b p b^2 r_a) / i_a \quad \rho = (b/a) (i_b / i_a)^{1/4} \quad (\text{C.4.7})$$

(c) Secondary Bending Stress in Single Skin Panels

The stresses at transverse and longitudinal bulkheads may be estimated from the same formulae as for double bottom configurations. However, the parameters ρ and torsion coefficient η should be taken as given in Table C.4.3.2 (also see Table C.4.3.3 for definitions).

(d) Bending Stress of Stiffeners Between Transverse Supports (e.g. Frames, Bulkheads)

The local bending stress of stiffeners with effective flange between transverse supports may be estimated by (Figure C.4.3.3):

$$s_b = K \cdot \left(\frac{M}{Z_s} \right) + \left(\frac{m_\delta EI}{\ell_e^2 Z_s} \right) r_\delta \cdot d \quad (\text{C.4.8})$$

where: M = moment at stiffener support = $\left(\frac{ps \ell_e^2}{12} \right) r_p$

$$r_\delta = 1 - 2 \left(\frac{x}{\ell_e} \right)$$

$$r_\pi = 6 \left(\frac{x}{\ell_e} \right)^2 - 6 \left(\frac{x}{\ell_e} \right) + 1$$

K = stress concentration factor

m_δ = 4.4 at the bulkhead where no stringers or girders support the frames adjacent to the bulkhead; else m_δ must be determined from a beam element analysis as per the procedure in Ref. C.3.

Z_s = section modulus of stiffener

It is of great importance for reliable assessments that bending stresses in longitudinals caused by relative deformation between supports are not underestimated. The appropriate value of relative deformation, δ , has to be determined for each particular case (Figure C.4.3.3). This usually will require 2-D or 3-D frame analysis or coarse mesh FEA.

(e) Tertiary Bending Stress of Plates Bounded by Stiffeners

The local longitudinal tertiary plate bending stress in the weld at the plate/transverse frame/bulkhead intersection midway between longitudinals is given by:

$$\sigma_b = 0.343 p (s / t_n)^2 \quad (\text{C.4.9})$$

Similarly the transverse stress at stiffener mid length is:

$$\sigma_b = 0.5 p (s / t_n)^2 \quad (\text{C.4.10})$$

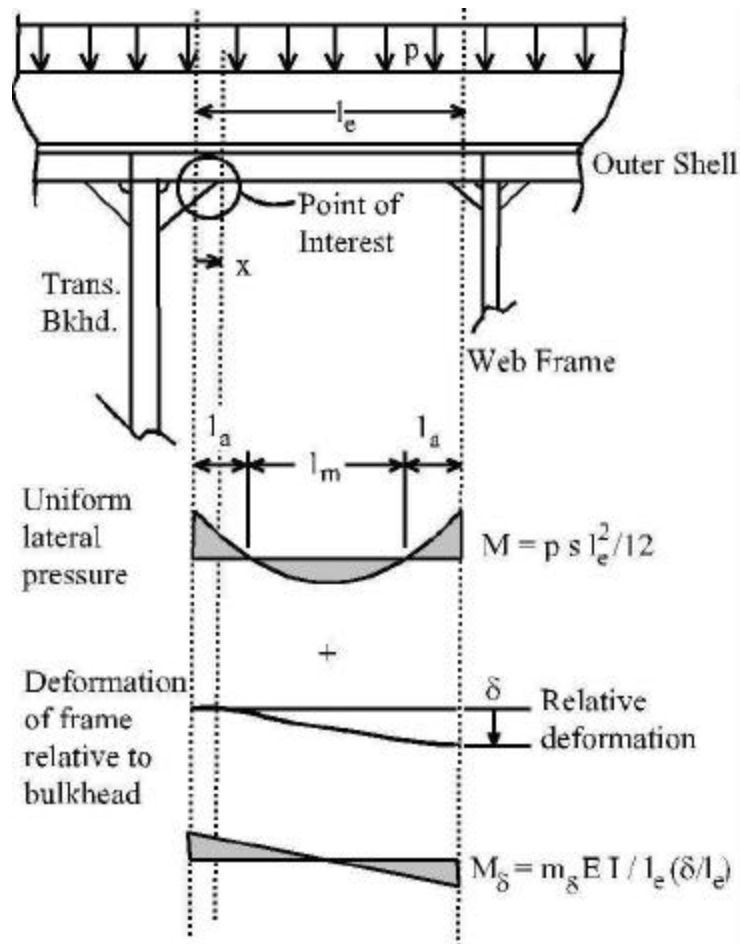


Figure C.4.3.3: Secondary Stresses in a Stiffener [Ref. C.3]

Table C.4.3.1: Support Bending Stress Coefficients K_b - Double Bottom Panels [Ref. C.3]

Case no. & Stress Location	Boundary Conditions	ρ	$\eta = 0.0$	$\eta = 0.5$	$\eta = 1.0$
Case no. 1: Support bending stress in long direction at middle of short end	Long edges: Simply supported	1.00	0.0952	0.0845	0.0767
		1.25	0.1243	0.1100	0.0994
	Short ends: Clamped	1.50	0.1413	0.1261	0.1152
		1.75	0.1455	0.1342	0.1251
		2.00	0.1439	0.1374	0.1300
		2.50	0.1388	0.1381	0.1356
		3.00	0.1371	0.1376	0.1369
		3.50	0.1371	0.1373	0.1373
		4.00	0.1373	0.1374	0.1373
		& up	0.1374	0.1374	0.1374
Case no. 2: Support bending stress in long direction at middle of short end	All edges: Clamped	1.00	-	-	0.0564
		1.10	-	-	0.0591
		1.20	-	-	0.0609
		1.30	-	-	0.0619
		1.40	-	-	0.0624
		1.50	-	-	0.0626
		1.60	-	-	0.0627
		& up	-	-	0.0627
Case no. 3: Support bending stress in short direction at middle of long edge	Long edges: Clamped	1.00	0.0952	0.0845	0.0762
		1.33	0.1026	0.0949	0.0878
		2.00	0.0972	0.0950	0.0926
	Short ends: Simply supported	2.66	0.0920	0.0925	0.0922
		4.00	0.0912	0.0915	0.0917
		& up	0.0916	0.0916	0.0916
Case no. 4: Support bending stress in short direction at middle of long edge	All edges: Clamped	1.00	-	-	0.0564
		1.10	-	-	0.0638
		1.20	-	-	0.0702
		1.30	-	-	0.0755
		1.40	-	-	0.0798
		1.50	-	-	0.0832
		1.60	-	-	0.0857
		1.70	-	-	0.0878
		1.80	-	-	0.0892
		1.90	-	-	0.0903
		2.00	-	-	0.0911
& up	-	-	0.0911		

Notes:

- 1) For intermediate values, use linear interpolation
- 2) See Table C.4.3.3 for definitions of ρ & η

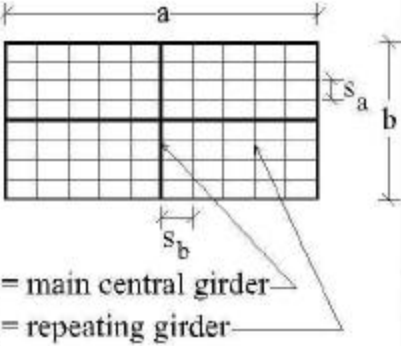
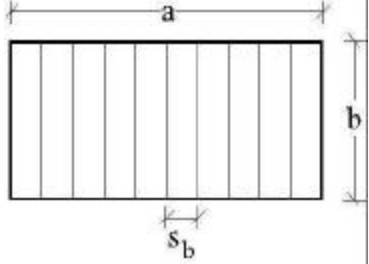
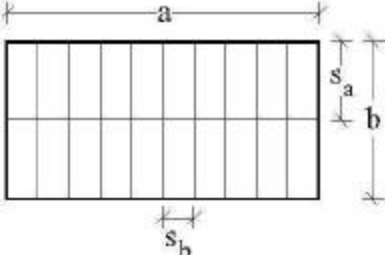
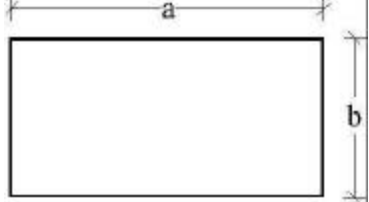
Table C.4.3.2: Support Bending Stress Coefficients K_b - Single Skin Panels [Ref. C.3]

Case no. & Stress Location	Boundary Conditions	ρ	$\eta = 0.0$	$\eta = 0.5$	$\eta = 1.0$
Case no. 5: Support bending stress in long direction at middle of short end	Long edges: Simply supported	1.00	0.0866	0.0769	0.0698
		1.25	0.1140	0.1001	0.0904
	Short ends: Clamped	1.50	0.1285	0.1148	0.1049
		1.75	0.1324	0.1221	0.1139
		2.00	0.1310	0.1250	0.1191
		2.50	0.1263	0.1257	0.1234
		3.00	0.1248	0.1253	0.1246
		3.50	0.1248	0.1250	0.1246
		4.00	0.1240	0.1250	0.1250
		& up	0.1250	0.1250	0.1250
Case no. 6: Support bending stress in short direction at middle of long edge	Long edges: Clamped	1.00	0.0866	0.0769	0.0698
		1.33	0.0934	0.0858	0.0799
		2.00	0.0885	0.0865	0.0843
	Short ends: Simply supported	2.66	0.0837	0.0842	0.0839
		4.00	0.0830	0.0832	0.0835
		& up	0.0834	0.0834	0.0834

Notes:

- 1) For intermediate values, use linear interpolation
- 2) See Table 4.3.3 for definitions of ρ & η

Table C.4.3.3: Definition of Stiffness and Geometry Parameters [Ref. C.3]

Type	Sketch	Formulas for ρ and η
<p>A: Cross stiffening Middle girder / stiffener in both directions are stiffer than the others</p>	 <p>I_b = main central girder $I_{b'}$ = repeating girder</p>	$i_a = \frac{I_{na'}}{s_a} + 2 \left(\frac{I_a - I_{na'}}{b} \right)$ $i_b = \frac{I_{nb'}}{s_b} + 2 \left(\frac{I_b - I_{nb'}}{a} \right)$ $r = \frac{a}{b} \sqrt[4]{\frac{i_b}{i_a}}$ $h = \sqrt{\frac{I_{pa} I_{pb}}{I_{na} I_{nb}}}$
<p>B: Modified cross stiffening One girder / stiffener in a-direction only</p>		$i_a = 2 \frac{I_a}{b}$ $i_b = \frac{I_{nb}}{s_b} + 2 \left(\frac{I_b - I_{nb}}{a} \right)$ $\rho = \frac{a}{b} \sqrt[4]{\frac{i_b}{i_a}}$ $\eta = 0.124 \sqrt{\frac{I_{pb}^2 b}{I_a I_{nb} s_b}}$
<p>C: Single stiffening Girders / stiffeners in b-direction only</p>		$i_a = 0$ $i_b = \frac{I_{nb}}{s_b}$ $\rho = \text{inf inite}$ $\eta = \text{in det er min ate}$
<p>D: Unstiffened plate</p>		$i_a = i_b = \frac{t^3}{12(1 - \nu^2)}$ $\rho = \frac{a}{b}$ $\eta = 1.0$

C.4.3.2 Level 3 Approach Using Finite Element Analysis

In the design of critical structural elements, or when the global structure is too complicated for simple parametric formulae, finite element analysis (FEA) may be used to obtain a reliable description of the overall stiffness and global stress distribution in the hull.

The global FEA is generally carried out with a relatively coarse mesh, the main objective being to obtain a good representation of the overall membrane panel stiffness in the longitudinal and transverse directions and for shear, sufficient for determination of nominal stresses. Stiffened panels may be modelled by means of anisotropic elements or, alternatively, using a combination of plate and beam elements.

The extent of the model is dependent on the type of response to be considered and the structural arrangement of the hull. If the FEA based design process involves only several localized details, the required extent of the local model is dependent on the stiffness variation of the hull over a certain length and this has to be captured in the global FEA model. The minimum hull module length required to accurately portray the structural response and provide the additional information not available from the simplified analysis approach typically includes several cargo holds (or watertight compartments of a naval vessel). The exact length requirement depends on the ship's overall geometry and nature and arrangement of the cargo or other loads.

For horizontal and torsional bending response of the hull of an open hatch ship, it is generally required that the extent of the global model covers the complete hull length, depth and breadth (a half breadth model may be used if antisymmetric boundary conditions can be assumed at the centerline). A complete finite element model may also be required for the evaluation of vertical hull girder bending of ships with complex superstructure arrangements (e.g., warships, passenger ships), and for ships of complex cross-section (e.g., catamarans).

Alternatively, a part of the hull (for example, the midship area) may be modelled. Hull girder loads should be applied individually at each end of the model to result in a value of unit load (e.g., bending moment) at the location of interest. Unit pressure loads will normally be distributed over the appropriate section of the hull. The loads should be balanced in order to give a minimum of reaction forces at the supports (boundary conditions). The loads and boundary conditions in the hull cross section should be evaluated carefully when modelling only a part of the hull to avoid unrealistic stiffness from the forebody/afterbody.

Figure C.4.3.4 shows an example of a global finite element model of a section of a bulk carrier. This model may be used to calculate nominal global stresses and deformations away from areas with stress concentrations. In areas where local stresses in web frames, girders or other areas (for example hatch corners) are to be considered, the global model should have a mesh producing deformations applicable as boundary conditions for local stress analysis. In such cases, the global and local models should be compatible. The local model may be directly applied as a substructure or super-element in the global model (if such techniques are available with the FEA software). The substructure technique ensures that forces and deformations in the global and local models are compatible and, if the substructure is detailed enough, local stress results may be obtained directly. The substructure technique is very effective where local structural assemblies (i.e., the substructure) are repeated several times in the overall assembly, but it does present added complexity into the analysis.

More commonly, the global and local analyses are conducted separately. Nodal forces and/or displacements obtained from the global model are applied as boundary conditions for the local model. In general, the stiffness of the local model should be comparable to that of the global model representation so that forces and displacements between the two models are compatible. However, due to the greater level of geometric detail and mesh refinement of the local model, this is rarely achievable. As such, it is preferable that nodal forces be transferred from the coarse model to the local model rather than forced displacements. It is important that the extent of the local model is sufficiently large that boundary effects due to prescribed forces or displacements are away from the areas where accurate stresses need to be determined.

The loads to be applied in the global analysis can be produced using any of the methodologies presented in Section C.3. The global analysis should be conducted for each load case individually (i.e., vertical bending, horizontal bending, torsional bending, external pressure, internal pressure). Each load case should be analyzed for a unit value of the applied load at the location being considered. In this manner, the stresses derived from subsequent local analysis will correspond to unit loading and therefore, be equal to the stress coefficients, A_i , which are required to generate the local stress spectrum from the combined loading spectra.

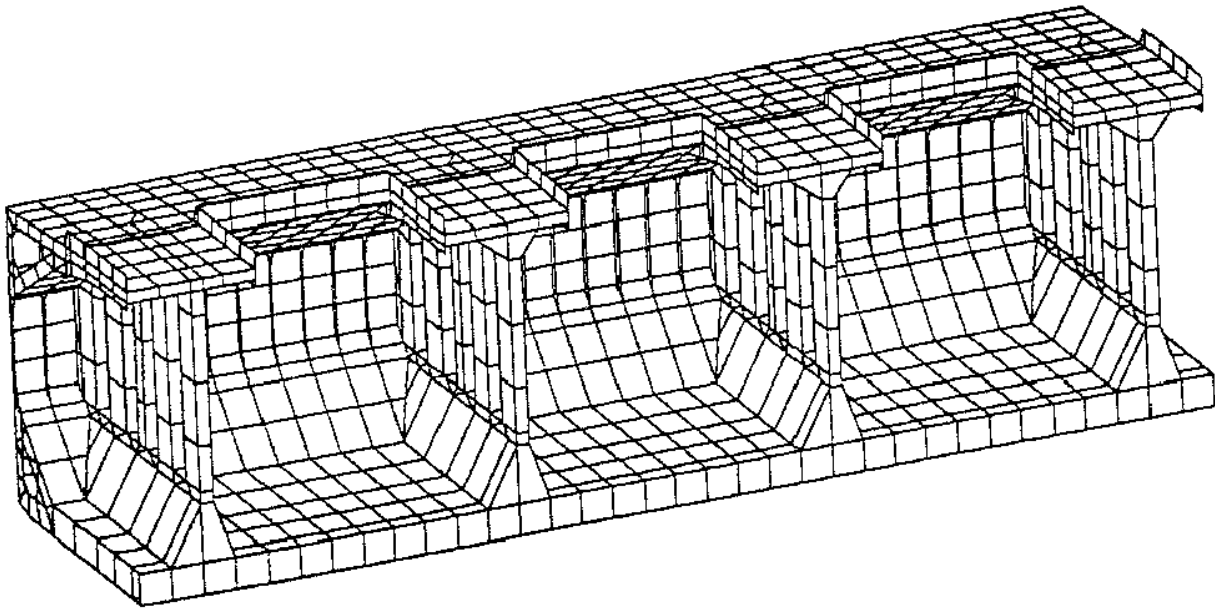


Figure C.4.3.4: Global Finite Element Model of Bulk Carrier [Ref. C.1]

C.4.4 Determination of Peak Stresses

Peak stresses may be estimated based on parametric approximations of stress concentration factors for ship details, when these are available. Alternatively, they may be determined based on local fine mesh finite element stress analysis of the joint.

C.4.4.1 Evaluation of Peak Total Stress

The peak total stress can be evaluated by :

$$\sigma_{tp} = \sigma_m + \sigma_b + \sigma_p + \sigma_r \quad (\text{C.4.11})$$

$$\sigma_{tp} = K_g \cdot K_w \cdot (K_{te} \cdot K_{t\alpha} \cdot \sigma_m + \sigma_b) + \sigma_r$$

In a fatigue design process based on a hot spot stress approach, the notch stress concentration factor (K_w) is omitted. The notch stress concentration associated with the weld toe is accounted for in the S-N curve or fatigue life data. The notch stress concentration due to an existing defect would only be considered in a damage tolerance or residual life assessment as opposed to the safe life fatigue design process presented in this Guide.

The nominal membrane, bending and peak stress components due to the applied loads (excluding residual stresses) may be derived, for a given stress distribution $\sigma(x)$ for $x = 0$ (at the surface) to $x = t_s$ (through the thickness), by the following analytical expressions Ref. C.27 (see Figure C.4.2.1):

$$s_m = \frac{1}{t_s} \int_0^{t_s} s(x) \cdot dx \quad (\text{C.4.12})$$

$$s_b = \frac{6}{t_s} \int_0^{t_s} s(x) \cdot \left(\frac{t_s}{2} - x \right) \cdot dx \quad (\text{C.4.13})$$

$$s_p(x) = s(x) - s_m - s_b(x) \quad (\text{C.4.14})$$

C.4.4.2 Stress Concentration Factors for Ship Details

Stress concentration factors (SCF) for a range of typical ship structure details are available in a number of references, including [Refs. C.1, C.3, C.28, C.29] Stress concentration factors for typical ship structural details (K_g) and for misalignment effects (K_{te} , $K_{t\alpha}$) are defined in Section 4.2.2 and presented in Appendix C.

It is necessary to exercise extreme care when applying stress concentration factors from different sources to ensure that the correct nominal stress definition is used. For example, in some cases the nominal stress is defined at the intersection point of a connection, in other cases, the global nominal stress may be defined at the weld toe or some distance from the weld toe.

Furthermore, the designer should be aware that sometimes the published stress concentration factor solutions are designed to calculate the "hot spot" stress or the "notch" stress as opposed to the local nominal stress. It is, essential, therefore, to make certain which form of peak stress will result from the application of the SCF.

C.4.4.3 Local Finite Element Analysis

If appropriate stress concentration factors are not available, the total stress distribution including local peak stresses may be calculated by local FEA. As discussed previously in the section on global FEA, the extent of the local model should be large enough that the calculated results are not significantly affected by assumptions made for boundary conditions and application of loads.

Figure C.4.4.1 shows a local finite element model of a ship detail. The local model should have a relatively fine mesh, especially in areas of stress concentration. It is important to have a continuous and not too steep change in the density of the element mesh in the areas where the local stresses are to be analyzed. The geometry of the elements (aspect ratio, corner angles, skewness and warp) at the point of interest should be as near optimal as possible (for example: element length/breadth aspect ratio less than 2, corner angles between 60° and 120° , avoid use of triangular elements with reduced order shape functions).

Local FEA of a joint is usually conducted to determine the local nominal and hot spot stress at the location of interest. If the peak notch stress has to be determined (i.e., for fatigue analysis approaches other than that used in this Guide), then the most common approach is to use local FEA to evaluate the hot spot stress. The hot spot stress value is then factored by a weld notch factor, K_w , derived from parametric equations or tables to provide an estimate of the peak notch stress in the joint.

Finite element size requirements in the stress concentration region are dependent on the type of element. The mesh size may be determined based on experience or by benchmark testing a similar mesh for a case where results have been presented in the literature. Figure C.4.4.2 provides some guidance on element sizes for 20-node solid, 8-node shell and 4-node shell element types suitable for determining the stress concentrations consistent with approach advocated by this Guide.

Normally the element stresses are derived at the Gaussian integration points. Depending on the element type, it may be necessary to perform several extrapolations in order to determine the stress at the weld toe. Referring to Figure C.4.4.3, all stress components are used for the extrapolation. The process is as follows:

- (1) Extrapolate the stresses to the surface from the Gauss points based on the assumed distribution function in the element (some FE programs will provide this on request);
- (2) Extrapolate surface stress to a line A-B centred on the hot spot of interest;
- (3) Calculate stress along line A-B at reference points taken at $t/2$ and $3t/2$ from hot spot;
- (4) Linearly extrapolate through reference points $t/2$ and $3t/2$ to determine stress at hot spot;
- (5) Having extrapolated stress components for the hot spot, the principal stresses are calculated at that location for fatigue analysis.

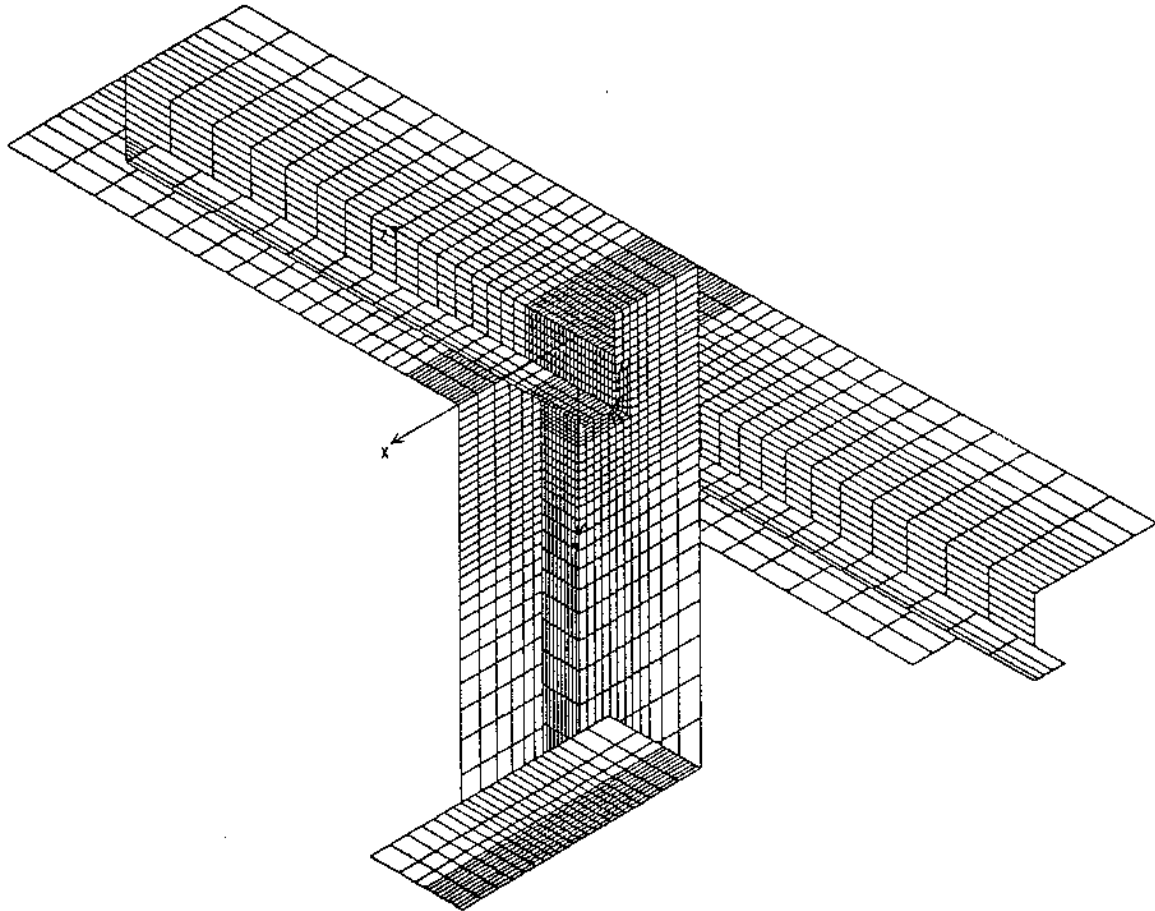
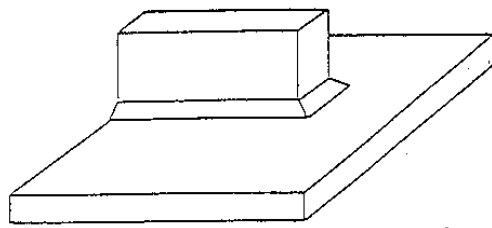
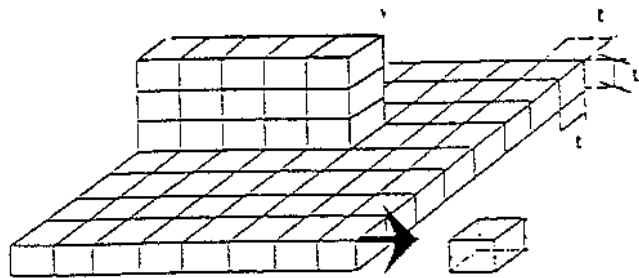


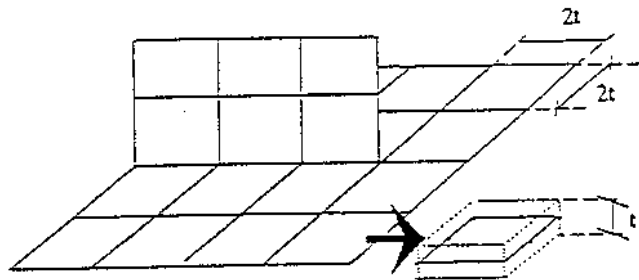
Figure C.4.4.1: Local Finite Element Model of Ship Detail [Ref.C.30]



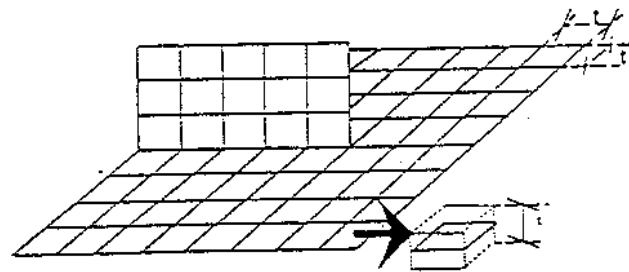
Structural Detail



Model with 20-node solid elements (size $t \times t \times t$)



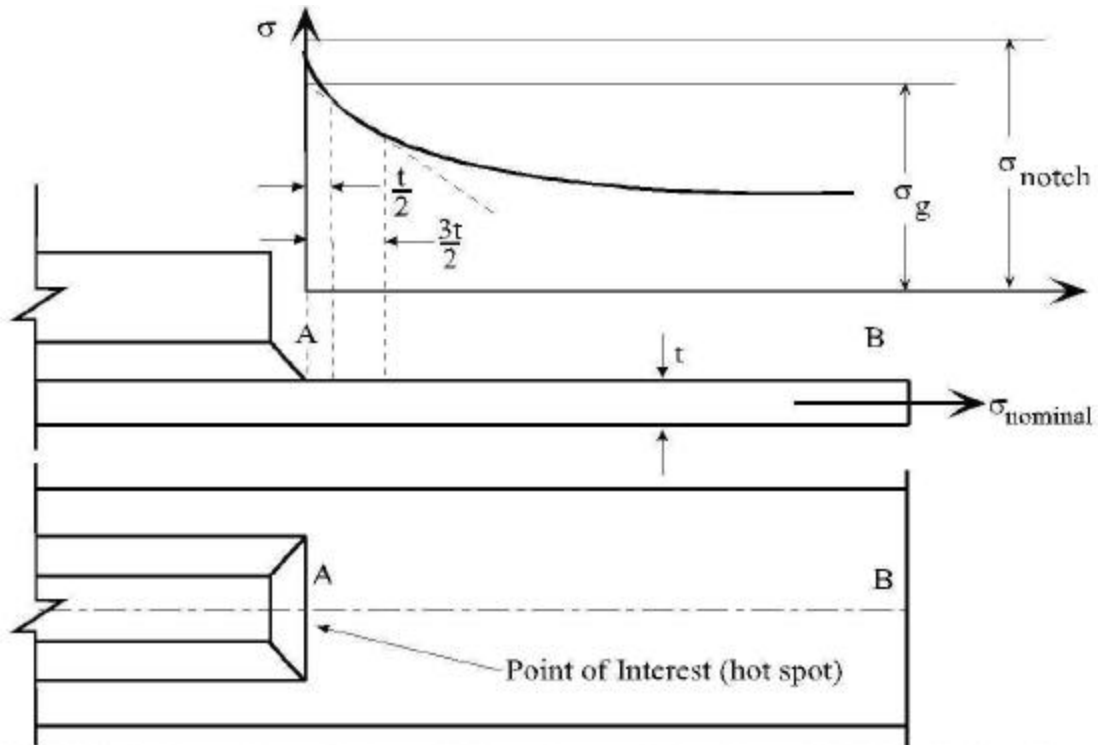
Model with 8-node shell elements (size $2t \times 2t$)



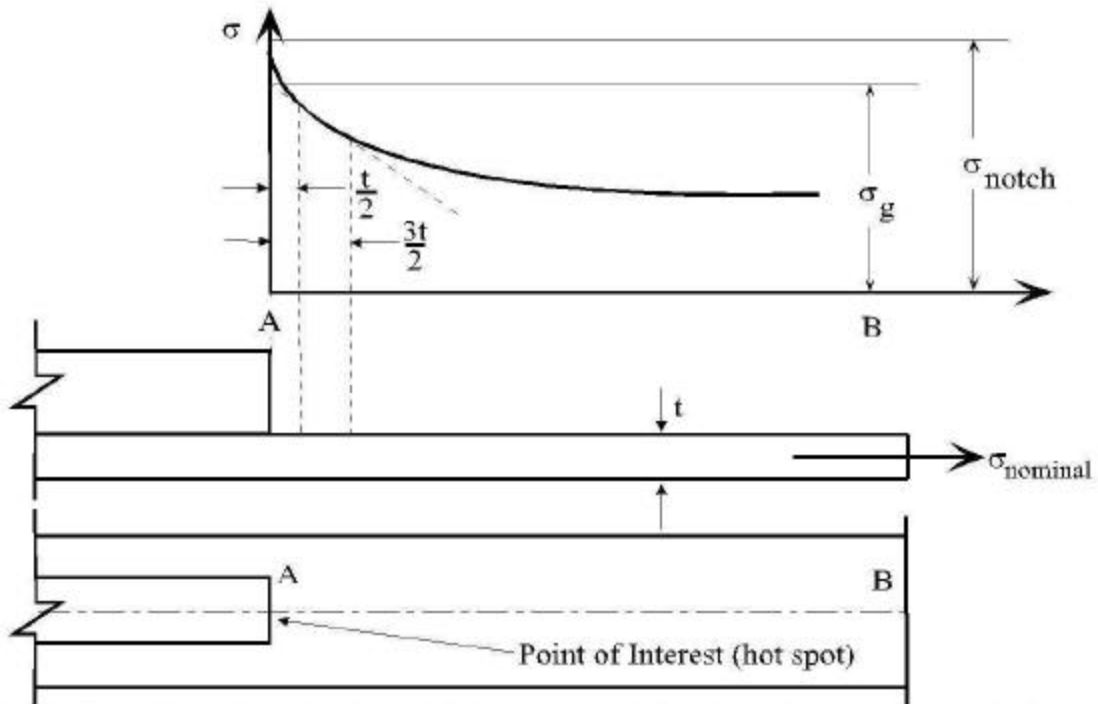
Model with 4-node shell elements (size $t \times t$)

Element Type	Element Size
20 – node isoparametric solid element	$t \times t \times t$
8 – node quadrilateral isotropic shell element	$2t \times 2t$
4 – node quadrilateral isotropic shell element	$t \times t$

Figure C.4.4.2: Recommended Element Sizes for Local Detail FEA [Ref. C.3]



FE Model Hot Spot Stress Extrapolation when weld toe is modeled (i.e. brick elements)



FE Model Hot Spot Stress Extrapolation when weld toe is not modeled (i.e. shell elements)

Figure C.4.4.3: Stress Distribution at an Attachment and Extrapolation of Stresses at Hot Spot [Ref. C.3]

C.5 DEVELOPMENT OF STRESS RANGE DISTRIBUTIONS

C.5.1 Introduction

This section describes procedures for combining the environmental, vessel and structural response information, *to produce the stress range distributions* required for the fatigue damage summation. This involves applying the response functions previously developed.

The processes covered in this section are the fifth and sixth steps outlined in the flow chart for the fatigue design procedures (Figure C.2.1). The two step process, described below, may be completed for the Level 2 or Level 3 approaches. These two generalized steps are expanded and described in the sections which follow.

Level 2	Level 3
5) Develop hot spot stress ranges from load and stress transfer functions	5) Develop hot spot stresses from load and stress transfer functions
6) Define long-term distribution for notch stress range	6a) Define short-term distribution for hot spot stress ranges 6b) Define long-term stress distribution by summing short-term distributions

C.5.2 Level 2 Approach

This step involves the combination of data and transfer functions to develop a characteristic stress range value from which a *long term* stress range distribution is inferred. The information required to develop a long term distribution includes:

Design Load Data

- Components of ultimate strength loading (M_{hog} , M_{sag} , p_{int} , p_{ext}) [Section C.3.2]
- Reference wave encounter period for the design load ($n_0 = 10^4$) [Section C.3.1]
- Load to load range approximation (e.g. fully reversing moments) [Section C.3.1]

Stress Transfer Functions

- Nominal stress transfer functions ($F_{sm}(load)$, $F_{sb}(load)$, K_G) [Section C.4.3]
- Hot spot stress transfer functions (K_g , K_{te} , $K_{t\alpha}$) [Section C.4.4]

Modifying Factors

- Encounter frequency correction factor (f_f) [Section C.3.2]

Determination of the hot spot stress range distribution (see C.4.2.4) is a three step process (illustrated for vertical bending moment):

- (1) Calculate reference nominal stress range

$$\Delta\sigma_{\text{nom}} = (\Delta M_v y_{\text{na}} / I_v) K_G f_f \quad (\text{C.5.1})$$

where: $\Delta M_v = 2M_v$ or $(M_{\text{hog}} + M_{\text{sag}})$ and f_f is the factor to transform the load from one probability level to another probability level.

- (2) Calculate reference hot spot stress range

$$\Delta\sigma_{\text{hotspot}} = K_g \cdot K_{te} \cdot K_{t\alpha} \Delta\sigma_{\text{nom}} \quad (\text{C.5.2})$$

- (3) Develop long term Weibull stress range distribution for the location of interest

$$Q(\Delta\sigma_{\text{hot spot}}) = 1 - \exp(-[\Delta\sigma_{\text{hotspot}} / q]^h) \quad (\text{C.5.3})$$

where: $Q(\)$ is the long term Weibull cumulative distribution function for a stress range

$\Delta\sigma_{\text{hot spot}}$.

$$q = \Delta\sigma_{\text{hot spot}} / [\ln(n_0)]^{1/h}$$

h is the Weibull shape parameter (see Section C.3.2.1.2)

Based on this approach, a long term probability distribution function of stress ranges in either continuous (Figure C5.2.1a) or discrete form (Figure C5.2.1b) is available for the fatigue damage summation.

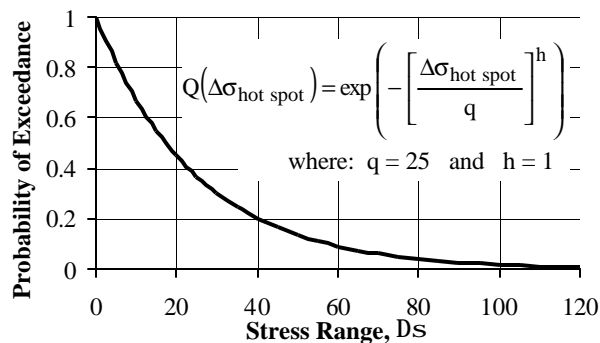


Figure C5.2.1a: Continuous Weibull Stress Range Distribution

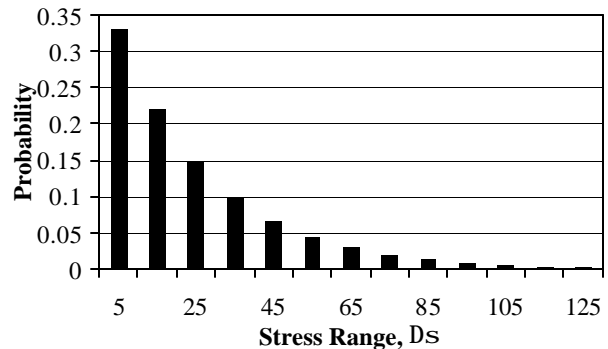


Figure C5.2.1b: Discrete Weibull Stress Range Distribution

The Weibull parameters (q and h) used to develop the continuous distribution are ready to be used in the closed form fatigue damage summation (see Section C.6.3.2), whereas, the discrete distribution may be used in the discrete fatigue damage summation described in Section C.6.3.1.

C.5.3 Development of a Stress Range Distribution for the Level 3 Approach

This step involves the combination of data and transfer functions to develop a characteristic stress range value from which a series of *short term* operational condition stress ranges are developed and then summed to produce a *long term* stress range distribution. The information required includes:

Design Load Data

- Composite wave scatter diagram(s)..... [Section C.3.1]
- Reference wave encounter period for the design load ($m_0 = 10^4$) [Section C.3.1]
- Load to load range approximation, F_{load} [Section C.3.1]
- Characteristic load response [Section C.3.2]
- Conditional probabilities for wave direction and velocity (f_v)..... [Section C.3.2]

Stress Transfer Functions

- Nominal stress transfer functions (Global FEA results)..... [Section C.4.3]
- Hot spot stress transfer functions from unit load, local FEA)..... [Section C.4.4]

Determination of the hot spot stress range distribution is a three step process (as illustrated for vertical bending moment):

- 1) Calculate reference hot spot stress range

$$\Delta\sigma_{hot\ spot} = 2 K_0 F_{load}(H_s, V, \theta, L.C. i) \dots \text{for each } (H_s, V, \theta, L.C.) \quad (C.5.4)$$

where: K_0 is the stress transfer function relating vessel load response to detail hot spot stress

F_{load} is the characteristic (RMS) load response of the vessel such as M_v, M_H, p

- 2) Develop Rayleigh short-term stress range distribution

$$F_{\Delta\sigma i}(\Delta\sigma_{hotspot}) = 1 - \exp(-\Delta\sigma_{hotspot}^2 / 8 m_{0i}) \dots \text{for each } (H_s, V, \theta, L.C.) \quad (C.5.5)$$

where: m_{0i} is the spectral zeroth moment for each of the i operational conditions inferred from reference hot spot range ($m_0 = \Delta\sigma_{hotspot\ RMS}^2$)

$F_{\Delta\sigma i}(\Delta\sigma_{hotspot})$ is the Rayleigh “short term” hot spot stress cumulative density function for the i^{th} operational condition.

- 3) Develop long-term stress range probability distribution function (Weibull or Histogram)

$$Q'(\Delta\sigma_{hotspot})_{long} = \sum r_i Q'(\Delta\sigma_{hotspot})_{short} \cdot p_i \quad (C.5.6)$$

$$Q'(\Delta\sigma_{hotspot})_{short} = F_{\Delta\sigma}(\Delta\sigma_{hotspot})_{upper} - F_{\Delta\sigma}(\Delta\sigma_{hotspot})_{lower}$$

where: p_i is the fraction of time at the i^{th} operational condition

r_i ratio of the i^{th} zero crossing rate (f_{zi}) to the average zero crossing rate for all operational conditions $r_i = f_{zi} / \bar{f}_z$

where: $\bar{f}_z = \sum p_i f_{zi}$

$Q'(\Delta\sigma_{hotspot})_{long}$ is the “long-term” hot spot stress probability distribution function

$Q'(\Delta\sigma_{hotspot})_{short}$ is the “short-term” hot spot stress probability distribution function

In this third step, a weighted sum of Rayleigh “short-term” distributions is used to develop a “long-term” distribution of stress ranges. A direct approach to this involves breaking each Rayleigh distribution, associated with an operational condition, into discrete stress range intervals and estimating their probability of occurrence, as shown in Figure C.5.3.1 and Table C.5.3.1.

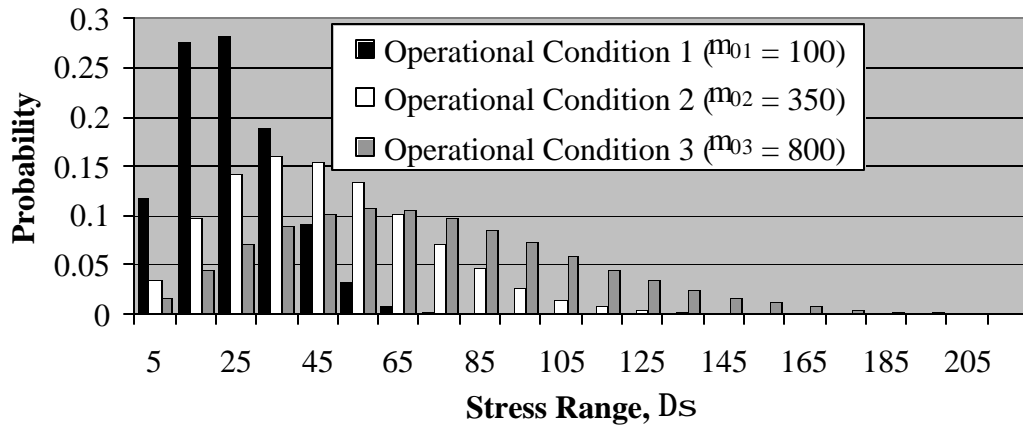


Figure C.5.3.1: Operational Condition Rayleigh Distribution Stress Range Interval Probabilities

Table C.5.3.1: Operational Condition Rayleigh Distribution Stress Range Interval Probabilities

Operational Condition	p_i	f_{z_i} [s]	Stress Range Interval*, Ds								
			5	15	25	35	45	55	65	75
1 ($m_{01} = 100$)	0.50	0.0556	0.118	0.276	0.282	0.189	0.091	0.033	0.009	0.002
2 ($m_{02} = 350$)	0.35	0.0909	0.035	0.098	0.142	0.160	0.155	0.133	0.103	0.072
3 ($m_{03} = 800$)	0.15	0.2000	0.016	0.045	0.071	0.090	0.102	0.107	0.105	0.097

* each 10 MPa stress range interval is identified by its mid point

The three data sets, presented as examples in Figure C.5.3.1 and Table C.5.3.1, represent the short-term Rayleigh distribution for three operational conditions that define the potential service conditions of a vessel. The weighted short-term distribution sum, used to develop a long-term distribution is completed, for n operational conditions as follows:

$$\bar{f}_z = \sum_{i=1}^n p_i \times f_{z_i} \quad (C.5.7)$$

$$\bar{f}_{z_i} = 0.0556(0.5) + 0.0909(0.35) + 0.2(0.15) = 0.0896$$

Table C.5.3.2: Weighted Short-Term and Resulting Long-Term Stress Range Interval Probabilities

Operational Condition	$\frac{p_i \cdot f_{z_i}}{\bar{f}_{z_i}}$	Stress Range Interval*, Ds								
		5	15	25	35	45	55	65	75
1 ($m_{01} = 100$)	0.310	0.0366	0.0856	0.0874	0.0586	0.0282	0.0102
2 ($m_{02} = 350$)	0.355	0.0124	0.0348	0.0504	0.0568	0.0550	0.0472
3 ($m_{03} = 800$)	0.33	0.0056	0.0151	0.0238	0.0301	0.0341	0.0358
Long-Term	--	0.0544	0.1355	0.1616	0.1155	0.1173	0.0932

* each 10 MPa stress range interval is identified by its mid point

With the probability of occurrence of Q' of each stress range interval estimated for the long term probability distribution (see Table C.5.3.2) in a discrete form, a cumulative distribution, Q , is obtained by summing stress range probabilities from the smallest stress range incrementally to the higher stress ranges such that the cumulative probability of each bin is equal to the sum of cumulative probabilities of all stress range bins up to the latest bin and the probability of occurrence of the latest bin. Ultimately, the value of the cumulative probability of the longest stress range bin should equal 1. Continuous representation is developed by fitting a Weibull distribution to this data. A least squares fit was used to establish the statistical parameters ($q = 36.5$ and $h = 1.62$) for the continuous Weibull distribution (Eq. C.5.3). This is shown in Figure C.5.3.2.

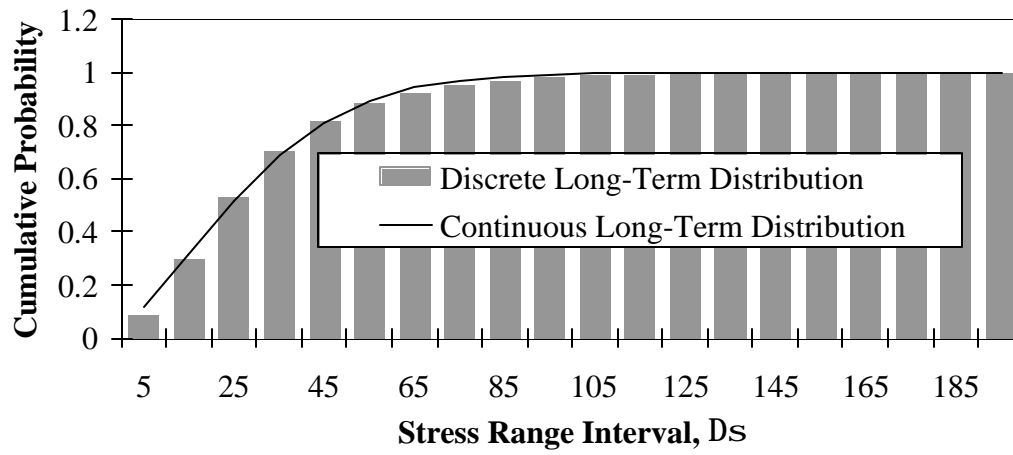


Figure C.5.3.2: Long-Term Stress Range Cumulative Probability Function

C.6 FATIGUE DAMAGE CALCULATIONS

C.6.1 Introduction

Once the expected stress ranges (including concentration factors) have been determined for any structural detail, the fatigue damage calculations can be undertaken. The standard approaches to this use the Miner-Palmgren linear damage summation model [Ref. C.31] to evaluate the cumulative damage (**D**) over the period selected, which is normally taken as the design lifetime of the ship.

This model assumes that the cumulative fatigue damage (**D**) is the sum of the fatigue damage inflicted by each stress cycle (d_i), independent of the sequence in which the stress cycles occur. The model further assumes that d_i can be mathematically expressed as $1/N_i$ where N_i is the average number of loading cycles to failure under constant amplitude loading at the i^{th} stress range.

$$D = \sum_{i=1}^{N_t} d_i = \sum_{i=1}^{N_t} \frac{1}{N_i} \quad (\text{C.6.1})$$

A damage sum exceeding unity is interpreted as a failure, although the meaning of this term is generally left somewhat vague. It can generally be interpreted as the generation of a through-thickness crack, several centimetres in length, in the detail under consideration. In practice, such a crack may or may not be of sufficient size to cause degradation of function (e.g., water tightness) or of unacceptable risk of fracture; i.e., a crack that needs to be repaired.

If the damage sum over N stress cycles is less than unity, the ratio of the projected fatigue life to the time for N cycles is D/N_{total} . Fatigue damage assessments can thus be used to rank structural details in the ship according to projected life, and develop inspection strategies that focus attention on the highest risk areas.

In order to derive the damage at each stress cycle, or at each stress range level, fatigue design (S-N) curves are used. These curves are derived from experimental data from fatigue tests conducted in air or in a simulated sea water environment, usually on relatively small specimens simulating simple welded details of various configurations. Derivation of the design S-N curves from the experimental data takes into account:

- differences in stress concentration factors between the structural detail and tested weld detail;
- adjustments to reflect desired safety factors (probabilities of failure);
- differences in the material thickness between the structural detail and the tested detail.

Other modifications may also be needed to account for fabrication processes, including the effects of mean stress levels, residual stresses and post-weld treatments such as grinding and peening. Obviously, if the design calculations assume certain “non-standard” factors, then the ship construction needs to ensure that these are actually incorporated in the fabrication processes.

Section 6.2 presents recommended fatigue design curves for use with the stress calculation methodologies described earlier, and a set of modification and correction factors which can be used in tailoring these to a specific application. Section 6.3 provides damage assessment calculation tools, which are standard mathematical applications of the recommended equations and distributions.

All aspects of the procedures are consistent with “state-of-the-practice” ship design. However, it should be noted that individual approval agencies (such as classification societies) may take different approaches to some or all issues. In addition, ongoing research in this area has recently led to changes in some of the equivalent codes and standards for offshore structures. Therefore, where a design is particularly sensitive to fatigue performance, the structural designer should consider undertaking additional background studies of the most recent research. The Triennial Proceedings of the International Ship and Offshore Structures Congress [Ref. C.32] can be a valuable starting point for such studies.

C.6.2 Fatigue Design Curves

C.6.2.1 Baseline (in Air)

Fatigue design or fatigue assessment procedures based on S-N curves can employ one of three different approaches:

- nominal stress approach;
- hotspot stress approach; and,
- notch stress approach.

The latter two are referred to as “local stress” approaches. All of these fatigue design approaches include the effects of: structural geometry, local detail geometry, weld toe (notch) geometry. The difference among the three approaches is how they capture these effects. It is possible to include these effects in the calculated stress range in terms of stress concentration factors (which increase the applied stress range) or to capture these effects in the S-N curve (which reduces the fatigue life). In principle, all three approaches should lead to similar outcomes if all of the effects are properly captured in the stress concentration factors and S-N curves. In practice, this may not happen due to various assumptions made in either deriving the design S-N curve or in calculating the applied stress range. The local stress (three) approaches evolved over time, to overcome the limitations of the nominal stress approach when applied to welded joints in complex structural details.

C.6.2.1.1 Nominal Stress Approach

Fatigue design procedures for welded plate-to-plate joints in steel bridges and steel offshore structures [Refs. C.33-C.38] classify basic welded joints subjected to uni-directional loading (axial or bending) into different categories of fatigue performance, and specify an S-N design curve for each category. The effects of welding residual stresses and stress concentrations in the classified joints are assumed to be built into the relevant design curves, so the fatigue design stress for a given joint is simply the nominal stress range at the anticipated crack initiation site. S-N design curves from the aforementioned procedures have been incorporated into fatigue design procedures recently introduced by certain classification societies and navies for welded joints in ship structures [Refs. C.1, C.2 and C.14].

However, the basic joints covered by the nominal stress approach are often embedded in complex structural details and sometimes located near openings and other structural discontinuities. The stresses within these details can be highly multi-axial even if the details are located in generally uni-directional nominal stress fields. Furthermore, the principal components of the actual stresses at a given point in a structural detail may be higher or lower than the nominal stress components. Therefore, when the aforementioned S-N design curves are applied to welded joints in ship structural details, the fatigue design stresses for the aforementioned situations should be based on the principal components of local nominal stresses. These stresses are the actual stresses at the anticipated crack initiation site, minus any stress concentrations in the basic joint (e.g., weld toes, weld reinforcement, attachments, eccentricity of lap joints), but including the global stress concentration of the structural detail and the global stress concentration effect of any nearby structural discontinuities (e.g., openings). Unfortunately, it is often difficult to classify a welded joint in a complex structural detail and to quantify the stress concentration effects that are not already built into the corresponding S-N curve, and, **therefore, the nominal stress approach is not recommended in this Guide.**

C.6.2.1.2 Notch Stress Approach

Det Norske Veritas [Ref. C.3] and Bureau Veritas (BV) have advocated the use of the notch stress approach. The notch stress range is defined as:

$$\Delta\sigma_{\text{notch}} = K_w \cdot \Delta\sigma_{\text{hotspot}} = K_g \cdot K_w \cdot \Delta\sigma_{\text{nom}} \quad (\text{C.6.2})$$

where $K_g \cdot K_w$ are the general detail and weld stress concentration factors, respectively (see Appendix C). Therefore, this includes the stress concentration effects of the local weld configuration and local notch geometry, as well as the overall stress concentration of the structural detail. The value of K_w can be computed from parametric equations or by fine mesh finite element analysis, but in either case, require a prior knowledge of the weld profile and toe geometry that may not always be known. DNV propose a default value of 1.5 for K_w , and it seems that design notch stress based fatigue design curve has been obtained from a hotspot stress fatigue design curve by adjusting the stress axis by a factor of 1.5. BV, on the other hand, provides a parametric equation for K_w that is a function of the weld toe angle and that can assume values from about 1.0 to 1.96 for toe angles varying from 12 to 45 degrees. The design S-N curve to be used in conjunction with the notch stress approach is that for the base material which seems to be a non-conservative approach as it does not take into account residual stresses at all. **The authors of this Guide do not see any inherent benefit in following the notch stress approach and therefore, it is not recommended in this Guide.**

C.6.2.1.3 Hotspot Stress Approach

The hot spot stress approach is widely used in fatigue assessment procedures for tubular joints in offshore structures [Refs. C.33, C.38 and C.39] and is being increasingly used as an alternative to the nominal stress approach for welded plate-to-plate joints in complex structural details [Ref. C.40]. It enables the fatigue lives of many types of structural details with nominally similar weld profiles, weld toe geometries and local joint configurations to be assessed using a single fatigue design curve.

The applicable stress range to be used in conjunction with the design hotspot stress S-N curve is the maximum principal stress range at the crack initiation site minus the stress concentrations of the weld reinforcement, weld toe geometry. The latter effects are built into the fatigue design curve, whereas the overall stress concentration of the structural detail must be explicitly included in the hotspot stress range. The hot spot stress range is defined as:

$$\Delta\sigma_{\text{hot spot}} = K_g \cdot \Delta\sigma_{\text{nom}} \quad (\text{C.6.3})$$

where K_g is the local detail (or structural) stress concentration factor.

For plate to plate joints, the hotspot stress fatigue design curve corresponds to the S-N curve for a transversely loaded butt weld (with reinforcement in place). Unfortunately, such S-N curves are not identical in various codes, standards or rules. For example, the American Bureau of Shipping's fatigue design procedures for tankers and bulk carriers [Refs. C.1, and C.41] suggest the use of the "Category E" nominal stress design curve (transversely loaded, full penetration butt weld made in positions other than down-hand or made using the submerged arc welding process or between plates of unequal width or unequal thickness) as the hotspot stress fatigue design curve to assess the fatigue strength of fillet-welded bracketed connections. The constant amplitude fatigue strength of "Category E" joints at 10^7 cycles is 47 MPa. Refs. C.42 and C.43, on the other hand, advocate the use of the S-N design curve for a better quality transverse butt weld to analyze fillet-welded joints, while the U.K. Department of Energy's widely referenced T-curve for hotspot stress analysis of tubular joints with full penetration fillet-like welds in offshore structures [Ref. C.44] nearly coincide with the aforementioned "Category D" curve. The constant amplitude fatigue strength of category D joints at 10^7 cycles is 53 MPa.

It is conceivable that the selection of the hot spot stress fatigue design curve is a reflection of the weld quality anticipated by the respective organizations. Since offshore structures and bridges are subject to extensive non-destructive inspection, it might ensure better quality welds warranting a category D hotspot fatigue design curve. In ships, on the other hand, non-destructive inspection is usually performed on a selective basis only and therefore selecting a category E curve for hotspot stress fatigue design may be prudent.

Notwithstanding these differences, **the hot spot stress approach is advocated in this Guide** because of its ease of application compared to the nominal or the notch stress approaches. The hotspot stress design fatigue curves, shown in Figure 6.2.1 for "air" and "sea water" environments, are adopted from the UK Health and Safety Executive (HSE) [Ref. C.45] which is believed to reflect the latest (1998) thinking on this subject.

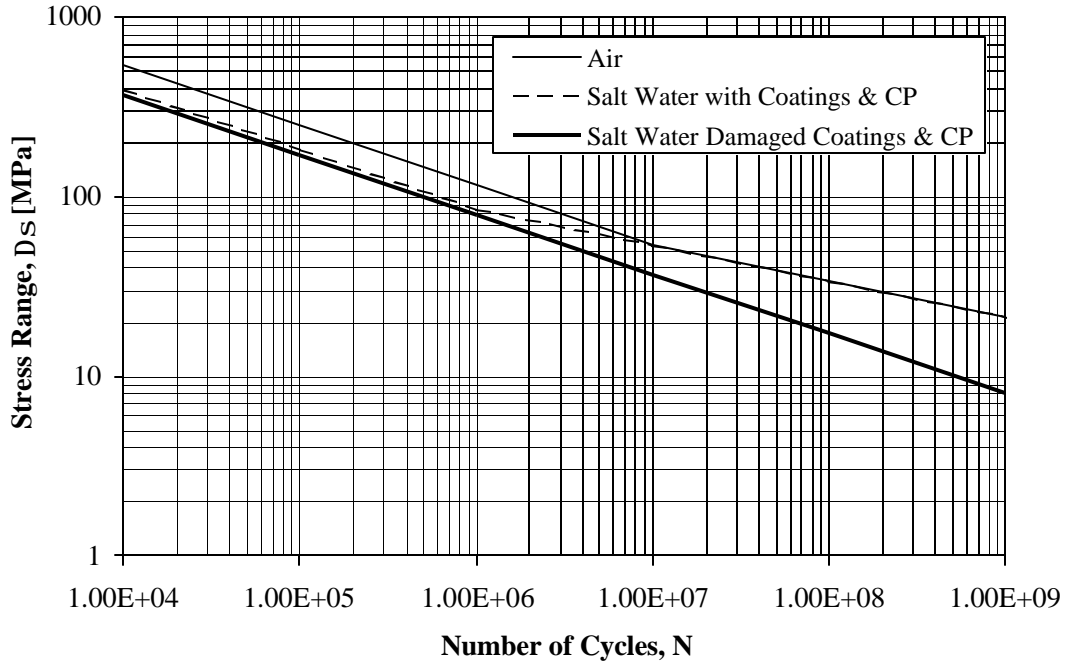


Figure 6.2.1: HSE Hot Spot Fatigue Design Curve for Plated as Welded Joints [Ref. C.45]

When compared to actual or implicit hotspot stress fatigue design curves adopted by various organizations (see Appendix A), the HSE curve is seen to be “in the middle of the pack”. For the air environment, the HSE hotspot stress design curve provides a bi-linear relationship between $\log N$ and $\log \Delta\sigma_{\text{hot spot}}$

for $N \leq 10^7$ cycle

$$\log N = \log a_1 - m_1 \cdot \log \Delta\sigma_{\text{hotspot}} = 12.182 - 3.0 \log \Delta\sigma_{\text{hotspot}} \quad (\text{C.6.3a})$$

for $N > 10^7$ cycle

$$\log N = \log a_2 - m_2 \cdot \log \Delta\sigma_{\text{hotspot}} = 15.637 - 5.0 \log \Delta\sigma_{\text{hotspot}} \quad (\text{C.6.3b})$$

where N is the fatigue life for a constant hotspot stress range $\Delta\sigma_{\text{hotspot}}$. The value of $\Delta\sigma_{\text{hotspot}}$ at the transition between the two sections of the bi-linear curve is 53 MPa.

The second part of the bi-linear curve reflects the reduced rate of damage accumulation at stress range levels below the 'fatigue limit' found in constant amplitude testing. A limit, as such, is not normally encountered in service experience with fluctuating stress ranges. Fracture mechanics theory, confirmed by experience, leads to the change in slope reflected by m_1 to m_2 .

This curve is generally valid for hotspot stress ranges greater than the nominal yield strength of steel plate (σ_{yield}), up to a maximum of $2\sigma_{\text{yield}}$ [Ref. C.31]. However, it should not be used if the largest principal component of the nominal stresses at the crack initiation site (i.e., the maximum stress excluding residual stresses and stress concentration effects of the weld toe, weld reinforcement, basic joint configuration, and overall stress concentration of the structural detail) exceeds 80% of the yield strength of the surrounding parent material. Neither of these limitations will normally be important to conventional ship designs.

S-N curves exist not only for fabricated details, but also for other types of structure including rolled sections, cut-outs of various types, etc. Normally in ship construction, fatigue will not be a problem in components that have not been processed to some degree, or at locations remote from such processing. However, if high fluctuating stresses are expected (for example) at the centre of a long, unsupported, rolled beam, the designer should refer to source documents such as Ref. C.33 for guidance on an appropriate curve. A type of detail where fatigue problems may be more likely in standard design is the thermal (flame) cut-out in plated structure. A wide range of fatigue tolerance can be found in such cut-outs, depending on the heat inputs and on the process controls. Equation C.6.3 will provide a conservative prediction of the fatigue life for all but poor quality cut-outs. If designers need to account for better quality (or post-treatment), then some guidance on relative performance can be found in Ref. C.46.

C.6.2.2 Corrosion and Corrosion Fatigue

The basic fatigue design curve in the section previous (Equation C.6.3) is for performance in a corrosion-free environment. In practice, general corrosion and localized corrosion (i.e., pitting) of unprotected steel are major sources of damage in ships, and the resulting reductions of net-section and stiffness can significantly increase cyclic stresses at welded details. Various corrosion fatigue mechanisms can also accelerate the initiation and propagation of fatigue cracks in welded structural details without proper protective coatings even if the details are cathodically protected.

To address this situation, it is normally accepted that all nominal stress levels should be calculated for net (i.e., end-of-life) scantlings. The generalized corrosion allowances applied to different types and areas of structures vary somewhat between classification societies but within fairly narrow ranges. ABS Rules [Ref. C.41] allow some account to be taken of the periods when the actual scantlings will remain above the minimum net (end of life) values by permitting reduction of general stress levels by a factor of 0.95. Although this type of further adjustment appears reasonable, its effects are minor in comparison with the uncertainties in other aspects of the procedure. **Therefore, this Guide recommends the basic, net scantling approach, as being consistent with most other approaches.**

As regards fatigue life, majority of codes and standards accept that unprotected joints exposed to sea water have their fatigue lives reduced by approximately a factor of two compared with those in air. Further, the fatigue design curves are no longer bi-linear, but instead retain their steeper gradient at lower stress ranges.

Some more recent data has suggested that the factor of two is non-conservative, and also that cathodic protection may not always provide effective fatigue corrosion protection. This is discussed in references [Ref. C.45 and C.47], and larger reductions in life (factors 2.5-3) are recommended under the UK Health and Safety Executive's latest guidance for offshore structures [Ref. C.48]. Following HSE again, the fatigue damage calculations for welded plate-to-plate details in the marine environment (submerged, splashed, or sprayed by sea water) are recommended to be based on the following relationships:

- with undamaged protective coatings and cathodic protection:

$$\begin{aligned} &\text{for } N \leq 1.026 \times 10^6 \text{ cycles} \\ &\log N = \log a_1 - m_1 \cdot \log \Delta\sigma_{\text{hotspot}} = 11.784 - 3.0 \log \Delta\sigma_{\text{hotspot}} \end{aligned} \quad (\text{C.6.4a})$$

$$\begin{aligned} &\text{for } N > 1.026 \times 10^6 \text{ cycles} \\ &\log N = \log a_2 - m_2 \cdot \log \Delta\sigma_{\text{hotspot}} = 15.637 - 5.0 \log \Delta\sigma_{\text{hotspot}} \end{aligned} \quad (\text{C.6.4b})$$

The value of $\Delta\sigma_{\text{hotspot}}$ at the transition between the two sections of the bi-linear curve is 84 MPa.

- without cathodic protection and proper protective coatings in a sea water environment (i.e., freely corroding joints):

$$\log N = \log a - m \cdot \log \Delta\sigma_{\text{hotspot}} = 11.705 - 3.0 \log \Delta\sigma_{\text{hotspot}} \quad (\text{C.6.4c})$$

The hotspot stress fatigue design curves represented by Equations C.6.4 for the sea water environment are included in Figure 6.2.1 shown earlier.

When structure can be assumed to be effectively protected over part of its life and unprotected over the remainder (e.g., for ballast tank coating systems with expected lifetimes of less than 20 years) then the fatigue life assessment can combine calculations using Equations C.6.3 and C.6.4. A simplified method for doing this is offered by DNV [Ref. C.3]

C.6.2.3 Probability of Failure

Equations C.6.3 are based on two standard deviations below the mean probability of test specimen failure at a calculated value of $D = 1$; i.e., a failure probability of approximately 2.3% at N_t lifetime cycles. This is regarded as acceptable for most ship structures for the level of “failure” implied. However, where a detail is critical to the safety of the ship and/or personnel, it will be advisable to apply an additional safety factor, either by choosing $D < 1$ or by adjusting the S-N curve used in the analysis. The selection of an alternative acceptable failure probability can make use of reliability analysis, as described in Section C.7.

The lifetime failure probability should also reflect an accurate assessment of the number of stress cycles at each stress range over the life of the ship, as discussed in Section C.5.

C.6.2.4 Thickness Effects

The inherent stress concentration factors for any type of detail, reflected in the S-N curve, may change with thickness due to differences in stress gradient and to residual stresses.

Therefore, it may be necessary to make a thickness correction above a value characteristic of the samples tested to develop the curve. The “classic” form of this correction is due to Gurney [Ref. C.49] in which, for fillet-welded joints with plate thicknesses exceeding a reference thickness t_r

$$\left(\frac{\sigma_{S-N}}{\sigma_c} \right) = \left(\frac{t_r}{t_s} \right)^{\frac{1}{4}} \quad (\text{C.6.5})$$

where:

t_s is the thickness of the plate through which fatigue cracks propagate;

$\Delta\sigma$ is the design stress range allowed by the relevant S-N curve; and,

$\Delta\sigma_c$ is the corrected design stress range.

This correction was subsequently incorporated into a number of fatigue design standards for steel bridges, offshore structures, and ships. Many of the classification society rules have t_r in the range of 22-25mm. However, HSE based on the latest available data, suggests that reference thickness should be 16mm, and that the exponent should be 0.3. Accordingly, when the thickness exceeds 16 mm, Equations C.6.3 and C.6.4 need to be modified to include the thickness correction that accounts for experimentally observed and theoretically predicted reductions in fatigue life of certain joints with increasing plate thickness (t):

$$\log N = \log a_1 - m_1 \cdot \log \Delta\sigma_{\text{hotspot}} - 0.3 \cdot m_1 \cdot \log(t/16) \quad (\text{C.6.6})$$

While the above correction is more conservative than that recommended in other codes and standards, it is compensated by more recent experimental and theoretical studies indicating that the correction is not required for longitudinal welds, transverse butt welds, and transverse fillet welds with weld stress concentrations comparable to that of butt welds (e.g., transverse fillet welds with smooth weld/base transitions achieved by grinding or special welding techniques).

C.6.2.5 Fabrication Considerations

While the in-air S-N curves have been used for steels with yield strength up to 690 MPa, the experimental data base in the sea water environments is based on steels with tensile yield strengths in the range of 300-500 MPa. There is currently little suitable public-domain data for higher-strength steels in a marine environment where there is potential for further deterioration in the fatigue performance due to increase in hydrogen embrittlement susceptibility with increase in strength. Up to 500 MPa the curves can be considered independent of material tensile strength [Refs. C.50 and C.51].

The weld toes of fatigue-prone structural details in ships are sometimes ground to improve the resistance to fatigue crack initiation. A factor of 2.2 (improvement) on the fatigue lives of welded details in air and cathodically protected welded details with proper protective coatings in a sea water environment can be achieved if grinding is used to produce a smooth concave weld profile, including the weld toe, which blends smoothly with the parent material. This correction can be implemented by adding $\log 2.2$ to the right-hand side of Equations C.6.3 to C.6.4. However, the effect will only be achieved with a high-quality weld and careful application of the grinding procedure. Adequate weld throat size must be retained and protective coatings must be properly applied before any corrosion can initiate.

It is known that ship structures experience residual stress relief during a “shake-down” period after they enter service. Weld toes, initially under considerable tension, may see this relaxed. This can mean that the ongoing stress cycles which they experience are fully or partly compressive, and thus significantly less damaging than purely tensile cycles. However, it is difficult to provide generic credit for this effect due to the lack of relevant data and the importance of local design.

If it can be demonstrated theoretically or experimentally that the sum of the applied stress range, fabrication restraint stresses, and welding residual stresses will be partly or completely compressive over a particular duration of the design life, then the effective stress range over that duration can be defined as the sum of the tensile part of the sum plus 60% of the compressive component. Otherwise, the effective stress range should be assumed to be equal to the applied stress range.

C.6.3 Damage Summation Model

Integration of the damage caused by the anticipated loads using the S-N curves recommended above can be done in a number of ways, depending on the nature of the stress data available and the level of accuracy required from the procedure.

C.6.3.1 Numerical Calculation

If the long-term Weibull distribution of hotspot stress ranges for the simplified method, or the sum of the short-term Rayleigh distributions of hotspot stress range for the full spectral analysis method is approximated by a histogram with k blocks of stress cycles of constant stress range, the damage inflicted by the distribution can be estimated as follows:

$$D \approx \sum_{i=1}^k d_i = \sum_{i=1}^k \frac{n_i}{N_i} \quad (\text{C.6.7})$$

where

- k = number of stress blocks
- n_i = number of stress cycles in stress block i
- N_i = number of stress cycles to failure at stress range $\Delta\sigma_i$

The number of blocks should be large enough to ensure reasonable numerical accuracy and should not be less than 20. Within each block, the stress range $\Delta\sigma_i$ selected should represent the mean stress range for that block [Ref. C.3].

C.6.3.2 Closed Form Solution for Level 2 Analysis

If the long-term distribution of hotspot stress ranges is defined by a Weibull distribution and if the fatigue design curve is a linear log N vs log $\Delta\sigma_{\text{hotspot}}$ relationship like Equation C.6.4, the cumulative damage (D) is defined by the following closed form solution:

$$D = \frac{\bar{f}_z t_d}{a} \sum_{n=1}^{N_{\text{load}}} p_n q_n^m \Gamma\left(1 + \frac{m}{h_n}\right) \quad (\text{C.6.8})$$

where

N_{load} = total number load conditions

p_n = fraction of design life in n^{th} load condition

t_d = design life of ship in seconds (20 years = 6.3×10^8 seconds)

h_n = Weibull shape factor for n^{th} loading condition

$$q_n = \text{Weibull stress range scale parameter for } n^{\text{th}} \text{ loading condition} = \frac{\Delta\sigma_o}{(\ln n_o)^{1/h}} \quad (\text{C6.9})$$

\bar{f}_z = average long term zero-crossing frequency (see Section C.5.3)

a and m = fatigue design curve parameters

$\Gamma(1+m/h_n)$ = gamma function

$\Delta\sigma_o$ = design stress range for the n^{th} loading condition

$1/n_o$ = the reference probability of exceedence on which $\Delta\sigma_o$ is based (typically 10^{-4})

If the long-term distribution of hotspot stress ranges is defined by a Weibull distribution and if the fatigue design curve is a bi-linear log N vs log $\Delta\sigma_{\text{hotspot}}$ relationship like Equation C.6.3, the cumulative damage (D) is defined by the following closed form solution

$$D = \bar{f}_z t_d \sum_{n=1}^{N_{\text{load}}} p_n \left[\frac{(q_n)^{m_1}}{a_1} \Gamma\left(1 + \frac{m_1}{h_n}; \left(\frac{? s_{\text{slope}}}{q_n}\right)^{h_n}\right) + \frac{(q_n)^{m_2}}{a_2} \Gamma\left(1 + \frac{m_2}{h_n}; \left(\frac{? s_{\text{slope}}}{q_n}\right)^{h_n}\right) \right] \quad (\text{C.6.10})$$

where:

a_1 and m_1 = the fatigue design curve parameters for $N \leq 10^7$ cycles

a_2 and m_2 = the fatigue design curve parameters for $N > 10^7$ cycles

$\gamma(;) =$ incomplete gamma function

$\Gamma(;) =$ complementary incomplete gamma function

$\Delta\sigma_{\text{slope}}$ = stress range at which change in slope occurs (See Sections C.6.1.1 and C.6.1.2)

Tables of full and incomplete gamma functions can be found in numerous standard texts, such as [C.6.59].

C.6.3.3 Cumulative Damage for Level 3 Analysis

If the long term stress range distribution is defined by a series of short-term Rayleigh distributions for different sea states and headings, and if the fatigue design curve is a linear log N and log $\Delta\sigma_{\text{hotspot}}$ relationship like Equation C.6.4, the cumulative damage is defined by the following closed-form solution:

$$D = \frac{\bar{f}_z t_d}{a} \Gamma\left(1 + \frac{m}{2}\right) \cdot \sum_{n=1}^{N_{\text{load}}} p_n \cdot \sum_{\substack{\text{all sea states} \\ \text{all headings} \\ i=1, j=1}} r_{ijn} \left(2\sqrt{2m_{0ijn}}\right)^m \quad (\text{C.6.11})$$

where

- r_{ij} = the relative number of stress cycles in short-term condition i, j (see Section C.5.3)
- m_{0ijn} = zero spectral moment of stress response process
- $\Gamma(1+m/2)$ = gamma function = 1.33 for $m = 3$

If the long term stress range distribution is defined by a series of short term Rayleigh distributions for different sea states and headings and if the fatigue design curve is a bi-linear log N and log $\Delta\sigma_{\text{hotspot}}$ relationship like Equation C.6.3, the cumulative damage is defined by the following closed-form solution:

$$D = \bar{f}_z t_d \sum_{n=1}^{N_{\text{load}}} p_n \cdot \sum_{\substack{\text{all sea states} \\ \text{all headings} \\ i=1, j=1}} r_{ijn} (B + C) \quad (\text{C.6.12})$$

$$B = \frac{\left(2\sqrt{2m_{0ijn}}\right)^{m_1}}{a_1} \Gamma\left(1 + \frac{m_1}{2}; \left(\frac{\Delta\sigma_{\text{slope}}}{2\sqrt{2m_{0ijn}}}\right)^2\right)$$

$$C = \frac{(q_n)^{m_2}}{a_2} \gamma\left(1 + \frac{m_2}{2}; \left(\frac{\Delta\sigma_{\text{slope}}}{2\sqrt{2m_{0ijn}}}\right)^2\right)$$

where: m_1 and m_2 are fatigue design curve first and second slopes respectively.

C.7 FATIGUE LIMIT STATE RELIABILITY ANALYSIS

A reliability analysis allows the effect of the variability in load and strength variables to be accounted for in the design process. In other words, a reliability analysis provides a vehicle for estimating the probability that a given mode of failure (e.g., fatigue, fracture, plastic collapse, etc.) will manifest in a given period of time. This analysis is based on the assumed variability of the parameters used to describe the structural system and its loading.

The following section is not intended as a comprehensive review of structural reliability analysis for ship structures. It is included here to make the reader aware that these techniques have been introduced and to provide a brief review. In general, structural reliability analysis may be completed by numerical simulation or by reliability index based approaches. In this section, only the reliability index based approaches are discussed.

C.7.1 Reliability Analysis

Class societies, the Ship Structure Committee and other organisations have developed reliability based design procedures [Refs. C.2, C.3, C.52 and C.54]. In general, a fatigue design procedure estimates the probability that a fatigue failure (p_f) will occur in a given structural element during a given time period. Failure is typically defined by the Miner's [Ref. C.31] summation usage factor (η) exceeding a value of unity in the time period of interest or the estimated fatigue life being shorter than the target design life. The approach presented below was formulated to allow an analytic solution by assuming that the loading is distributed according to a Weibull statistical distribution. For simplicity the fatigue assessment does not include a thickness correction. The physical basis for the fatigue failure limit state is expressed as follows:

$$p_f = N \frac{(Kq)^m}{a_{S-N}} - Q \left(1 + \frac{m}{h} \right) \quad (C.7.1)$$

where: η = fatigue usage factor predicting failure when $\eta = 1$

N = number of stress cycles within time t_d , t/\bar{T}

m = slope parameter of the S-N curve

a_{S-N} = scale parameter of the basic S-N curve for members with $K = 1$

K = stress concentration factor

h = slope parameter of the Weibull distribution for stress range

q = scale parameter for the Weibull distribution for stress range

t_d = design life

\bar{T} = long-term average stress period

In the above fatigue limit state equation not all of the parameters are considered random or uncertain. Those parameters which are considered random would be represented by a characteristic (e.g., mean or percentile) value and a measure of uncertainty (e.g., standard deviation or coefficient of variation = standard deviation/mean). It has been suggested that the slope of the S-N curve (m) should be considered a random variable, as well as, the S-N curve scale parameter (a).

The stress concentration factor (K) is generally considered uncertain and its statistical parameters may be estimated as a function of the statistical parameters of its component parts as shown in Equation C.7.2 below, by replacing $G(x)$ and x with $K(k)$ and k_i , respectively. The wave characteristic period (T) which along with the design life (t) determines the number of wave encounters or stress cycles (N), is considered uncertain and should be considered a random variable. As far as the magnitude of the environmental loading is concerned, it is suggested that the slope parameter of the Weibull stress, load or wave height distribution (h) be considered deterministic whereas, the scale parameter (q) should be considered random. It is commonly assumed that the uncertain (random) variables are statistically independent.

Typically a first order reliability method (FORM) based on a Taylor series expansion of the deterministic expression for the expected Miner's summation usage factor, is used to estimate the reliability or probability of fatigue failure. In this case the limit state equation $G(x)$ is $1 - \eta(x)$ and is used to estimate the probability of the usage factor exceeding unity in a given period of time (e.g., $P_f = \text{Prob}(G(x) = 1 - \eta(x) < 0)$). The mean and standard deviation of the limit state equation are estimated as:

$$\mu_G = G(\mu_x) \quad \text{and} \quad \sigma_G^2 = \sum_{i=1}^n C_i^2 \sigma_{x_i}^2 + \sum_{i=1}^n \sum_{j=1}^n \rho_{ij} C_i C_j \sigma_{x_i} \sigma_{x_j} \quad (\text{C.7.2})$$

$$\text{Where: } C_i = \frac{\partial G(x)}{\partial x_i} \quad \text{(the first partial derivative of the limit state function } G(x) \text{ with respect to one of the function variables, } x)$$

and ρ_{ij} is the correlation coefficient for variables i and j , which is typically assumed to be equal to zero based on an assumed independence of the variables, x_i and x_j .

The reliability against failure is estimated in terms of a reliability index (β):

$$\beta = \frac{\mu_G}{\sigma_G} \quad (\text{C.7.3})$$

The reliability index (β) is directly related to the reliability and probability of failure as follows:

$$P_f = 1 - \text{Reliability} = 1 - \Phi^{-1}(\beta) \quad (\text{C.7.4})$$

where Φ^{-1} is the inverse standard normal distribution function

A refinement of the first order technique described above, the design point approach, identifies a combination of the limit state function random variables which minimises β and thus maximises the probability of failure [Refs. C.56 to C.58].

C.7.2 Inherent Assumptions or Approximations in Structural Reliability Analysis

While reliability analysis is a useful tool to demonstrate the relative (i.e., comparison of design alternatives or scantlings) likelihood of failure or risk associated with a structural detail, the quality of its absolute numerical results are a function of the quality of the model and data it employs, as in all engineering calculations.

In performing a reliability analysis, the user should be aware of the sensitivity of the analysis results to the potential sources of error or assumptions made in the analysis process. A general structural reliability analysis assumes that:

- All sources of variability or uncertainty are represented in the analysis. If this is not the case, then the analyst has made a conscious decision to omit sources of variability that are considered of lesser significance or that are unquantifiable. Sources of variability that are difficult to quantify in a structural reliability analysis may include human behaviour or error, fabrication tolerances or defects and gross differences in structural configuration.
- The variability of the load and strength variables are accurately defined. A statistically significant sample or database of information is available to characterise the variable and develop a representative statistical distribution.
- The dependence of variables and/or modes of failure are understood, quantifiable and incorporated in the reliability analysis. The analyst may choose, as a numerical simplification, to neglect dependencies or assume complete dependence or independence, but should be aware of the effects of these assumptions.
- Since the limit state equation is used to define failure, it is essential that it accurately predicts the expected mode of failure. Some detailed reliability analysis work includes a random variable representing the uncertainty in the limit state equation based on a comparison of experimental and predicted behaviour.
- In a reliability analysis, the failure mechanism or mode of failure must be specified, therefore, it is important to ensure that the proposed mode(s) of failure correspond to the actual mode(s) of failure. For instance, a reliability analysis of a tension member based on a plastic collapse limit state would not be valid if reality the structural member fails in a brittle (fracture) mode.

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PART D - EXAMPLES

D.1 EXAMPLE 1: LEVEL 2 FATIGUE ANALYSIS SAMPLE APPLICATION

D.1.1 Objective

The objective of this example is to perform a Level 2 Fatigue Analysis for an opening in the deck of a conventional dry cargo bulk carrier. An illustration of the location of interest is shown in Figure D.1.1.1. Specifically, the location of interest is at the aft end of a midship cargo hatch coaming. The coaming is supported by stiffeners that intersect the deck plate. This location is susceptible to fatigue cracking and an estimation of the fatigue life will be completed.

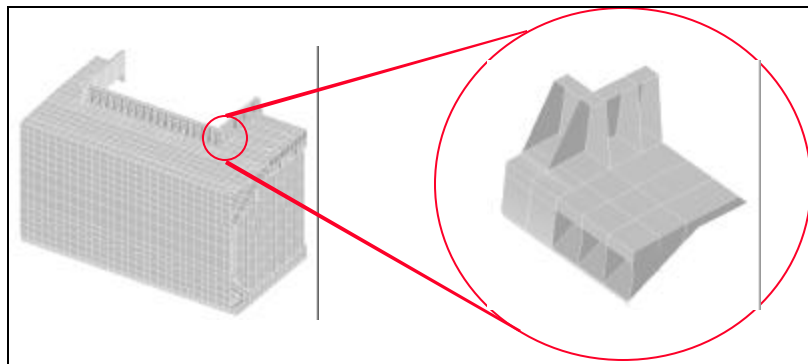


Figure D.1.1.1: Location of Interest for Fatigue Calculation

D.1.2 Assessment Approach

The approach followed in this example is summarized in seven steps:

- (1) Define problem data;
- (2) Estimate extreme characteristic loads;
- (3) Estimate service characteristic loads;
- (4) Estimate local nominal stresses;
- (5) Calculate hot spot stresses;
- (6) Define service stress range distribution; and
- (7) Perform fatigue damage summation.

D.1.2.1 Define Problem Data

The bulk carrier in this example is a ‘handy-size’ bulk carrier operating on short term contracts, (“tramp” service) which takes it all over the world. The vessel is rated at a service speed of 15 knots with its principle particulars outlined in Table D.1.2.1.

Table D.1.2.1: Vessel Characteristics

Principal Particulars:	
Length (L)	190.00 m
Beam (molded) (B)	27.60 m
Depth	14.80 m
Draft Max (SW)	10.93 m
Draft Max (TFW)	11.40 m
Displacement (SW)	47,043 tonnes
Displacement (TFW)	48,075 tonnes
Block Coefficient (C_B)	0.80
Deck Section Modulus (Z_{deck})	$2.398 \times 10^{10} \text{ mm}^3$

The structural geometry in the region of interest is shown in Figure D.1.2.1. The geometry in question consists of 12 mm deck plate, a coaming bracket and a longitudinal. The angle between the coaming bracket and deck plate is 80° degrees. The longitudinal runs underneath the plate in line with the coaming bracket.

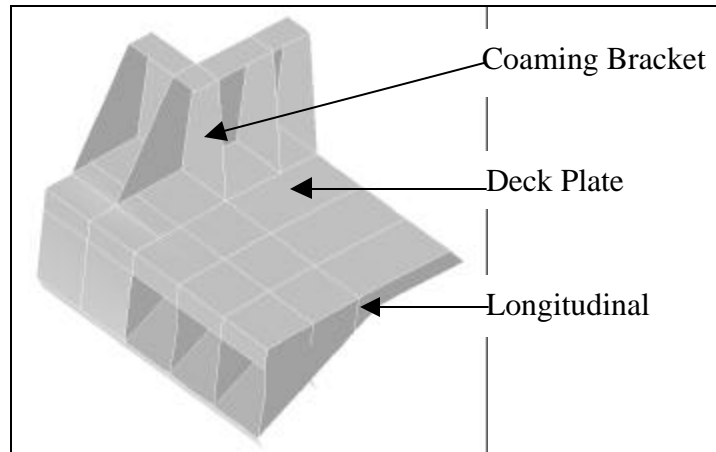


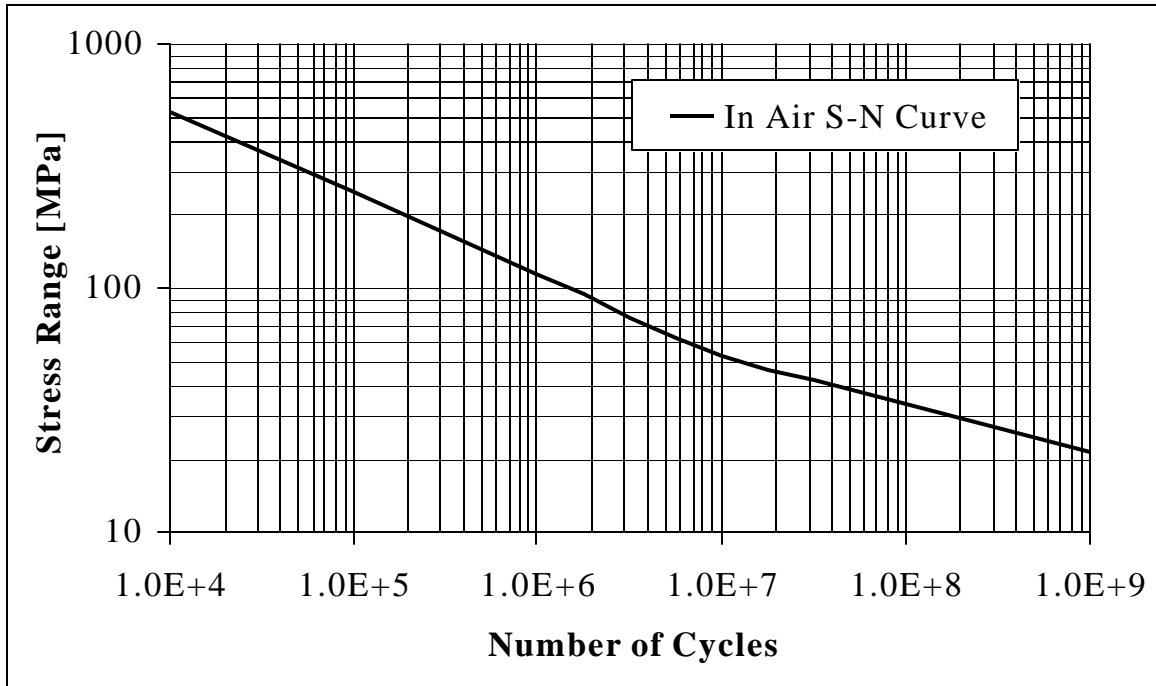
Figure D.1.2.1: Structural Geometry near Region of Interest

The structure is fabricated out of AH32 steel, which has the following material properties: yield strength 300 MPa and ultimate tensile strength 400 MPa. The hot spot stress life (SN) curve for the material in a non-corrosive environment is taken as:

$$\text{Log}(N) = \text{Log}(a) - m\text{Log}(\sigma_{\text{Hot Spot}}) \quad (\text{D.1.1})$$

Table D.1.2.2: Hot Spot SN Curve

	Log(a)	m
$N < 10^7$	12.182	3.0
$N > 10^7$	15.637	5.0

**Figure D.1.2.2: In Air Hot Spot SN Curve****D.1.2.2 Estimate Extreme Characteristic Load**

Both the maximum hogging and sagging moment are calculated for the location of the deck. These moments are determined through the class society rules. Since the crack is located on the deck, we can assume that the primary loading is due to hull girder bending, resulting in the following sagging moments:

$$M_{dS} = -0.11k_{wm} C_w L^2 B(C_B + 0.7) \quad [\text{kNm}]$$

$$C_w = 10.75 - \left[\frac{300 - L}{100} \right]^{3/2} \quad \text{for } 100\text{m} < L < 300\text{m} \quad (\text{D.1.2})$$

$$k_{wm} = 1.0 \quad (\text{between } 0.4L \text{ and } 0.65L)$$

Substituting for the above constants results in:

$$C_w = 10.75 - \left(\frac{300-190}{100} \right)^2 = 9.596$$

$$M_{dS} = -0.11(1.0)9.596(190)^2 27.6(0.8 + 0.7)$$

$$M_{dS} = -1,577,577 \text{ kNm}$$

Similarly the hogging moment can be calculated from:

$$M_{dH} = 0.19 k_{wm} C_w L^2 BC_B \quad [\text{kNm}]$$

$$M_{dH} = 0.19 (1.0) 9.596 (190)^2 27.6(0.8) \quad (\text{D.1.3})$$

$$M_{dH} = 1,453,283 \text{ kNm}$$

D.1.2.3 Estimate Service Characteristic Loads

The above calculated maximum hogging and sagging moments are based on the extreme event with an assumed probability of exceedance of 1×10^{-8} , while it has been demonstrated that the majority of fatigue damage results from smaller wave events (i.e., probability of exceedance in the 1×10^{-4} range. If the loads are assumed to be distributed according to a Weibull distribution, then the conversion of the characteristic bending moment from the extreme event to the typical service level of 1×10^{-4} is performed based on the ratio:

$$f_f = \frac{M_{\text{service}}}{M_{\text{Extreme}}} = \left[\frac{\ln p_2}{\ln p_1} \right]^{1/h} \quad (\text{D.1.4})$$

$$\text{where: } h = 2.21 - 0.54 \log(L) = 2.21 - 0.54 \log(190) = 0.9795$$

Substituting the data gives:

$$f_f = \left[1 - \frac{\ln(10^{-4})}{\ln(10^{-8})} \right]^{1/0.9795}$$

$$f_f = 0.4928$$

Table D.1.2.3: Extreme and Service Bending Moments

Moment	Extreme (kNm)	Service (kNm)
Sagging	-1,577,577	-777,430
Hogging	1,453,283	716,178

D.1.2.4 Estimate Local Nominal Stress

The nominal stress at the deck in a region remote from the opening can be determined by simply dividing the service bending moments by deck section modulus (Z_{deck}) from Table D.1.2.1. The local nominal stress includes the stress concentration caused by the hatch opening. The local nominal stress can be determined through:

$$s_{nom} = \frac{K_G M}{Z_{deck}} \quad (D.1.5)$$

where M is the service bending moment, K_G is the global stress concentration. The global stress concentration can be determined from Figure D.1.2.3. From this figure, a K_G factor of 5.0 is estimated for this region. However, the peak stress concentration coincides with the deck plate hatch opening corner illustrated in Figure D.1.2.4. The location of interest is approximately 900 mm away from this location, therefore a reduction in the K_G is substantiated based on the distance from the maximum stress concentration to the location of interest. Therefore at the location of interest, a K_G factor of 2.0 is suggested based on the above discussion.

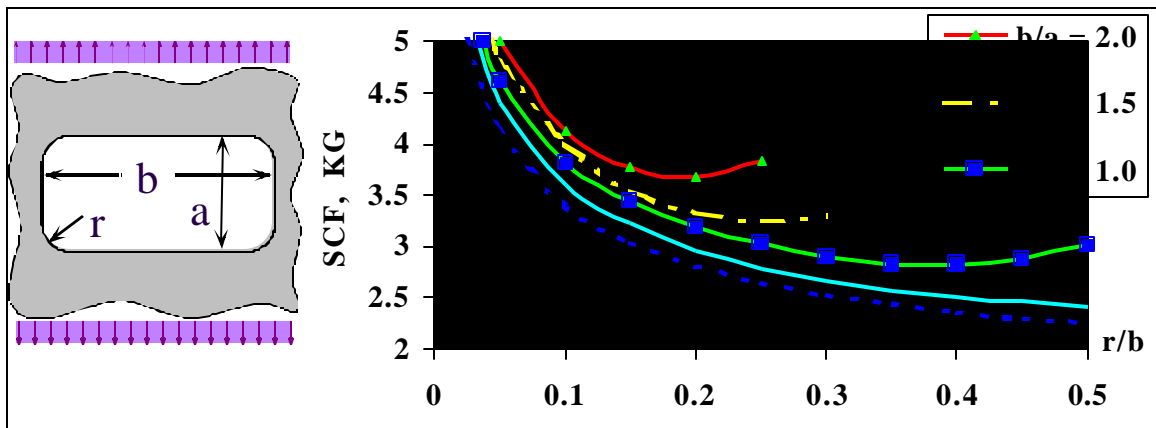


Figure D.1.2.3: Global Stress Concentration (K_G)

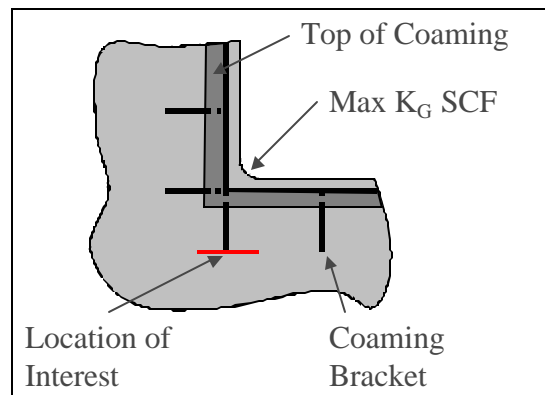


Figure D.1.2.4: Location of Peak Stress Concentration

The local nominal stress is computed from equation D.1.5 as

$$s_N = \left[\frac{2M \times 10^6}{1.174 \times 10^{10}} \right]$$

$$s_N = 1.7036 \times 10^{-4} M$$

D.1.2.5 Determination of Hot Spot Stress

The hot spot stress includes the geometrical details and effects due to local geometrical configuration such as misalignment, angular distortion, etc. In this example, an idealized welding geometry is used with no misalignment (i.e. $K_{te}, K_{t\alpha} = 1$).

$$s_{HotSpot} = K_g K_{te} K_{t\alpha} s_{nom} \quad (D.1.6)$$

The local stress concentration caused by the coaming bracket angle (K_g) can be determined from Equation (D.1.7), and the variables are shown graphically in Figure D.1.2.5

$$K_g = \frac{4}{3} \left[1 + \frac{t_w}{160 t_s} \right]$$

$$K_g = \frac{4}{3} \left[1 + \frac{(12)(80)}{(160) t_s} \right] \quad (D.1.7)$$

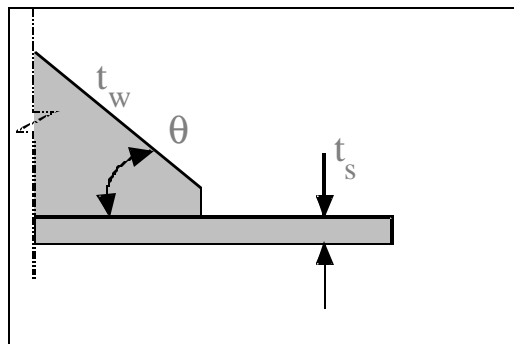


Figure D.1.2.5: Coaming Toe Attachment

Two cases are considered in this example, with and without a doubler plate, where the thickness of the plate (t_s) is changed.

Table D.1.2.4: Hot Spot Stresses at Location of Interest

Load Condition	Nominal Stress	Hot Spot	
		Doubler	No Doubler
		$t_p=24$ mm	$t_p=12$ mm
		$K_g=5/3$	$K_g=2$
	σ_{nom}	$\sigma_{hot\ spot}$	$\sigma_{hot\ spot}$
Hogging	59.7 MPa	99.5 MPa	119.4 MPa
Sagging	-64.8 MPa	-108 MPa	-129.6 MPa

D.1.2.6 Define Service Stress Range Distribution

The stress range is based on the difference between the hogging and sagging conditions. Therefore the hot spot stress range at the 'service probability' of 10^{-4} encounters can be determined as the algebraic difference between the hog and sag conditions.

$$\begin{aligned} \Delta \sigma_{hot\ spot} &= \sigma_h - \sigma_s \\ \Delta \sigma_{hot\ spot} &= 99.5 - (-108) = 207.5 \text{ MPa (doubler case)} \end{aligned} \quad (D.1.8)$$

Similarly, $\Delta \sigma_{hot\ spot} = 249.0$ MPa for the no doubler case.

The stress range distribution is assumed to follow a Weibull distribution of the following form:

$$\begin{aligned} Q(\Delta \sigma) &= \exp\left(-\left[\frac{\Delta \sigma}{q}\right]^h\right) \quad (\text{see Eq. C.5.3}) \\ q &= \frac{\Delta \sigma_{hot\ spot}}{[\ln(n_o)]^{1/h}} \end{aligned} \quad (D.1.9)$$

where $h = 0.9795$ (from before), and in this example, q for the doubler case is calculated as:

$$q = \frac{207.5}{[\ln(10^4)]^{1/0.9795}} = 21.5 \text{ MPa}$$

where $n_o = 1/\text{characteristic encounter probability}$
 $= 1/10^{-4}$

and $Q(\Delta \sigma)$ is the probability of exceedance for the stress range ($\Delta \sigma$). Thus $Q(\Delta \sigma)$ for doubler case has the form:

$$Q(\sigma) = \exp\left(-\left[\frac{\sigma}{21.5}\right]^{0.9795}\right) \quad (D.1.10)$$

Similarly, $\sigma = 25.8$ MPa for the case without a doubler plate.

D.1.2.7 Perform Fatigue Damage Summation

The fatigue damage is assessed using the Miner-Palmgren linear damage model. This model assumes that the cumulative damage is the sum of individual cyclic damage and that the damage is independent of the stress cycle sequence. The damage summation is in the following form (see Eq. C.6.7):

$$D = \sum d_i = \sum \frac{n_i}{N_i} \quad (D.1.11)$$

Two methods are described for the damage summation, a discrete and closed form solution approach.

D.1.2.7.1 Discrete Approach

In this approach, the stress range distribution is broken down into a histogram of at least 20 bins. The fatigue life calculation uses the mid range value for each bin. The fatigue damage d_i resulting from each stress range magnitude or bin is determined and then the total damage is calculated as the summation of the d_i 's, as follows.

Table D.1.2.5: Sample Discrete Fatigue Damage Summation (Doubler Case)

Stress Range Mid Value, $\Delta\sigma$ [MPa]*	Stress Range Probability, p_i †	Number of Cycles per Year, n_i †	Cycles to Failure, N_i ‡	Damage Ratio, $d_i = n_i / N_i$
10	6.06E-01	3030410	4.34E+10	6.99E-05
30	2.35E-01	1173067	1.78E+08	6.58E-03
50	9.43E-02	471272.6	1.39E+07	3.40E-02
70	3.83E-02	191609.9	4.43E+06	4.32E-02
↓	↓	↓	↓	↓
450	2.54E-09	0.012713	1.67E+04	7.62E-07
470	1.08E-09	0.005397	1.46E+04	3.69E-07
			Sum	1.98E-01

(No Doubler Case)

Stress Range Mid Value, $\Delta\sigma$ [MPa]*	Stress Range Probability, p_i ‡	Number of Cycles per Year, n_i †	Cycles to Failure, N_i ¥	Damage Ratio, $d_i = n_i / N_i$
10	5.41E-01	2706253	4.34E+10	6.24E-05
30	2.44E-01	1218098	1.78E+08	6.83E-03
50	1.13E-01	567131.2	1.39E+07	4.09E-02
70	5.34E-02	266860.2	4.43E+06	6.02E-02
↓	↓	↓	↓	↓
450	5.25E-08	0.262716	1.67E+04	1.57E-05
470	2.57E-08	0.128301	1.46E+04	8.76E-06
Sum				3.53E-01

* All probable stress ranges are captured in 20 MPa stress range intervals (i.e. 0 to 20 mid value = 10, 20 to 40 mid value = 30, etc.)

‡ Probabilities for each stress range interval is estimated based on the difference between the probability of exceedance of its upper and lower bounds using equation D.1.10 (i.e. $p_i = Q(\Delta\sigma_{lower}) - Q(\Delta\sigma_{upper})$)

† Assuming a 20 year design life with 10^8 wave encounters, the number of cycles per year for each stress range is calculated as the product its probability and the number of wave events per year (i.e., $10^8 p_i / 20$).

¥ Cycles to failure are calculated using equation D.1.1 or using Figure D.1.2.2.

D.1.2.7.2 Closed Form Solution

Alternatively, a closed form solution can be obtained for this problem. This solution has the benefit that the stress distribution is not discretized into a series of bins the solution process is continuous along its domain. The closed form solution is of the following form (equation C.6.10):

$$D = \overline{f_z} t_d \left[\frac{(q)^{m_1}}{a_1} \Gamma \left(1 + \frac{m_1}{h}; \left(\frac{? \text{ slope}}{q} \right)^h \right) + \frac{(q)^{m_2}}{a_2} \Gamma \left(1 + \frac{m_2}{h}; \left(\frac{? \text{ slope}}{q} \right)^h \right) \right] \quad (D.1.12)$$

Where: a_1 and m_1 = the fatigue design curve parameters for $N \leq 10^7$ cycles (Table D.1.2.2)

a_2 and m_2 = the fatigue design curve parameters for $N > 10^7$ cycles (Table D.1.2.2)

$\gamma(\cdot)$ = incomplete gamma function

$\Gamma(\cdot; \cdot)$ = complementary incomplete gamma function

$\Delta\sigma_{\text{slope}}$ = stress range at which change in slope occurs

h and q = Weibull distribution parameters

$\overline{f_z}$ = average zero crossing period

t_d = design life in seconds (20 years = 6.3×10^8 s)

D.1.2.7.3 Fatigue Life

Regardless of the solution approach used for the linear damage summation the same results should be obtained.

Table D.1.2.6: Cumulative Damage Results

Portion of Fatigue Life used in a Year	No Doubler	With Doubler
Discretized Approach	0.353	0.198
Closed Form Solution	0.260	0.143

The fatigue life estimates can be determined from the inverse of the damage summation ($1/D$). If we assume that the vessel is in operation only 85% of the time, the expected fatigue life is ($1/0.85D$):

Table D.1.2.7: Fatigue Life Estimates

	No Doubler [Years]	With Doubler [Years]
Discrete	3.3	5.9
Closed Form	4.5	8.2

The results of this example indicate that fatigue cracking will initiate or become visible after a service life of four to eight years. This analysis does not indicate that the predicted fatigue cracks will grow to a critical size in the estimate time period.

It should be noted that the Level 2 approach presented here, includes several conservative assumptions having a cumulative effect. In general, the rule based load estimation is conservative and so is the rule based stress analysis with analytic stress concentration factors. For example, the bracket toe detail geometric stress concentration factor did not include the presence of deck longitudinals which would increase the bending stiffness of the plate locally and thus reduce the stress concentration factor.

A more detailed or precise analysis of the scenario described in this example could be completed by performing a Level 3 analysis, similar to that presented in Section D.2. This example develops a load history based on the expected operation of the vessel. In addition, the behavior of the vessel structure is more accurately considered using finite element analysis.

D.2 EXAMPLE #2 - LEVEL 3 ANALYSIS

Problem Statement:

Perform a Level 3 Fatigue Analysis for an opening in the deck of a conventional dry cargo bulk carrier, as shown in Figure D.1.1.1 (previous section). As in the last example, the location of interest was at the aft end of a midship cargo hatch coaming. The coaming is supported by stiffeners that intersect the deck plate. In the context of the last example, the spectral approach is being employed to produce a more realistic estimate of the fatigue life.

D.2.1 Assessment Approach

The approach that was applied is summarized in six steps:

- (1) Establish the operational profile;
- (2) Calculation of loads;
- (3) Determine reference hot spot stress ranges;
- (4) Define short-term stress range distribution;
- (5) Define long-term stress range history;
- (6) Perform fatigue damage summation.

D.2.2 Establish Operational Profile

D.2.2.1 Overview

For this analysis, an operational profile of the vessel needs to be established. Whenever a ship is at sea, its “operating mode” is defined by its speed, heading relative to the waves, and the sea conditions encountered by the vessel. The operational profile is a compilation of all individual operating modes, covering the entire operational envelope of the vessel and is usually presented in the form of a probability of speed and heading at prescribed sea conditions.

D.2.2.2 Vessel Description

The analysis was conducted for the same vessel used in Example 1 (see Section D.1.2.1).

D.2.2.3 Establish Operational Profile

D.2.2.3.1 Establishing Ship Transit Profile

Operational profiles were developed from the operational data recorded in the ship's log covering a one-year period. The encountered wave climates were recorded in the ship's log in the terms of Sea State, and so the operational profile was developed on this basis.

For a detailed description of the development of the profile and joint probability distributions from the logs and examples, see Reference D.2.

The resulting distributions of Speed with Sea State and Heading with Sea State are shown in Tables D.2.2.1 and D.2.2.2. The route of the vessel is transcribed into the time spent in each Marsden Zone in Table D.2.2.3, and the time spent in each loading condition and in port is given in Table D.2.2.4.

Table D.2.2.1: Joint Probability of Speed and Sea State

		Sea State						
Speed (kn)	1	2	3	4	5	6	7	SUM
10-12	0.0078	0.0000	0.0000	0.0072	0.0130	0.0326	0.0148	0.0755
12-14	0.0305	0.0315	0.2140	0.2112	0.1957	0.1509	0.0000	0.8339
14-16	0.0000	0.0196	0.0261	0.0285	0.0067	0.0097	0.0000	0.0906
SUM	0.0383	0.0511	0.2401	0.2470	0.2155	0.1932	0.0148	1.0000

Table D.2.2.2: Joint Probability of Relative Heading and Sea State

		Sea State						
Rel. Heading	1	2	3	4	5	6	7	SUM
Head Seas	0.0066	0.0064	0.0316	0.0338	0.0313	0.0319	0.0148	0.1564
Strbd. Bow	0.0212	0.0187	0.1154	0.1191	0.1124	0.1022	0.0000	0.4890
Strbd. Beam	0.0038	0.0114	0.0365	0.0371	0.0268	0.0224	0.0000	0.1380
Strbd. Quart.	0.0027	0.0071	0.0250	0.0252	0.0191	0.0157	0.0000	0.0948
Following	0.0034	0.0099	0.0322	0.0327	0.0238	0.0199	0.0000	0.1218
SUM	0.0378	0.0534	0.2407	0.2479	0.2134	0.1921	0.0148	1.0000

Table D.2.2.3: Listing of Geographical Zones (Marsden Zones) ship traversed and relative time spent in each zone.

Marsden Zone	Time in Zone	Marsden Zone	Time in Zone	Marsden Zone	Time in Zone
5	0.0140	28	0.0290	59	0.0140
11	0.0140	29	0.0300	60	0.0340
13	0.0280	30	0.0690	61	0.0340
16	0.0150	32	0.0070	62	0.0280
17	0.0070	33	0.0140	66	0.0140
18	0.0140	36	0.0270	67	0.0340
19	0.0070	37	0.0350	68	0.0350
20	0.0480	39	0.0050	69	0.0070
21	0.0620	40	0.0500	75	0.0360
22	0.0200	47	0.0210	84	0.0340
25	0.0070	50	0.0350	85	0.0280
26	0.0340	56	0.0280	90	0.0280
27	0.0340	58	0.0210	~	
sum:	0.3036	sum:	0.3706	sum:	0.3258
				total sum:	1.0000

Table D.2.2.4: Operational Profile Summary

	Total Operation Time (hours)	Number of Data Entries
Loaded Condition	3624.4	151
Ballast Condition	1563.6	66
Time in Port	3552.0	~
Total	8739.8	~

This procedure results in the distribution of time spent by the vessel in various geographical zones, and the distribution of headings and speeds with various sea conditions. The following is a commentary on the development of the sea operational profiles:

- a) The resulting probabilities in an operational profile will depend on the quality and quantity of the recorded operational states aboard the vessel. For example, vessel speed, heading, and environmental conditions can be collected on a 10 minute, hourly, four-hourly (watch) or, sometimes, daily basis. Further, the actual measurements are prone to errors, as the majority of the measurements except ship speed are based on visual observations.

- b) For a vessel engaged in “tramp” service or otherwise irregular routes (such as in this example), it is desirable to obtain records over a long duration (several years). This permits assessment of the variability in data and will improve the statistical significance of the resulting probabilities. In this example, the data is representative of only one year in the ship’s life.
- c) The probability tables indicate the relative amount of vessel time at sea in each operational state (e.g., speed/Sea State, or relative heading/Sea State or transiting a Marsden Zone). As indicated, the sum of all of the operational state probabilities is equal to one. This means that the “volume” under the three-dimensional surface plot of probability (Speed; Sea State) or probability (Rel. Heading; Sea State) is equal to unity.
- d) Although heading information for the ship was recorded in degrees and covered 360 degrees, in the development of the vessel relative heading/Sea State probability table, the headings were discretized into five headings. This discretization scheme was selected for two reasons. First, headings relative to wave direction more precise than 45 degrees are difficult to determine consistently, and thus are not generally recorded in the ship logs. Second, by assuming port/starboard symmetry of vessel response to incident waves, pair-wise groupings of relative headings was possible:

- Head Seas
 - Strbd. Bow same as Port Bow
 - Strbd. Beam same as Port Beam
 - Strbd. Quartering same as Port Quartering
- Following Seas

In a statistical sense, this meant that the probabilities of pair-wise groups (e.g., Strbd. Bow and Port Bow, etc.) were simply added together to yield overall probability of “Strbd. Bow” entry, as shown in the Operational Profile Table D.2.2.2

- e) The operational profile was developed from the ship’s log. Voluntary speed reduction and/or change in course to avoid bad weather is a common practice and is left to the discretion of the master. The data thus may be biased in this regard. If the ship reduced speed and/or changed heading reduce motions during heavy weather, this was presumably recorded in the ship log, and therefore it is reflected in the operational profile.
- f) The source for the environmental data used in this analysis was the Global Wave Statistics, [Ref D.3]. Over 55 million visual observations of wind speed and direction, as well as wave height, period, and direction, obtained by the United Kingdom Meteorological Office marine data bank, were used to compile wave statistics for 104 zones (sometimes referred to as a Marsden Zones).

- g) In this analysis, for each Marsden Zone, the wave statistics are presented on an annual basis as well as for uni-directional seas only. Wave data has been normalized to approximately 1000 observations.

D.2.2.3.2 Transposing Sea States and Wave Heights

Wave statistics in Marsden Zones are given in terms of significant wave heights, and zero crossing periods [Ref. D.3]. In this example, the operational profile data is defined in terms of Sea State. Therefore a transformation of wave statistics from wave heights to Sea State is required. The transformation consists of the following steps:

- Assume that the significant wave height follows a three-parameter Weibull cumulative probability distribution given by:

$$F(H_S) = 1 - e^{-\left[\frac{(H_S - m)}{q}\right]^h} \quad (D.2.1)$$

where m is the location parameter, q is the scale parameter, and h is the shape parameter of the Weibull distribution.

- Taking the natural logarithm of both sides of the equation D.2.1 twice, it can be reduced to an equation of a straight line if plotted on the $\ln-\ln$ versus \ln scale.
- The wave data is then plotted, and the line that fits the data best is drawn using linear regression analysis.
- The parameter m is found by iteration, so that the regression error is minimized. The parameters h and q are determined from the slope and intercept of the line.
- With the cumulative probability distribution function calibrated, the probabilities of Sea States are calculated as

$$f(\text{Sea State}) = F(H_{S2}) - F(H_{S1}) \quad (D.2.2)$$

Note that this transformation from wave heights to Sea States was necessary only because the operational profile was recorded on the basis of Sea State. For operational profiles based on wave height this step is not necessary.

D.2.2.3.3 Calculation of Composite Wave Scatter Diagram

The composite probability distribution of wave heights and periods for all Marsden Zones combined is calculated to produce a “Composite Marsden Zone” scatter diagram. The joint probability of each significant wave height/zero crossing period combination is multiplied by the appropriate entry in the Table D.2.2.3 of the Operational Profile Data for n zones in which ship operates. The results are then summed for appropriate combinations of wave height and zero crossing period.

This is expressed in terms of an equation as:

$$f_{mc}(H_S;T_Z) = \sum_{i=1}^N \mu_i f_{mi}(H_S;T_Z) \quad (D.2.3)$$

and is provided as a sample calculation in the following table:

Table D.2.2.5: Wave Data by Marsden Zone [from Ref D.3]

H _s (m)	5<T _z <6 seconds			
	Marsden Zone. #5	Marsden Zone. #11	Marsden Zone#...	Marsden Zone. #90
0-1	0.0790	0.0940	...	0.0080
1-2	0.1400	0.1210	...	0.0130
2-3	0.0700	0.0630	...	0.0040
3-4	0.0230	0.0270	...	0.0010
4-5	0.0060	0.0110	...	0.0000
5-6	0.0010	0.0040	...	0.0000
6-7	0.0000	0.0020	...	0.0000
7-8	0.0000	0.0010	...	0.0000
8-9	0.0000	0.0000	...	0.0000
9-10	0.0000	0.0000	...	0.0000
10-11	0.0000	0.0000	...	0.0000
11-12	0.0000	0.0000	...	0.0000
12-13	0.0000	0.0000	...	0.0000
13-14	0.0000	0.0000	...	0.0000
>14	0.0000	0.0000	...	0.0000
SUM	0.3190	0.3230	...	0.0260

The data from Table D.2.2.3 and Table D.2.2.5 are combined via the equation above. For example,

$$f_{mc}(H_S=0-1\text{ m}, 5 < T_Z < 6\text{ s}) = 0.0790 \cdot 0.014 + 0.0940 \cdot 0.014 + \dots + 0.0 \cdot 0.0080 = 0.0524$$

where 0.014, 0.014, and 0.0280 are relative time spent in the zones 5, 11 and 90 respectively (Table D.2.3). Entry of 0.0524 appears in the Composite Marsden Zone Scatter Diagram (Table D.2.6) for H_s = 0-1 m and 5<T_z <6 seconds.

In this example the ship traversed 38 Marsden Zones, and a typical scatter diagram for each Marsden Zone is divided into 11 zero crossing periods (e.g., $T_z < 4$ sec., $4 < T_z < 5$ sec., $5 < T_z < 6$ sec., etc.). The above process is repeated for all the wave height ranges and crossing periods and the result is a Composite Marsden Zone Scatter Diagram (**Table D.2.2.6**). This diagram is unique for the example operational profile. Different profiles would result in a different Composite Marsden Zone scatter diagram

Table D.2.2.6: Composite Marsden Zone Scatter Diagram

H _s (m)	T _z (sec.)											SUM
	<4	4-5	5-6	6-7	7-8	8-9	9-10	10-11	11-12	12-13	>13	
0-1	19.5	51.3	52.4	29.7	10.8	2.7	0.4	~	~	~	~	166.8
1-2	5.5	38.5	84.5	96.1	68.5	31.9	10.3	2.4	0.5	~	~	338.1
2-3	1.1	11.6	36.4	60.8	67.0	47.8	22.8	7.7	2.0	0.5	~	257.7
3-4	0.3	3.0	11.3	22.5	32.3	31.1	19.6	8.6	2.9	0.8	0.1	132.3
4-5	0.0	0.8	3.4	7.4	12.2	14.6	11.5	6.1	2.3	0.8	0.2	59.3
5-6	~	0.2	1.0	2.6	4.7	6.2	5.6	3.5	1.4	0.5	0.2	25.8
6-7	~	0.0	0.3	0.9	1.7	2.5	2.6	1.7	0.9	0.4	0.1	11.1
7-8	~	0.0	0.2	0.3	0.7	1.0	1.2	0.9	0.6	0.2	0.0	5.1
8-9	~	~	0.0	0.1	0.3	0.4	0.6	0.5	0.3	0.0	~	2.3
9-10	~	~	~	0.0	0.1	0.2	0.2	0.2	0.2	0.0	~	0.9
10-11	~	~	~	~	~	0.0	0.2	0.1	0.0	~	~	0.3
11-12	~	~	~	~	~	~	~	0.0	~	~	~	0.0
12-13	~	~	~	~	~	~	~	~	~	~	~	0.0
13-14	~	~	~	~	~	~	~	~	~	~	~	0.0
>14	~	~	~	~	~	~	~	~	~	~	~	0.0
SUM	26.4	105.4	189.5	220.4	198.3	138.5	74.8	31.7	11.2	3.2	0.5	999.8

It can be seen that the sum of all probabilities is not equal to 1000. The small difference is due to rounding errors and is not significant. It will also be noted that the values obtained from the application of equation C.3.1 have been multiplied by 1000. Table D.2.2.6 then represents a single (composite) wave data set for the encounter probabilities of waves for this specific route.

As noted earlier, to calculate the joint speed/ heading wave probabilities it is necessary in this case to convert Sea state data from the ship's operational data set, to wave height data as used in the wave environment data set. This is illustrated in Table D.2.2.7 for the $5 < T_z < 6$ sec condition. This step is not necessary if operational data is available in terms of Wave Heights.

Table D.2.2.7: Transformation of Sea State and Wave Height.

H _s [m]	Composite Marsden Z.		Sea State	Range of H _s (m)	Combined Marsden Z.
0-1	0.0524	⇒	1	0-0.1	0.0000
1-2	0.0845		2	0.1-0.5	0.0524
2-3	0.0364	⇒	3	0.5-1.25	0.0699
3-4	0.0113		4	1.25-2.5	0.0500
4-5	0.0034	⇒	5	2.5-4	0.0143
5-6	0.0010		6	4-6	0.0026
6-7	0.0003	⇒	7	6-9	0.0002
7-8	0.0002			sum	0.1895
8-9	0.0000				
9-10	0.0000				
10-11	0.0000				
11-12	0.0000				
12-13	0.0000				
13-14	0.0000				
>14	0.0000				
Sum	0.1895				

D.2.2.3.4 Calculation of Speed/Heading Probabilities

Having established the distribution of time at heading and speed for the range of Sea States, and having developed a wave climate (height and zero crossing period) for a composite “zone” which reflects the distribution of time in all the geographical zones, it remains to combine these to obtain the joint probabilities of speed, heading and sea conditions.

The probability of operating in a particular mode characterized by speed V , relative heading θ , and Sea State - as defined by wave height and zero crossing period ($H_s; T_z$), - is equal to the product of the probability of operating in that range of speed times the probability of being in that range of heading times the probability of being in that Sea State. This is defined as:

$$f_{\text{STOTAL}} = \underbrace{f_{\text{mc}}(H_s; T_z)}_{\text{I}} \cdot \underbrace{f_V(V | (H_s; T_z))}_{\text{II}} \cdot \underbrace{f_{\theta}(\theta | (H_s; T_z))}_{\text{III}} \quad (\text{D.2.4})$$

Term I is the probability of wave height/zero crossing period in the Combined Marsden Zone. Term II is the conditional probability of speed V given a Sea State ($H_s; T_z$), and term III is the conditional probability of heading θ for a given Sea State ($H_s; T_z$):

$$f_V(V | (H_s; T_z)) = \frac{\text{prob}(V \text{ and } (H_s; T_z))}{\text{prob}(H_s; T_z)_V} \quad (\text{D.2.5})$$

$$f_{\theta}(\theta | (H_S; T_Z)) = \frac{\text{prob}(\theta \text{ and } (H_S; T_Z))}{\text{prob}(H_S; T_Z)} \quad (\text{D.2.6})$$

where:

- $\text{prob}(V \text{ and } (H_S; T_Z)) =$ is the joint probability of speed and sea state (entries in the Operational Profile Table D.2.2.1 with Sea State transformed to Wave Height and Crossing Period).
- $\text{prob}(H_S; T_Z)_V =$ is the probability of the sea state given in the bottom row of the Operational Profile Table D.2.2.1.
- $\text{prob}(\theta \text{ and } (H_S; T_Z)) =$ is the joint probability of heading and sea state (entries in the Operational Profile Table D.2.2.2).
- $\text{prob}(H_S; T_Z)_\theta =$ is the probability of sea state given in the bottom row of the Operational Profile Table D.2.2.2.

The process is illustrated as follows. From Table D.2.2.1

Sea State								SUM
Speed (kn)	1	2	3	4	5	6	7	
10-12	0.0078	0.0000	0.0000	0.0072	0.0130	0.0326	0.0148	0.0755
12-14	0.0305	0.0315	0.2140	0.2112	0.1957	0.1509	0.0000	0.8339
14-16	0.0000	0.0196	0.0261	0.0285	0.0067	0.0097	0.0000	0.0906
SUM	0.0383	0.0511	0.2401	0.2470	0.2155	0.1932	0.0148	1.0000

$$f_V(V | (H_S; T_Z)) = \frac{\text{prob}(V \text{ and } (H_S; T_Z))}{\text{prob}(H_S; T_Z)_V} = f_V(V = (12 \text{ to } 14 \text{ kn}) | (H_S; T_Z))$$

$$= \frac{0.2140}{0.2401}$$

and from Table D.2.2.2

Sea State								SUM
Rel. Heading	1	2	3	4	5	6	7	
Head Seas	0.0066	0.0064	0.0316	0.0338	0.0313	0.0319	0.0148	0.1564
Strbd. Bow	0.0212	0.0187	0.1154	0.1191	0.1124	0.1022	0.0000	0.4890
Strbd. Beam	0.0038	0.0114	0.0365	0.0371	0.0268	0.0224	0.0000	0.1380
Strbd. Quart.	0.0027	0.0071	0.0250	0.0252	0.0191	0.0157	0.0000	0.0948
Following	0.0034	0.0099	0.0322	0.0327	0.0238	0.0199	0.0000	0.1218
SUM	0.0378	0.0534	0.2407	0.2479	0.2134	0.1921	0.0148	1.0000

$$f_{?}(?|(H_S;T_Z)) = \frac{\text{prob}(? \text{ and } (H_S;T_Z))}{\text{prob}(H_S;T_Z)} = f_{?}(? = (\text{Head Seas}))(H_S;T_Z)$$

$$= \frac{0.0316}{0.2407}$$

Total probability of a given operating mode then is:

$$f_{\text{STOTAL}} = \underbrace{f_{\text{mcomb}}(H_S;T_Z)}_{\text{I}} \cdot \underbrace{f_V(V|(H_S;T_Z))}_{\text{II}} \cdot \underbrace{f_{?}(?|(H_S;T_Z))}_{\text{III}}$$

$$f_{\text{STOTAL}} = \underbrace{0.0699}_{\text{I}} \cdot \underbrace{\frac{0.2140}{0.2401}}_{\text{II}} \cdot \underbrace{\frac{0.0316}{0.2407}}_{\text{III}} = 0.00818$$

where Term I comes from Table D.2.2.7.

Summarizing this calculation, the probability of occurrence of Sea State 3 in Head seas, with the vessel speed between 12 to 14 knots is 0.00818. These calculations are repeated for all combinations of speed and headings. Therefore, in this example, there are 3 ranges of speed, 5 ranges of heading and 7 Sea States. Thus, the theoretical number of operating mode combinations for a **single** zero crossing period is $3 \times 5 \times 7 = 105$ entries.

The calculated data for a single zero crossing period is as shown in Table D.2.8

The values shown in the table are probabilities standardized by multiplying by 1000. Thus, the probability of occurrence of Sea State 3, in Head seas with a vessel speed between 12 to 14 knots is $8.2/1000 = 0.0082$. In the table sign “~” indicates absolute zero, and numbers are given with one significant digit. This means that probabilities less than 0.0001 are not shown.

For the operational profile used in this example there are 3 ranges of speed, 5 ranges of heading, and 7 sea states covering 11 zero crossing periods. The theoretical number of operating mode combinations is $3 \times 5 \times 7 \times 11 = 1155$ entries. The fact that some probabilities are zeros or very small, reduces the number of combinations for which vessel response need be evaluated. In this example the total number of operational modes was 231.

Table D.2.2.8: Probabilities of all combinations of Speed, Heading and Sea State for zero crossing period T_z between 5 and 6 seconds in Combined Marsden Zone.

SPEED	Sea State 1					Sea State 2				
	Head Seas	Strbd. Bow	Strbd. Beam	Strbd. Quart.	Follng	Head Seas	Strbd. Bow	Strbd. Beam	Strbd. Quart.	Follng
10-12	--	--	--	--	--	--	--	--	--	--
12-14	--	--	--	--	--	3.85	11.33	6.89	4.29	5.97
14-16	--	--	--	--	--	2.39	7.03	4.28	2.66	3.71

SPEED	Sea State 3					Sea State 4				
	Head Seas	Strbd. Bow	Strbd. Beam	Strbd. Quart.	Follng	Head Seas	Strbd. Bow	Strbd. Beam	Strbd. Quart.	Follng
10-12	--	--	--	--	--	0.20	0.70	0.22	0.15	0.19
12-14	8.18	29.88	9.45	6.46	8.33	5.84	20.55	6.40	4.36	5.64
14-16	1.00	3.64	1.15	0.79	1.02	0.79	2.78	0.86	0.59	0.76

SPEED	Sea State 5					Sea State 6				
	Head Seas	Strbd. Bow	Strbd. Beam	Strbd. Quart.	Follng	Head Seas	Strbd. Bow	Strbd. Beam	Strbd. Quart.	Follng
10-12	0.13	0.46	0.11	0.08	0.10	0.07	0.23	0.05	0.04	0.05
12-14	1.91	6.86	1.64	1.17	1.45	0.34	1.08	0.24	0.17	0.21
14-16	0.07	0.24	0.06	0.04	0.05	0.02	0.07	0.02	0.01	0.01

SPEED	Sea State 7				
	Head Seas	Strbd. Bow	Strbd. Beam	Strbd. Quart.	Follwng
10-12	0.21	--	--	--	--
12-14	--	--	--	--	--
14-16	--	--	--	--	--

D.2.3 Calculation of Loads

D.2.3.1 Overview

A linear strip theory program is used to calculate the structural response of the hull girder to the range of sea conditions in the problem. This example focuses on vertical bending moment of the hull in the midships area.

D.2.3.1.1 Selection of a Wave Spectrum

While the information that has been established to date identifies the degree of exposure to each sea condition that is represented in the geographical zones, the seas have been identified by wave height and zero crossing period. It remains to select a distribution for the wave energy for these Sea States. It was assumed that representation of the sea spectra as a fully developed sea was sufficient, and the irregular description of sea was modeled using the Bretschneider Spectrum [Ref D.4].

The input parameters for the spectrum are the significant wave height H_s and the peak period T_p . For each Sea State, the H_s used corresponded to the mid value of the Sea State wave height range. The relationship between the peak wave period T_p , required by the loads program and the zero crossing wave period (T_z) used so far, based on the wave data is:

$$T_p = 1.408T_z \quad (D.2.7)$$

Note that the program makes the necessary calculations to transcribe from stationary wave data to encountered wave data due to ship speed and heading relative to the waves.

D.2.3.1.2 Establishing Vertical Bending Moment

Responses of the example ship in the conditions identified in the previous section were calculated using the frequency domain linear strip theory code SHIPMO 7 [Ref. D.5]. Output for each operating mode (ship speed and heading) and sea condition included: the Response Amplitude Operator (RAO); Root Mean Square (RMS) loads; and zero crossing period, T_z of the response in irregular seas. Inherent in the use of strip theory for these calculations is the assumption of a linear relationship between input wave height and the vessel response.

Table D.2.3.1 shows a sample of the output of Shipmo7 for one Sea State.

Table D.2.3.1: Ship Response Calculation: Output from Shipmo7.

Sea State = 3				<i>Sig. Wave H. = 0.88 m</i>	
				SHIPMO 7 Output	
				Vert. Bend. Moment	
Speed (kn.)	Heading	T _p (sec.)	prob*100 0.	T _z (sec.)	RMS (kNm)
13	Head	4.93	0.6	2.20	3803.538
13	Strbd. Bow	4.93	2.1	3.00	3620
13	Strbd. Beam	4.93	0.7	4.90	7481.692
13	Follow.	4.93	0.6	22.50	2640.077
13	Head	6.34	4.2	3.80	7250.077
13	Strbd. Bow	6.34	14	4.70	7789.231
13	Strbd. Beam	6.34	4.4	5.70	9486.154
13	Strbd. Quart.	6.34	3.4	12.80	13314.85
13	Follow.	6.34	4.4	18.40	5894.615
15	Strbd. Bow	6.34	1.7	4.40	7920.692
15	Strbd. Beam	6.34	0.5	5.70	9351
13	Head	7.75	8.2	5.30	13084.62
13	Strbd. Bow	7.75	29.9	5.80	13080.31
13	Strbd. Beam	7.75	9.5	6.30	9622.077
13	Strbd. Quart.	7.75	6.5	13.20	22347.15
13	Follow.	7.75	8.3	17.50	13793.92
⋮ ↓ ⋮	⋮ ↓ ⋮	⋮ ↓ ⋮	⋮ ↓ ⋮	⋮ ↓ ⋮	⋮ ↓ ⋮

For this example 231 SHIPMO7 runs were completed, accounting for the operating combinations with any non-zero probability of occurrence. Values of RMS Vertical Bending Moment (M) were divided into 20 equal bins, covering the range of BM's from the minimum to the maximum calculated value. The probability of each bending moment range is equal to the sum of all probabilities corresponding to the RMS values falling within the boundaries of each bin. The resulting histogram is shown below.

Range of RMS (kNm)	Bin #	prob.*1000	
0	9959	1	182.0
9959	19918	2	198.9
19918	29877	3	100.4
29877	39836	4	118.6
39836	49795	5	107.8
49795	59754	6	45.1
59754	69713	7	46.0
69713	79672	8	80.5
79672	89631	9	15.4
89631	99590	10	23.2
99590	109549	11	13.1
109549	119508	12	7.5
119508	129467	13	19.1
129467	139426	14	6.1
139426	149385	15	4.9
149385	159344	16	1.8
159344	169303	17	1.5
169303	179262	18	3.6
179262	189221	19	4.2
189221	199180	20	7.0
199180<		21	5.7
Sum:			992.4

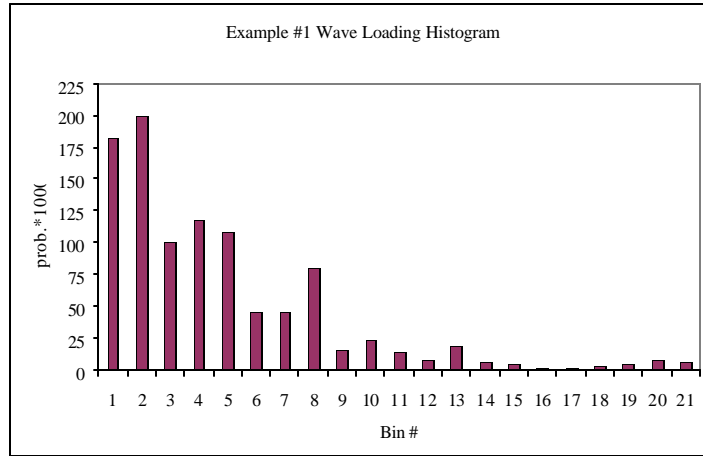


Figure D.2.3.1: Vertical Bending Moment (RMS) Distribution

Before these bending moment loads were transferred into the stress ranges in a particular part or detail of the ship structure, there was a further step to be completed to obtain an **annual**¹ distribution of vertical bending moment. It is necessary to repeat the same procedure as described up to this point using any other ship operating condition, for example in this case, the “Ballast” operational profile data. This would result in different stationary operational conditions, associated probabilities and responses.

This has not been undertaken in the example as the procedure has been demonstrated. The result of a complete analysis would be two histograms, one for “Loaded” and one for “Ballast” condition. The annual wave loading distribution is obtained by multiplying each bin of the histograms by an appropriate weighting factor that reflects the relative time spent in each condition. Note that the Bin # boundaries should be identical in both histograms. For example, RMS range in bin #1 for the ship loaded condition operational profile is from 0 to 9959 kNm. The range of RMS in Bin #1 for the ballast condition profile should be the same, i.e., from 0 to 9959 kNm.

The annual probability distribution of wave loading is calculated from:

$$\text{prob}(\text{Bin } \#i)_{\text{total}} = \text{prob}(\text{Bin } \#i)_{\text{loaded}} \cdot \frac{3624.4}{(3624.4 + 1563.6)} + \text{prob}(\text{Bin } \#i)_{\text{ballast}} \cdot \frac{1563.6}{(3624.4 + 1563.6)} \quad (\text{D.2.8})$$

where 3624.4 is the number of hours in loaded condition and 1563.6 is the number of hours in ballast condition. These numbers were taken from Table D.2.2.4 of the operational profile data.

¹ Recall that Operational Profiles for example ship were developed on an annual basis.

D.2.4 Determine Reference Hot Spot Stress Range

The vertical bending moment histogram defines the bending moments experienced by the whole vessel under each operational condition. These bending moments are used to define the stress range at the location of interest for the fatigue analysis. In order to relate the global vertical bending moments to the localized stresses at a particular detail, theoretical or numerical methods can be employed. In this example, a linear elastic finite element model was constructed to relate the vertical moment to the local stress for the location of interest.

D.2.4.1 Finite Element Model

Both a global and refined substructure model of the hatchway was constructed. Sub-structuring allows an efficient method of determining localized stresses while remaining computationally efficient. Boundary conditions from the global model are passed to the refined substructure model. The global model was subjected to a unit bending moment of 2000 kN*m.

The location of interest in the welded intersection between an angled support bracket and the base plate. The bracket supports the coaming around a large central hatchway. Fatigue cracking is expected in the base plate with the crack running perpendicular to the longitudinal axis of the vessel. Figure D.2.4.1 illustrates both the global and local models used in the finite element model. Superimposed on the substructure model are the fatigue crack location and direction of principle stress

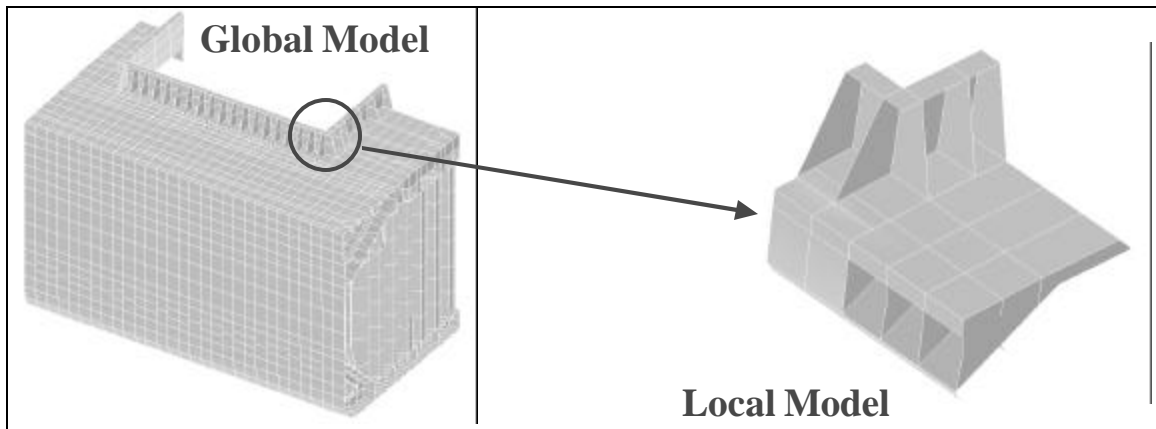


Figure D.2.4.1: Global and Substructure FEM Model

D.2.4.2 Determine Reference Hot Spot Stress Ranges

The hot spot stress approach was used in the analysis. Figure D.2.4.2 plots the principle longitudinal stress in the base plate starting from the corner of the coaming and emanating in the longitudinal direction. Two structural conditions are used, with and without a doubler plate at the location of interest. Figure D.2.4.3 shows the extrapolation of the stress to the hot spot. The hot spot is located 855 mm from the coaming. Points C and D define the distances for the linear extrapolation to the location of the hot spot stress. These points are located 12 and 36 mm respectively away from the coaming bracket to deck intersection when considering the case with a doubler plate.

The hot spot stress range is calculated by assuming fully reversing moments. For the doubler plate case:

$$\begin{aligned}\Delta\sigma_{\text{hot spot}} &= 2 K_o M_v \dots \text{for each } (H_s, T_z, V, \theta, \text{L.C.}) \\ &= 2 (0.262) M_v / 2000 \\ &= 2.62 \times 10^{-4} M_v\end{aligned}$$

where M_v is in kNm and $\Delta\sigma_{\text{hot spot}}$ is in MPa.

and K_o is the transfer function between hot spot stress and design moment.

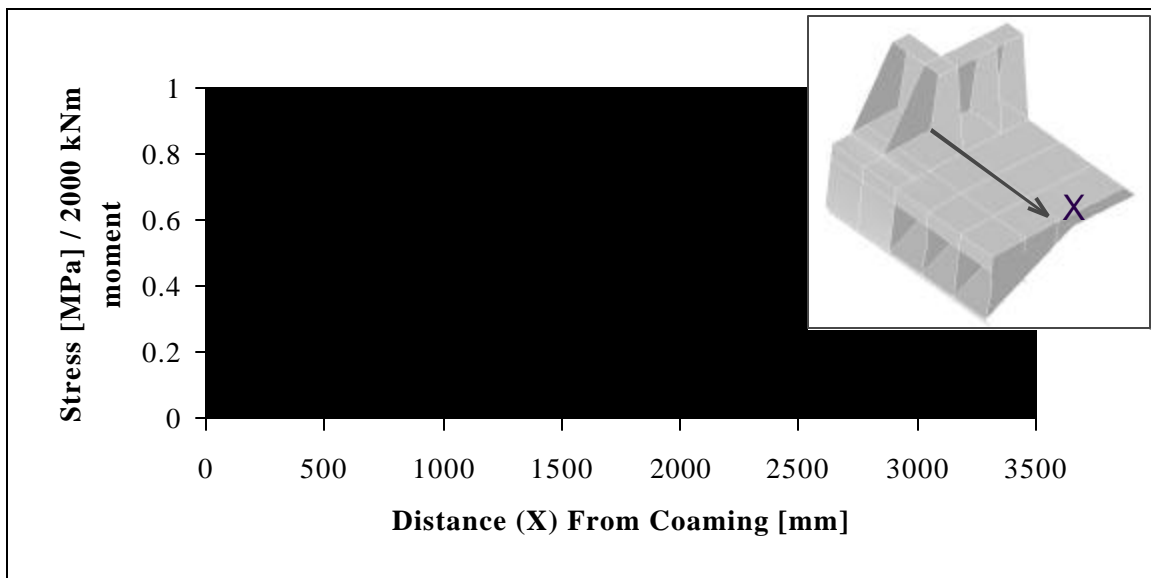


Figure D.2.4.2: FEM Stress Distribution along the longitudinal

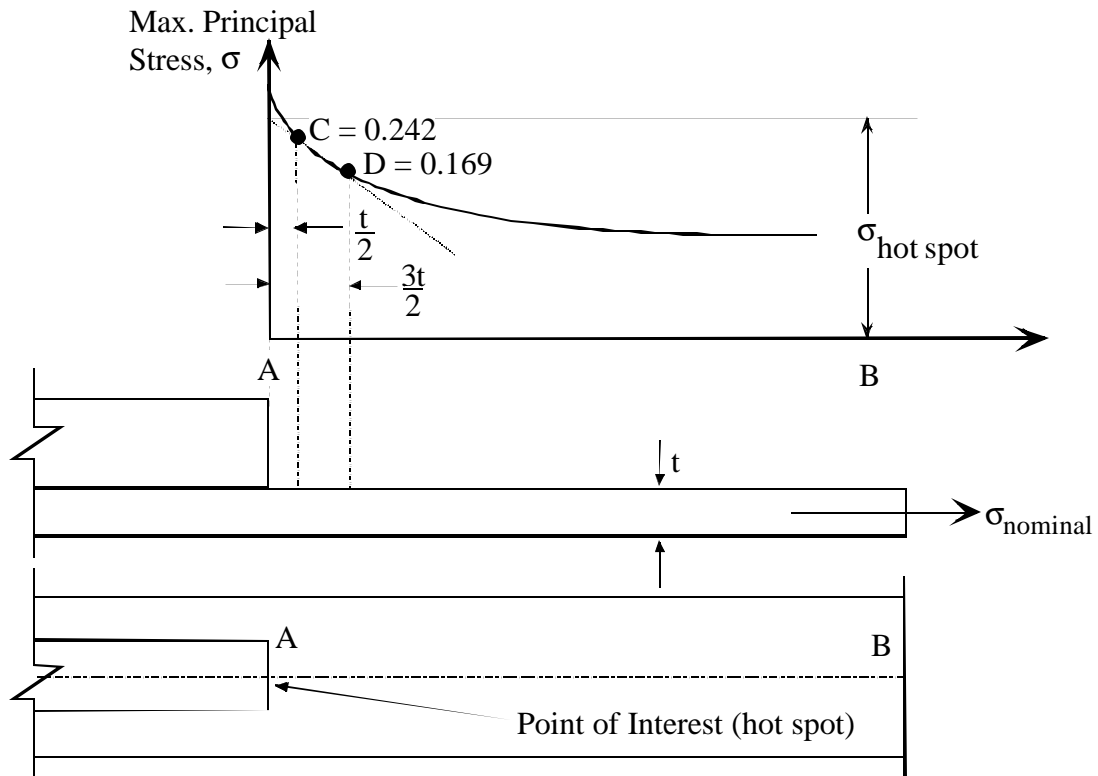


Figure D.2.4.3: Extrapolation of Hot Spot Stress (With Doubler)

Using the transfer function in **Table D.2.4.1**, it is possible to convert the vertical bending moments calculated previously into the hot spot stress range.

Table D.2.4.1: Hot Spot Transfer Functions for the Location of Interest

	Doubler	No Doubler
K_o	0.262	0.292
$\Delta\sigma_{\text{Hot Spot}}$	$2.62 \times 10^{-4} M_v$	$2.91 \times 10^{-4} M_v$

D.2.5 Define Short-Term Stress Range Distribution

A short-term stress range distribution is constructed for each of the operational conditions defined in the matrix of stationary conditions (Section D.2.2.3.4). The short range distribution is a Rayleigh distribution with the following form (equation C.5.5).

$$F_{\Delta\sigma_i}(\Delta\sigma) = 1 - \exp\left(\frac{-\Delta\sigma^2}{8m_{oi}}\right) \quad (D.2.8)$$

where m_0 is the zeroth moment of the wave spectrum and is calculated in terms of the RMS design moment (M_{vi}), for each operational condition as:

$$m_{oi} = (2K_o M_{vi})^2 \quad (D.2.9)$$

D.2.6 Long-Term Stress Range History

The short-term behaviors are summed to develop a long-term stress range distribution as follows:

$$Q(\Delta\sigma) = \sum_{\text{all conditions}} F_{\Delta\sigma_i}(\Delta\sigma) r_i p_i \quad (D.2.10)$$

where p_i is the fraction of time at each of the i operational conditions and r_i is the ratio of crossing rate and average crossing rate (zero crossing period, \bar{T}). The average zero crossing rate and crossing rate ratio are calculated as follows:

$$\bar{T} = \sum_{\text{all conditions}} p_i T_{zi} \quad \text{and} \quad r_i = \frac{T_{zi}}{\bar{T}} \quad (D.2.11)$$

If the short-term distributions are summed in a discrete fashion, the long-term distribution is fitted using a least squares regression to select the appropriate Weibull distribution parameters q and h , where:

$$Q(\Delta\sigma) = 1 - \exp\left(-\left[\frac{\Delta\sigma}{q}\right]^h\right) \quad (D.2.12)$$

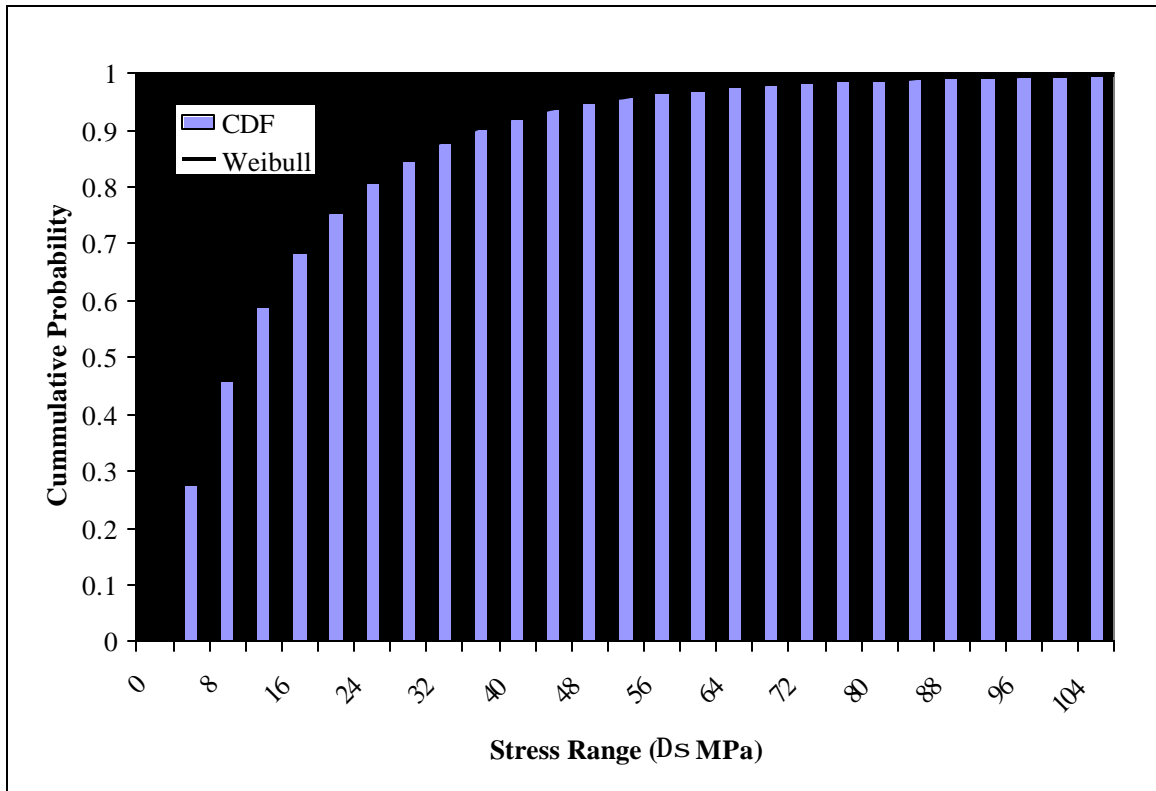


Figure D.2.6.1: Long-Term Stress History

The least square Weibull Statistics calculated for the two configuration cases are tabulated in Table D.2.6.1.

Table D.2.6.1: Weibull Parameters for the Long-Term Stress History

	With Doubler	No Doubler
q(Scale)	26.6	35.7
h(Shape)	1.102	1.060

D.2.7 Perform Fatigue Damage Summation

As in the first example, the fatigue damage is assessed using the Miner-Palmgren linear damage model, where the damage summation is in the following form:

$$D = \sum d_i = \sum \frac{n_i}{N_i} \quad (\text{D.2.13})$$

where D is the cumulative damage, d_i is the damage from i^{th} stress cycle, N_i is the average number of loading cycles to failure at the i^{th} stress range and n_i is the number of cycles at each stress range in the period of interest. The fatigue damage accumulated in one year at each stress range is presented in Table D.2.7.1. The summation of the fatigue damage for the doubler and no doubler cases, indicate annual cumulative damages of 0.25 and 0.71, respectively.

By assuming that the vessel is at sea only 85% of the time, the fatigue life may be calculated as $1 / (0.85 * D)$ and are presented in Table D.2.7.2.

Table D.2.7.1: Sample Discrete Fatigue Damage Summation (Level 3)
(Doubler Case)

Stress Range Mid Value, $\Delta\sigma$ [MPa]*	Stress Range Probability, p_i †	Number of Cycles per Year, n_i †	Cycles to Failure, N_i ‡	Damage Ratio, $d_i = n_i / N_i$
10	5.18E-01	2591235	4.34E+10	5.98E-05
30	2.73E-01	1366097	1.78E+08	7.66E-03
50	1.22E-01	611533.4	1.39E+07	4.41E-02
70	5.17E-02	258325.1	4.43E+06	5.83E-02
↓	↓	↓	↓	↓
450	1.82E-10	0.000912	1.67E+04	5.47E-08
470	6.05E-11	0.000303	1.46E+04	2.07E-08
			Sum	2.49E-01

(No Doubler Case)

Stress Range Mid Value, $\Delta\sigma$ [MPa]*	Stress Range Probability, p_i †	Number of Cycles per Year, n_i †	Cycles to Failure, N_i ‡	Damage Ratio, $d_i = n_i / N_i$
10	4.18E-01	2089411	4.34E+10	4.82E-05
30	2.58E-01	1292383	1.78E+08	7.24E-03
50	1.47E-01	735191.6	1.39E+07	5.30E-02
70	8.14E-02	407146.2	4.43E+06	9.18E-02
↓	↓	↓	↓	↓
450	2.99E-07	1.492549	1.67E+04	8.94E-05
470	1.50E-07	0.748971	1.46E+04	5.11E-05
			Sum	7.15E-01

* All probable stress ranges are captured in 20 MPa stress range intervals (i.e. 0 to 20 mid value = 10, 20 to 40 mid value = 30, etc.)

† Probabilities for each stress range interval is estimated based on the difference between the probability of exceedance of its upper and lower bounds using equation D.1.10 (i.e. $p_i = Q(\Delta\sigma_{lower}) - Q(\Delta\sigma_{upper})$)

‡ Assuming a 20 year design life with 10^8 wave encounters, the number of cycles per year for each stress range is calculated as the product its probability and the number of wave events per year (i.e. $10^8 p_i / 20$).

§ Cycles to failure are calculated using equation D.1.1 or using Figure D.1.2.2.

Table D.2.7.2: Summary of Fatigue Damage

	Annual Damage, D	Fatigue Life (Years)
Doubler	0.25	4.7
No Doubler	0.71	1.7

Figure D.2.7.1 plots the annual fatigue damage for an individual stress range. This representation illustrates the stress range levels that contribute most significantly to the fatigue damage accumulation. While in general, larger stress ranges produce more fatigue damage, they are not observed as often as lower stress range events.

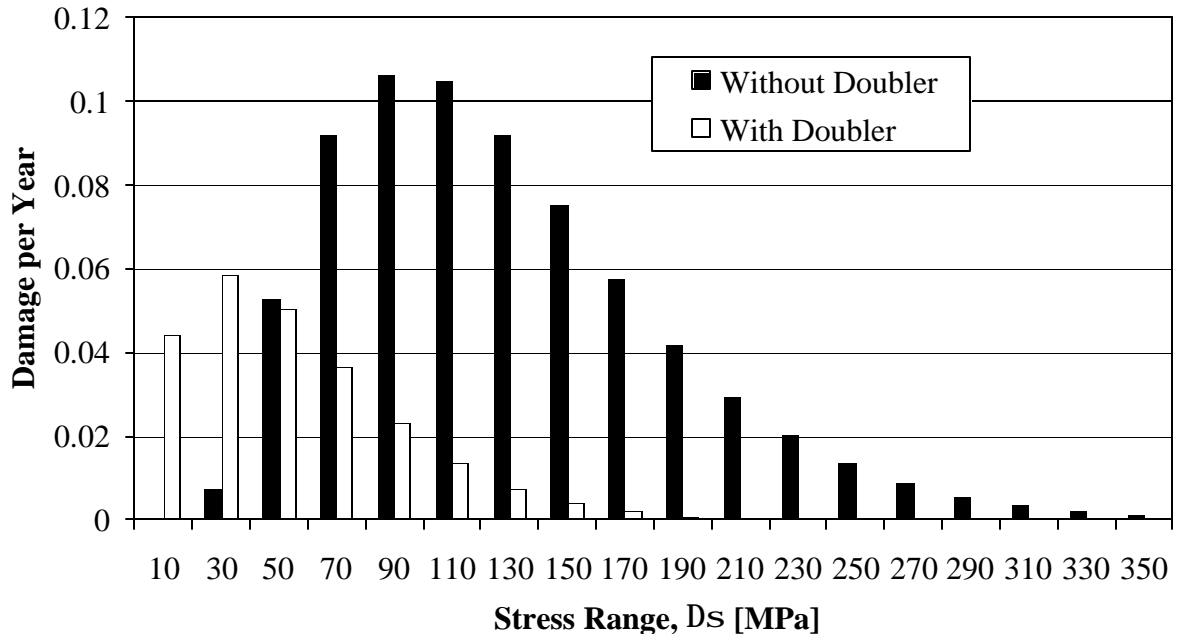


Figure D.2.7.2: Yearly Fatigue Damage

As can be seen by comparing Examples 1 and 2, the spectral analysis (Level 3) should provide a much more representative estimate of fatigue life, since it is based on the operational experience of the case vessel. Another point of interest, as a design case, is that the introduction of a doubler plate improves fatigue life about 300%. Despite this local fatigue life improvement, the finite element analysis indicates that the fillet weld at the edge of the doubler plate, attaching it to the deck, is a potential fatigue crack initiation site which should be investigated.

A final observation is that the designer must be conscious of the degree of error associated with the fatigue life prediction. Thus, despite the convergence on a single value for life in the process, the designer must consider the number of possible sources of error in the process; in developing the operational profile, the spectral prediction of bending moment, in the application of stress concentration factors, and in the experimental data used to develop the damage estimates. Sensitivity analyses are recommended, and the designer would be prudent to expect margins of error of at least 20%.

D.3 EXAMPLE 3: - SIDE PRESSURE CALCULATIONS

D.3.1 Objective

To conduct a comprehensive fatigue analysis, based on the spectral method (Level 3) for a stiffened panel including a side shell structural detail for a product tanker operating on a route between the U.S. West Coast and Alaska. The fatigue life is to be estimated for a bracket loaded by lateral side pressure at the intersection with transverse bulkheads.

D.3.2 Approach

The basic stages in the analysis follow those defined in Example 2. However, the loading on the structure was considered in two parts: first, quasi-static pressure due to vertical displacement of the point of interest, and second, the dynamic pressure at the point of interest due to a wave impinging on the vessel side shell. These components are shown schematically in Figure D.3.2.1.

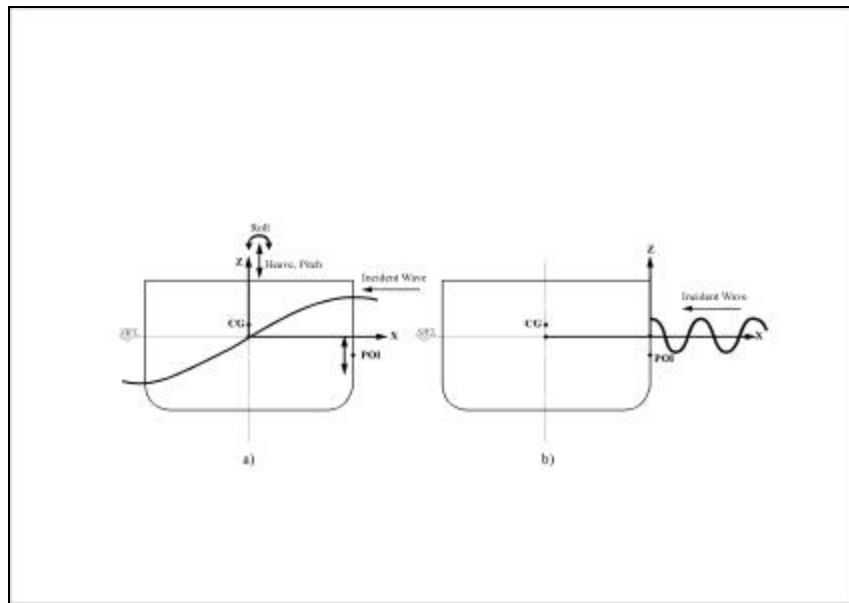


Figure D.3.2.1: Schematic of Side Shell Load Components

Several fatigue failures near the waterline and on the weather side have been reported in the literature [e.g., Ref. D.6]. Most of these failures occurred next to a bulkhead or web frame in way of an attachment bracket, and were associated with local pressure acting on the ship's side. This effect is essentially non-linear. However, in this example, an approach based on linear theory will be employed to address the problem.

In this example only the outside pressure will be considered, i.e., only atmospheric pressure is acting on the inner plating of the side shell. In terms of the example, this case corresponds to the cargo leg of the voyage, where a ballast tank, containing the structure of interest, is empty.

D.3.3 Operational Profile Data

D.3.3.1 Definition of the Operational Profile

Operational profile data used in the example was for a tanker operating on the West Coast of North America between California and Alaska. The principal particular of the example ship and operational profile data used in the example are in Tables D.3.3.1 and D.3.3.2 respectively. The operational profile data is based on historical records.

Table D.3.3.1: Principal Data: Example Ship

Ship Length (Lpp)	187.15 m
Ship Beam	28.96 m
Draft (Loaded Condition)	12.4 m
Displacement (Loaded Condition)	58,800 tonnes
KG	12.2 m

Table D.3.3.2a: Joint Probability of Speed and NATO Sea State for Example Ship

SPEED (kn)	NATO Sea State						
	1	2	3	4	5	6	7
0 - 6	0.0028	0.0000	0.0009	0.0000	0.0000	0.0010	0.0007
6 - 10	0.0056	0.0031	0.0033	0.0082	0.0086	0.0261	0.0010
10 - 14	0.0129	0.0219	0.0503	0.0378	0.0449	0.0896	0.0019
14 - 18	0.0900	0.1253	0.1322	0.1007	0.1338	0.0968	0.0005
0.9999	0.1113	0.1503	0.1866	0.1467	0.1873	0.2135	0.0041

Table D.3.3.2b: Joint Probability of Heading and NATO Sea State for Example Ship

HEADING	NATO Sea State						
	1	2	3	4	5	6	7
Head Seas	0.0084	0.0114	0.0167	0.0128	0.0160	0.0221	0.0005
Strbd. Bow	0.0234	0.0326	0.0380	0.0295	0.0383	0.0375	0.0005
Strbd. Beam	0.0283	0.0375	0.0439	0.0355	0.0454	0.0497	0.0009
Strbd. Quartering	0.0421	0.0586	0.0729	0.0565	0.0724	0.0800	0.0013
Following	0.0098	0.0136	0.0164	0.0129	0.0165	0.0176	0.0003
1.0000	0.1120	0.1537	0.1880	0.1472	0.1886	0.2070	0.0035

Table D.3.3.2c: Relative Time Spent in Marsden Zones for Example Ship

Marsden Zone	% of time, (μ_i)
6	26.00
7	18.00
13	11.00
14	20.00
22	25.00

D.3.3.2 Composite Scatter Diagram and Probabilities of Stationary Conditions

The procedure for calculations of wave composite scatter diagram for this example was identical to one used in spectral approach of the Example #2. As before, the source of wave statistic data used was Global Wave Statistics [Ref. D.3]. In the example annual wave data for unidirectional seas was utilised.

The composite probability distribution of wave heights and periods for all Marsden zones combined was calculated according to the following equation:

$$f_{mc}(H_S; T_Z) = \sum_{i=1}^{N'} \mu_i f_{mi}(H_S; T_Z) \quad (D.3.1)$$

where $f_{mi}(H_S; T_Z)$ is the probability distribution of wave heights and periods in Marsden Zones 6, 7, 13, 14, and 22, μ_i are given in Table D.3.3.2c, and $N' = 5$ (the number of Marsden Zones). The resulting Composite Scatter Diagram is shown in Table D.3.3.3.

Table D.3.3.3: Composite Marsden Zone Scatter Diagram

H_s [m]	T_z											Sum
	<4	4-5	5-6	6-7	7-8	8-9	9-10	10-11	11-12	12-13	>13	
0-1	0.0015	0.0153	0.0373	0.0362	0.0175	0.0046	0.0012	0.0000	0.0000	0.0000	0.0000	0.1135
1-2	0.0003	0.0070	0.0460	0.0948	0.0870	0.0441	0.0145	0.0035	0.0007	0.0000	0.0000	0.2979
2-3	0.0000	0.0017	0.0180	0.0589	0.0830	0.0626	0.0295	0.0096	0.0023	0.0005	0.0000	0.2662
3-4	0.0000	0.0003	0.0053	0.0237	0.0453	0.0452	0.0275	0.0114	0.0038	0.0008	0.0000	0.1632
4-5	0.0000	0.0000	0.0012	0.0082	0.0194	0.0243	0.0181	0.0090	0.0035	0.0008	0.0003	0.0848
5-6	0.0000	0.0000	0.0003	0.0028	0.0080	0.0110	0.0096	0.0059	0.0023	0.0008	0.0003	0.0409
6-7	0.0000	0.0000	0.0000	0.0012	0.0030	0.0050	0.0049	0.0032	0.0015	0.0003	0.0002	0.0192
7-8	0.0000	0.0000	0.0000	0.0003	0.0010	0.0021	0.0020	0.0016	0.0010	0.0003	0.0000	0.0083
8-9	0.0000	0.0000	0.0000	0.0000	0.0005	0.0010	0.0010	0.0011	0.0003	0.0002	0.0000	0.0040
9-10	0.0000	0.0000	0.0000	0.0000	0.0003	0.0004	0.0008	0.0003	0.0003	0.0000	0.0000	0.0020
10-11	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
11-12	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
12-13	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
13-14	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
>14	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
Sum	0.0017	0.0243	0.1080	0.2261	0.2650	0.2003	0.1091	0.0456	0.0157	0.0036	0.0008	1.0000

The operational profile data and the Composite Marsden Zone scatter diagram are combined to obtain the probabilities of stationary (operating) conditions.

The next consideration is that the wave environmental data and the vessel operational profile data should be defined on the same basis. In this case, operational profile data was ordered using the NATO Sea State scale, and environmental data was organized by significant wave height. Thus, to calculate the stationary condition probabilities it is necessary to convert the wave height probabilities from Table D.3.3.3 to Sea State probabilities. Furthermore, for a comprehensive analysis this would be done for all zero crossing periods in the Composite scatter diagram. For the purpose of this example, the Composite scatter diagram was reduced into two wave height distributions. This was accomplished by grouping wave height data for zero crossing period bins from 4< to 9-10 seconds together for the first distribution and wave height data for period bins from 10-11 to 13> seconds for the second.

Grouping of the wave periods was done for two reasons: reduction of the number of calculations; and an expectation of that there would be a dominant source of pressure variation at POI associated with each distribution. This will be described in detail in Section 6. Table D.3.3.4 shows these two distributions and the conversion to Sea States. The procedure used for Sea State-wave height conversion is outlined in Section D.2.2.3.2.

Table D.3.3.4: Transformation of Wave heights to Sea States for Combined Scatter Diagram

Include zero crossing period T_z bins from <4 to 9-10 seconds				Include zero crossing period T_z bins from 10-11 to >13 seconds			
Sea State	Prob.	Range of H_s (m)	Prob	Sea State	Prob.	Range of H_s (m)	Prob
1	0.0000	0-1	0.1135	1	0.0000	0-1	0.0000
2	0.1127	1-2	0.2936	2	0.0000	1-2	0.0043
3	0.2342	2-3	0.2538	3	0.0027	2-3	0.0124
4	0.3112	3-4	0.1473	4	0.0134	3-4	0.0159
5	0.1864	4-5	0.0712	5	0.0221	4-5	0.0136
6	0.0745	5-6	0.0317	6	0.0202	5-6	0.0092
7+	0.0153	6-7	0.0141	7+	0.0072	6-7	0.0052
Sum	0.9344	7-8	0.0053	Sum	0.0656	7-8	0.0030
		8-9	0.0025			8-9	0.0015
		9-10	0.0014			9-10	0.0006
		10-11	0.0000			10-11	0.0000
		11-12	0.0000			11-12	0.0000
		12-13	0.0000			12-13	0.0000
		13-14	0.0000			13-14	0.0000
		>14	0.0000			>14	0.0000
		Sum	0.9344			Sum	0.0656

The stationary conditions characterised by probabilities of speed and heading for a given sea state are given in Tables D.3.4.1a and D.3.4.1b. The probabilities given in the tables are normalised by a 1000. Thus, in Table D.3.3.5a, the probability of speed between 14-18 knots and head seas at sea state 3 with waves having zero crossing period from less than 4 up to 10 seconds is $14.697/1000=0.014697$. The algorithm for the calculations of probabilities was discussed in details in section D.2.2.3.4 and is not repeated here. The theoretical number of stationary conditions is 280 (4 speeds x 5 headings x 7 sea states x 2 periods), although in practice there are a number of zero probabilities.

D.3.4 Side Shell Loading Calculations

The pressure force on the side shell of the vessel can be modelled as two components:

- (1) the quasi-static head that a point of interest on the side shell experiences due to vertical motions of the vessel;
- (2) the dynamic pressure due to waves impinging onto the side shell. This phenomenon is referred to as a standing wave.

Each load component was introduced in section C.3.2.2 and are briefly discussed here.

**Table D.3.4.1a: Probabilities of all combinations of Speed, Heading and Sea State
for zero crossing periods Tz between <4 and 9-10 seconds in
Combined Marsden Zone**

SPEED (kn)	Sea State: 1					Sea State: 2				
	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following
0_6	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
6_10	0.000	0.000	0.000	0.000	0.000	0.172	0.491	0.564	0.882	0.205
10_14	0.000	0.000	0.000	0.000	0.000	1.221	3.485	4.003	6.266	1.455
14_18	0.000	0.000	0.000	0.000	0.000	6.985	19.940	22.905	35.848	8.322
Sum	0.0	0.0	0.0	0.0	0.0	8.4	23.9	27.5	43.0	10.0

SPEED (kn)	Sea State: 3					Sea State: 4				
	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following
0_6	0.099	0.225	0.260	0.432	0.097	0.000	0.000	0.000	0.000	0.000
6_10	0.368	0.841	0.972	1.613	0.363	1.503	3.473	4.174	6.651	1.512
10_14	5.591	12.764	14.748	24.476	5.513	6.967	16.094	19.347	30.827	7.008
14_18	14.697	33.555	38.770	64.341	14.492	18.550	42.849	51.510	82.074	18.659
Sum	20.8	47.4	54.8	90.9	20.5	27.0	62.4	75.0	119.6	27.2

SPEED (kn)	Sea State: 5					Sea State: 6				
	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following
0_6	0.000	0.000	0.000	0.000	0.000	0.036	0.061	0.081	0.131	0.029
6_10	0.729	1.745	2.073	3.305	0.754	0.974	1.649	2.190	3.524	0.776
10_14	3.783	9.055	10.755	17.146	3.910	3.343	5.662	7.519	12.100	2.665
14_18	11.280	27.001	32.070	51.130	11.659	3.611	6.116	8.122	13.071	2.878
Sum	15.8	37.8	44.9	71.6	16.3	8.0	13.5	17.9	28.8	6.3

SPEED (kn)	Sea State: 7 +					SPEED (kn)
Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following		
0_6	0.412	0.370	0.723	0.994	0.200	0_6
6_10	0.550	0.493	0.964	1.325	0.266	6_10
10_14	1.099	0.986	1.929	2.651	0.533	10_14
14_18	0.275	0.246	0.482	0.663	0.133	14_18
Sum	2.3	2.1	4.1	5.6	1.1	

Table D.3.4.1b: Probabilities of all combinations of Speed, Heading and Sea State for zero crossing periods Tz between 10-11 and 14> seconds in Combined Marsden Zone.

SPEED (kn)	Sea State: 1					Sea State: 2				
	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following
0_6	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
6_10	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
10_14	0.000	0.000	0.000	0.000	0.000	0.000	0.001	0.001	0.002	0.000
14_18	0.000	0.000	0.000	0.000	0.000	0.002	0.005	0.006	0.010	0.002
Sum	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

SPEED (kn)	Sea State: 3					Sea State: 4				
	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following
0_6	0.001	0.003	0.003	0.005	0.001	0.000	0.000	0.000	0.000	0.000
6_10	0.004	0.010	0.011	0.019	0.004	0.065	0.150	0.180	0.287	0.065
10_14	0.064	0.146	0.169	0.281	0.063	0.301	0.695	0.835	1.330	0.302
14_18	0.169	0.385	0.445	0.738	0.166	0.801	1.849	2.223	3.542	0.805
Sum	0.2	0.5	0.6	1.0	0.2	1.2	2.7	3.2	5.2	1.2

SPEED (kn)	Sea State: 5					Sea State: 6				
	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following
0_6	0.000	0.000	0.000	0.000	0.000	0.010	0.017	0.022	0.035	0.008
6_10	0.087	0.207	0.246	0.393	0.090	0.263	0.446	0.592	0.953	0.210
10_14	0.449	1.075	1.277	2.036	0.464	0.904	1.531	2.034	3.273	0.721
14_18	1.340	3.207	3.809	6.072	1.385	0.977	1.654	2.197	3.535	0.779
SUM	1.9	4.5	5.3	8.5	1.9	2.2	3.6	4.8	7.8	1.7

SPEED						SPEED
Sea State: 7 +						(kn)
(kn)	Head Seas	Stb. Bow	Stb. Beam	Stb. Quart.	Following	(kn)
0_6	0.193	0.173	0.339	0.466	0.094	0_6
6_10	0.258	0.231	0.452	0.621	0.125	6_10
10_14	0.515	0.462	0.904	1.242	0.250	10_14
14_18	0.129	0.115	0.226	0.311	0.062	14_18
SUM	1.1	1.0	1.9	2.6	0.5	

D.3.4.1 Quasi-Static Pressure

The calculation of the quasi static pressure is given by (see Equation C.3.13)

$$\Delta p_s = \rho g | (z_2 - z_1) | \quad (D.3.2)$$

In the subsequent analysis the difference in pressure calculated from (D.3.2) will be referred to as “quasi-static pressure”.

It is expected that the static head ($z_2 - z_1$) will be derived from a vertical motion response prediction for the POI (combining local roll, heave, and pitch components) from the spectral analysis of motion response. It is recommended that the single amplitude response be used due to intermittent emergence of the point of interest, noting that when the point of interest emerges from the water, the outside pressure falls to zero.

D.3.4.2 Pressure Due to Wave Impinging on the Side Shell

The pressure component due to wave action against the side of the vessel is investigated by analyzing incident regular waves on a two-dimensional free-surface-piercing body (i.e., the side shell). The approach assumes the following:

- The wave length of the incoming waves is small, such that the hull will not oscillate due to the wave excitation. Thus the side shell is stationary.
- The side shell is vertical at the intersection with the free surface.
- The effect of the waves is only felt on the weather side of the ship. On the lee side of the ship there is a “shadow” region where there is no wave action.
- The angle of incoming waves is perpendicular to the wall. This is true for beam seas and can be extended over bow and quartering seas.

With the above assumptions, the problem of wave action striking the side of the vessel reduces to analysis of an incident wave impinging on a vertical wall. The primary characteristic of this approach is the “Standing Wave”, a wave that has twice the wave height of the incoming waves on the upstream side of the vertical wall. Field observations aboard ships confirm that this type of wave behaviour occurs on the weather side of the ship [D.6].

Dynamic pressure p_d is given by (see equation C.3.14) as:

$$p_d = 2 \left[\frac{1}{2} \rho g H \cos(kx) \cos(\omega t) \right] \quad (D.3.3)$$

The linear solution for the “Standing Wave” pressure distribution can be written [D.7] as:

$$p_{\text{total}} = -\rho g z + p_d \quad (\text{D.3.4})$$

where z corresponds to the “stationary” elevation of POI from the still water line.

Note that defining $H=1$ effectively defines a transfer function (RAO) for dynamic pressure.

The magnitude of total pressure at point of interest is considered for two cases:

- (a) when the wave crest arrives at the side shell; and,
- (b) when the wave trough arrives at the side shell.

In the subsequent analysis, this pressure difference will be referred to as “dynamic pressure”. The approach for calculation of dynamic pressure is not ship dependent. The only information required is the vertical location (from the still waterline) of the point of interest. However, the assumptions of small ship oscillation should be verified against the predicted (or actual) ship motions.

D.3.4.3 Treatment of the Side Pressure Components

It is not generally expected that the phase difference between pressure at the POI due to vertical motion (static pressure) and due to wave impingement (dynamic pressure) will be known. Thus a “net pressure” analysis on the POI will not be possible. Thus, it will be necessary to conduct separate fatigue damage assessments for each loading component, each taken over the life of the vessel. This assumes that the two components do not constructively combine to create greater pressure magnitudes, i.e. that they are about 180 degrees out of phase. This is generally valid if roll is the dominant response mode. If this is not the case, the predicted life may well be unconservative as two distributions of lower intensity will generally result in less damage than a single distribution of higher loadings, due to the general form of an S-N curve.

D.3.5 Development of RAO's

In this section, methods for calculations of Response Amplitude Operators (RAO), response functions and associated Root Mean Square (RMS) values for static and dynamic pressure are described.

D.3.5.1 Quasi-static Pressure (Vertical Displacement of POI)

Response of the vessel in terms of vertical displacement of the POI was calculated using the linear strip theory program Shipmo 7. The procedure was as follows:

- Calculate RAO's. The Shipmo program produced RAO's in regular seas for the six degrees of freedom motions at the vessel's CG. Figure D.3.5.1 shows an RAO of the roll motion for the example ship.
- Specify Sea Spectrum. In this example the two-parameter (significant wave height and peak wave period) Bretschneider spectrum was used.
- Define the location of the POI. In this example, the location was taken as:
 - vertical location = 2.5 m below still waterline;
 - longitudinal location = 30 m aft from AP, and
 - 14.48 m to the starboard from centreline.
- Calculate RMS values. Shipmo 7 calculated RMS values for absolute displacements, velocities and accelerations in vertical, transverse and longitudinal directions at the location of the POI. Shipmo 7 also calculated RMS of the relative displacement of the point of interest, where "relative displacement" is the distance from the POI to the actual free surface. Associated with this value is also a probability of emergence (with exceedance parameter of 0.01) of the POI and the rate of emergence expressed in number of emergences per hour.
- Static pressure RMS at the POI due to vertical displacement is taken as:

$$\text{RMS}_{\text{Quasi-Stat. press.}} = \text{RMS}_{\text{Rel. displ.}} \cdot g \left[\frac{\text{N}}{\text{m}^2} \right] \quad (\text{D.3.5})$$

The mean square (area under the vertical displacement response spectrum curve) is the average of the square of the vertical displacement amplitude, and the root mean square (RMS) is the square root of that value. Therefore, $\text{RMS}_{\text{Stat. press.}}$ represents a single amplitude static pressure.

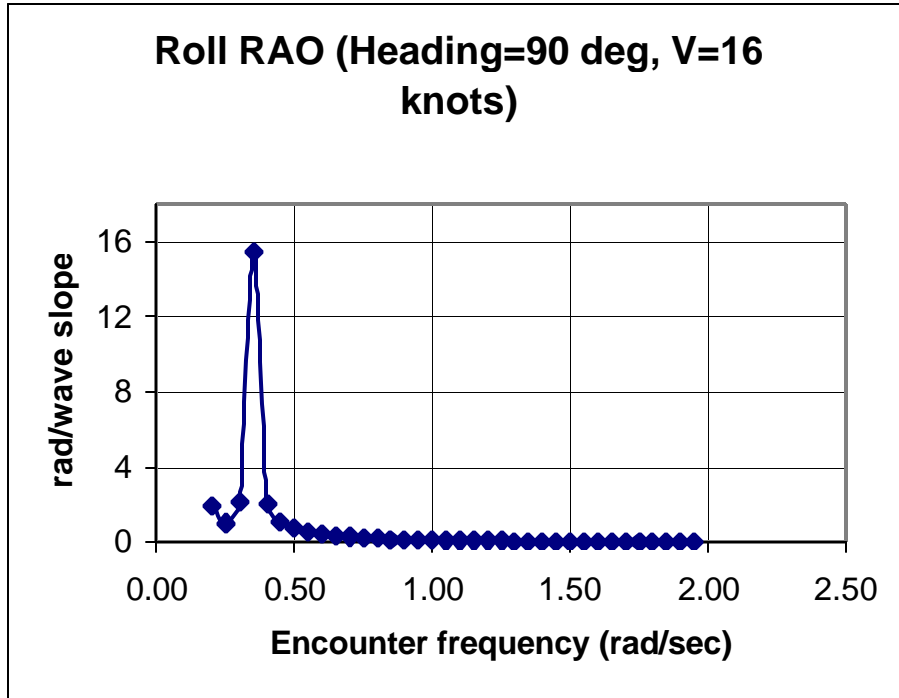


Figure D.3.5.1: Roll RAO against Encounter Frequency for Beam Seas and Speed of 16 knots

D.3.5.2 Dynamic Pressure (wave action against the side of the vessel)

For the dynamic pressure case RAO's, the response functions and associated RMS values were calculated using a spreadsheet. Figure D.3.5.2 shows calculated dynamic pressure RAO for beam seas. Note that the RAO units have dimension of N/m^3 (pressure per unit wave height).

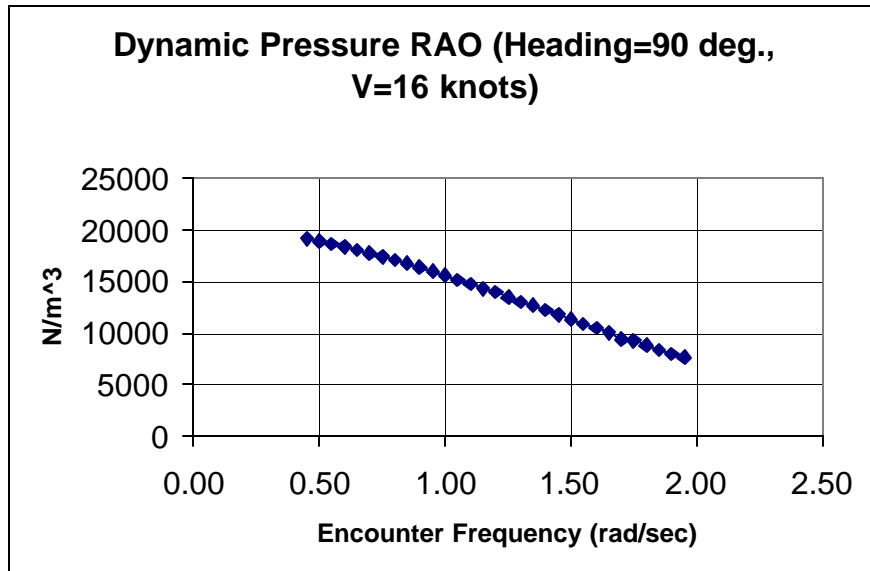


Figure D.3.5.2: Dynamic Pressure RAO against Encounter Frequency for Beam Seas

The Bretschneider sea spectrum was corrected to the encounter frequency by (Eq. C.3.2):

$$S(\omega_e) = \frac{S(\omega)}{\left|1 - \frac{2V}{g} \cos(\alpha)\right|} \quad (\text{D.3.6})$$

and the response function was calculated from:

$$R(\omega_e) = \text{RAO}(\omega_e)^2 \cdot S(\omega_e) \quad (\text{D.3.7})$$

where $R(\omega_e)$ is the response function. Phase information is retained because dynamic pressure is in phase with the wave elevation. In other words, a wave crest accompanies maximum dynamic pressure, and a wave trough accompanies minimum dynamic pressure.

Response statistics are calculated as follows:

- Zeroth moment:

$$m_0 = \int_0^{\infty} R(\omega_e) d\omega_e \cong \sum_{i=1}^N R_i(\omega_{ei}) \cdot \omega_{ei} \quad (\text{D.3.8})$$

- Second moment:

$$m_2 = \int_0^{\infty} \omega_e^2 R(\omega_e) d\omega_e \cong \sum_{i=1}^N \omega_{ei}^2 R_i(\omega_{ei}) \cdot \omega_{ei} \quad (\text{D.3.9})$$

- Root mean Square:

$$\text{RMS}_{\text{Dyn.press.}} = \sqrt{m_0} \quad (\text{D.3.10})$$

- Response Zero Crossing Period:

$$T_z = 2\pi \sqrt{\frac{m_0}{m_2}} \quad (\text{D.3.11})$$

D.3.6 Calculations of Responses

In Section D.3.3, the composite scatter diagram (Table D.3.3.3) was divided into two marginal distributions. However, the response calculations require more detailed wave period information. For a comprehensive analysis this would not be an issue as the response calculations would be carried out for every period in the scatter diagram. In this example, however, the response calculations were performed for the two aforementioned marginal distributions and the periods for each were calculated by selecting a period which would give the same order of magnitude of number of cycles as if the calculations were performed for every wave zero crossing period. Table D.3.6.1 summarizes these results.

Table D.3.6.1: Calculation of Lifetime Average Zero Crossing Period

Tz (sec.)	Number of Cycles	Tz (sec.)	Number of Cycles
<4	308,257	10.5	2,737,660
4.5	3,401,445	11.5	860,267
5.5	12,384,466	12.5	181,205
6.5	21,940,064	>13	35,519
7.5	22,281,287	Sum:	3,814,652
8.5	14,862,620	Tz (average) = 10.8	3,831,040
9.5	7,240,486	$N = prob \cdot \frac{20(\text{years})^{365 \cdot 24 \cdot 3600}}{T_z}$	
Sum:	82,418,625		
Tz (average) = 7.2	81,853,440		

The selected wave zero crossing periods of 7.2 and 10.8 seconds then had to be converted for use with the Bretschneider spectrum, which requires significant wave height and wave peak period. Conversion of zero crossing period to peak period is expressed as [D.7]:

$$T_p = 1.408 T_z \quad (\text{D.3.12})$$

Using this expression, T_p is 10.14 and 15.2 seconds. Converting to circular frequency this correspond to 0.62 rad/sec and 0.413 rad/sec. From Figure D.3.5.1, it can be seen that resonance frequency for roll is around 0.35 rad/sec. Thus, the peak period of the calculated wave average zero crossing is very close to the roll resonance frequency.

Thus it can be said that marginal wave height distribution which includes zero crossing periods from 10-11 to >13 seconds (see Table D.3.3.4) was dominated by the roll response, and therefore the static pressure component is significant. Conversely, the circular frequency of 0.62 rad/sec is sufficiently far from the resonant roll frequency (Figure D.3.5.1) such that the marginal wave height distribution from <4 up to 9-10 seconds will be dominated by the dynamic pressure component. This is consistent with the assumption made in formulating dynamic pressure; that is the ship is essentially vertically stationary.

Calculation of responses and response statistics were done for the array of 175 stationary conditions (specified by probability of occurrence of Sea State, heading and speed) identified in Tables D.3.4.1a and D.3.4.1b. A sample of the calculations is given in Table D.3.6.2. In Table D.3.6.2, Columns 1 to 3 contain the stationary condition information, and column 4 is the normalised probability of the stationary conditions. Columns 5 to 8 are Shipmo 7 output pertinent to the POI, and columns 9 and 10 are dynamic pressure calculations.

It can be seen from the Table that the number of emergences and associated probabilities increases with increasing Sea State. Also, the dynamic pressure is calculated for beam quartering and bow seas. This is probably a conservative estimate due to the assumption that the angle of incoming waves is perpendicular to the wall.

Table D.3.6.2: Sample of Shipmo7 Output and Dynamic Pressure Calculations

Sea State 2				Sig. WH = 0.3					
Speed	Heading	Tp	prob.	Rel. Vert. Displacement			Emerg.	Dynamic pressure	
				Tz*	RMS	Rate/hr.	prob.	Tz	RMS
8	Head	10.14	0.172	9.40	0.070	0.0	0.0000		
8	Stbd.Bow	10.14	0.491	9.10	0.071	0.0	0.0000	6.07	1128.0
8	Stbd. Bm	10.14	0.564	9.80	0.076	0.0	0.0000	7.71	1128.0
8	Stbd.Quat.	10.14	0.882	12.70	0.045	0.0	0.0000	10.34	1128.0
8	Follow.	10.14	0.205	14.40	0.063	0.0	0.0000		
Sea State 3				Sig. WH.= 0.88					
Speed	Heading	Tp	prob.	Rel. Vert. Displacement			Emerg.	Dynamic pressure	
				Tz	RMS	Rate/hr.	prob.	Tz	RMS
8	Head	10.14	0.368	9.40	0.205	0.0	0.0000		
8	Stbd. Bow	10.14	0.841	9.10	0.208	0.0	0.0000	6.07	3307.0
8	Stbd. Bm	10.14	0.972	9.80	0.222	0.0	0.0000	7.71	3307.0
8	Stbd.Quat.	10.14	1.613	12.70	0.132	0.0	0.0000	10.34	3307.0
8	Follow.	10.14	0.363	14.40	0.185	0.0	0.0000		
Sea State 5				Sig. WH.= 3.25					
Speed	Heading	Tp	prob.	Rel. Vert. Displacement			Emerg.	Dynamic pressure	
				Tz	RMS	Rate/hr.	prob.	Tz	RMS
12	Head	10.14	3.783	8.80	0.809	26.7	0.0344		
12	Stbd. Bow	10.14	9.055	8.80	0.875	37.1	0.0561	5.49	12213.0
12	Stbd. Bm	10.14	10.755	9.80	0.826	19.7	0.0396	7.71	12213.0
12	Stbd.Quat.	10.14	17.146	14.70	0.560	0.3	0.0009	12.70	12213.0
12	Follow.	10.14	3.91	17.70	0.717	2.9	0.0137		
Sea State 6				Sig. WH.= 5					
Speed	Heading	Tp	prob.	Rel. Vert. Displacement			Emerg.	Dynamic pressure	
				Tz	RMS	Rate/hr.	prob.	Tz	RMS
16	Head	10.14	3.611	8.50	1.338	241.5	0.2918		
16	Stbd. Bow	10.14	6.116	8.50	1.475	248.6	0.3628	5.00	18790.0
16	Stbd. Bm	10.14	8.122	9.70	1.271	127.3	0.2555	7.71	18790.0
16	Stbd.Quat.	10.14	13.071	16.90	0.888	13.3	0.0612	15.68	18790.0
16	Follow.	10.14	2.878	21.80	1.130	28.1	0.1779		
12	Head	15.2	0.904	10.50	0.844	29.1	0.0454		
12	Stbd. Bow	15.2	1.531	10.30	0.966	50.7	0.0943	6.07	11400.0
12	Stbd. Bm	15.2	2.034	11.90	1.235	76.3	0.2355	8.33	11400.0
12	Stbd.Quat.	15.2	3.273	16.90	0.658	1.5	0.0061	13.01	11400.0
12	Follow.	15.2	0.721	20.40	0.692	2.1	0.0100		

D.3.7 Calculation of Stress Ranges

A sketch of the structural detail for which the stress ranges due to outside lateral pressure was calculated is given in Figure D.3.7.1. For a bracket termination on top of stiffener, the stresses to be considered related to lateral pressure are due to:

- Local stiffener bending;
- Relative deflection between web frame and transverse bulkhead;
- Panel bending - single hull vessel.

All three components will be considered in this example. Theoretical background and appropriate equations have been introduced in Section C.

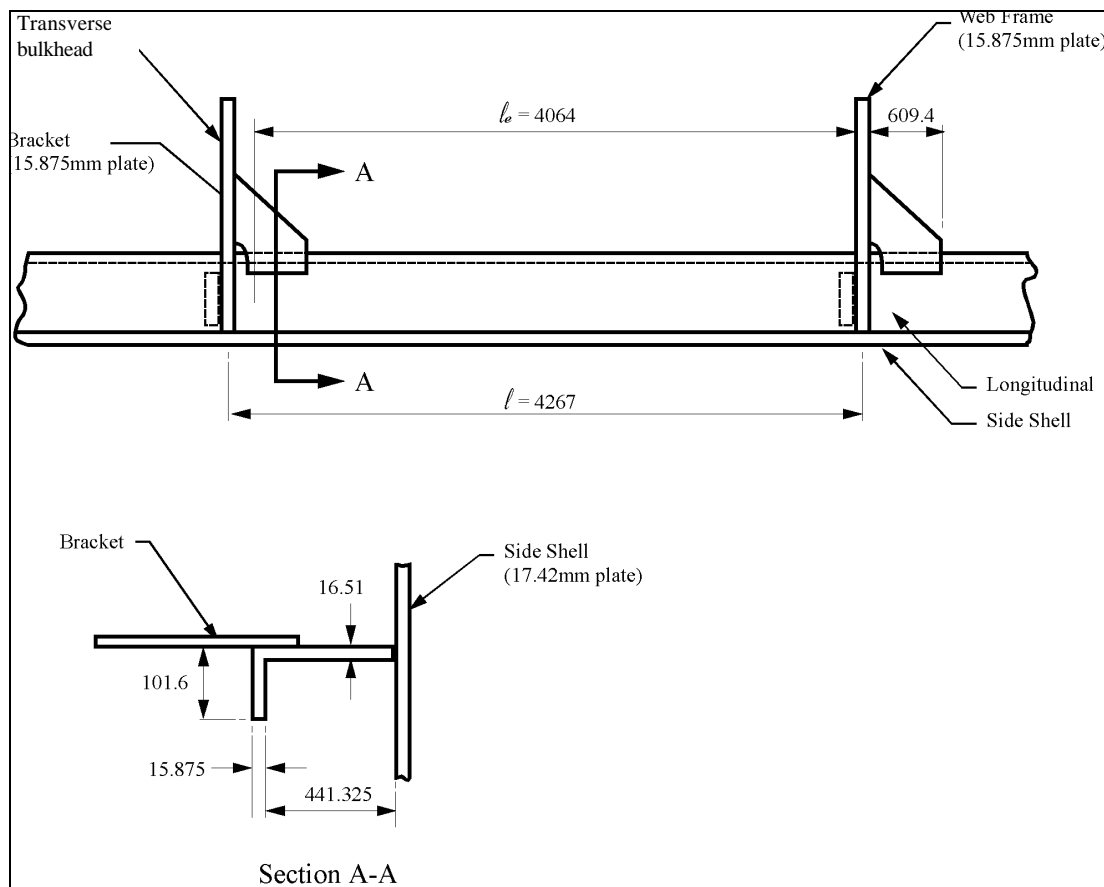


Figure D.3.7.1: Description of Structural Detail

D.3.7.1 Stiffener Bending

Stresses due to local stiffener bending can be calculated from (see the first part on the right hand side of the equation C.4.8):

$$s_2 = K \frac{M}{Z_S} \quad (D.3.13)$$

where:

K = Stress concentration factor

The hot spot stress is related to the nominal stress using a factor K as:

$$K = K_g K_n$$

and where factors are defined as following:

- Term K_g is taken as per stiffener support K-factors [D8]. For the case of stiffener bending due to lateral load, K_g is taken as 1.8¹. An additional factor of 1.15 is used to take into account bracket/stiffener overlap (see figure D.3.7.1).
- K_n represents additional stress concentration factor due non-symmetrical stiffener and is calculated as per [D.8, Section 7]:

$$K_n = \frac{1 + \beta^2}{1 + \beta^2 \mu}$$

where:

$$\mu = \frac{3(1 + \frac{\mu}{280})}{1 + \frac{\mu}{40}} \quad \mu = \frac{l_e^4}{b_f^3 t_f h^2 [\frac{4h}{t_w^3} + \frac{s}{t_p^3}]}$$

$$\beta = 1 - \frac{t_w}{b_f} \quad \beta = \frac{(h - t_f)^2 t_w}{4Z_S}$$

¹. Ref. D.8 for the case considered recommended K factor $K_g K_w = 2.7$. Here K_w is related to the local geometry of the notch. Since in this Guide notch stress concentrations are not considered in fatigue calculations, K_w contribution needed to be extracted. K_w stress concentration factor equals 1.5 if not stated otherwise [D.8], therefore K_g alone equals 1.8.

M Moment at stiffener support adjusted to hot spot position at the stiffener (e.g. bracket toe). It is calculated as (see Equation C.4.8):

$$M = \frac{ps\ell_e^2}{12} r_p$$

p is lateral dynamic pressure

s stiffener spacing

ℓ_e effective span of stiffener

r_p Moment interpolation factor for interpolation to hot spot position along the stiffener length (see figure C.4.3.3)

$$r_p = 6\left(\frac{x}{\ell_e}\right)^2 - 6\left(\frac{x}{\ell_e}\right) + 1.0$$

Z_S Section modulus of stiffener with associated effective plate flange.

The properties of the stiffener containing the POI are listed in Table D.3.7.1.

Table D.3.7.1: Geometry of the Stiffener Considered

Stiffener spacing:	$s = 914.4$ mm
Web frame spacing:	$l = 4267$ mm
Effective span of stiffener:	$\ell_e = 4064$ mm
Location (POI):	Panel CG @ 2500 mm below still waterline Longitudinal location @ 30 m aft from FP
Stiffener moment of inertia	Trans. Axis $I = 4.23 \times 10^8$ mm ⁴
Sectional Modulus Z_S :	1.44×10^6 mm ³ (flange top) 2.33×10^6 mm ³ (baseline)
x - position:	$x = 406.4$ mm
Plate thickness:	$t_p = 17.4$ mm
Height of stiffener:	$h = 441.3$ mm
Web thickness:	$t_w = 16.5$ mm
Flange width;	$b_f = 101.6$ mm
Flange thickness:	$t_f = 15.9$ mm
Bracket thickness:	$t_b = 15.9$ mm

Calculations:

$$r_p = 6\left(\frac{406.4}{4064}\right)^2 - 6\left(\frac{406.4}{4064}\right) + 1.0 = 0.46$$

$$K = K_g \cdot 1.15 \cdot K_n$$

Using the data from Table D.3.7.1 K_n term is calculated as follows:

$$\mu = 148.9$$

$$\gamma = 0.9728$$

$$\beta = 0.8376$$

$$\gamma = 0.5184 @ Z_S = 1.44 \times 10^6$$

$$K_n = \frac{1 + 0.9728 \cdot 0.8376^2}{1 + 0.9728 \cdot 0.8376^2 \cdot 0.5184} = 1.243$$

$$K = K_g \cdot 1.15 \cdot K_n = 1.8 \cdot 1.15 \cdot 1.243 = 2.573$$

Stresses due to stiffener bending per unit lateral dynamic pressure (1 kN/m²) is (Equation D.3.13):

$$s_2 = K \frac{ps \ell_e^2}{12Z_S} r_p = 2.573 \frac{1 \cdot 10^3 \cdot \frac{1}{10^6} \cdot 914.4 \cdot 4064^2}{12 \cdot 1.44 \cdot 10^6} \cdot 0.46 = 1.034 \left[\frac{\text{N}}{\text{mm}^2} \right]$$

D.3.7.2 Stress due to Relative Deflection between Web Frame and Transverse Bulkhead

Stresses due to the relative deflection between the transverse bulkhead and the web frame can be approximated according to the following formula (see the second part on the right hand side of the equation C.4.8):

$$s_d = K \frac{m_d EI}{\ell_e^2 Z_S} r_d d \quad (\text{D.3.14})$$

$$K = K_g \cdot 1.15$$

Here K_g is taken as 1.467 [Reference D8] recommends $K_g K_w$ as 2.2 for stress in longitudinal direction, and $K_w=1.5$). As before factor of 1.15 is taken to account for the bracket/stiffener overlap.

m_δ Moment factor due to relative deflection between transverse supports (taken as 4.4 – no stringers supporting the frames [D.8]).
 r_δ Moment interpolation factor for interpolation to hot spot position along the stiffener length (see equation C.4.8).

$$r_d = 1 - 2\left(\frac{x}{\ell_e}\right)$$

δ Deformation of the nearest frame relative to the transverse bulkhead. According to Reference D.8, δ can be approximated as follows:

$$d = \left(1 - \left(1 - \frac{2z}{D}\right)^2\right) d_m \quad (\text{mm})$$

and

$$d_m = \frac{110 l_s^3 D_m^2}{E I_b \sqrt{1 + N_s}} p \quad (\text{no side stringers})$$

Here:

N_s Number of cross ties in the web frame
 l_s Span between bulkhead and transverse frame
 D_m Depth of ship in meters.
 z Vertical distance from baseline to considered longitudinal.

Data for these values can be found in Table D.3.7.2.

Table D.3.7.2: Web Frame and Panel Geometry

Longitudinals:	
Stiffener spacing:	$s_a = 914.4 \text{ mm}$
Stiffener moment of inertia	$I_a = I_{na} = 4.23 \times 10^8 \text{ mm}^4$
Smear out stiffness	$i_a = 4.23 \times 10^8 / 914.4 = 4.626 \times 10^5 \text{ mm}^3$
Moment of inertia of effective breadth w.r.t. N.A:	$I_{pa} = 1.51 \times 10^8 \text{ mm}^4$
Sectional Modulus Z_s :	$1.44 \times 10^6 \text{ mm}^3$ (flange top) $2.33 \times 10^6 \text{ mm}^3$ (baseline)
Effective span of stiffener:	$\ell_e = 4064 \text{ mm}$
x - position:	$x = 406.4 \text{ mm}$
Length of panel	$a = 29870 \text{ mm}$
Transverses:	
Web frame spacing:	$s_b = 4267 \text{ mm}$
Web moment of inertia:	$I_b = I_{nb} = 5.21 \times 10^{10} \text{ mm}^4$
Moment of inertia of effective breadth w.r.t N.A:	$I_{pb} = 1.68 \times 10^{10} \text{ mm}^4$
Smear out stiffness	$i_b = 5.21 \times 10^{10} / 4267 = 122.10 \times 10^5 \text{ mm}^3$
Sectional Modulus Z_s :	$3.93 \times 10^7 \text{ mm}^3$ (flange top) $8.47 \times 10^7 \text{ mm}^3$ (baseline)
Distance between bulkhead and transverse web:	$\ell_s = 4267 \text{ mm}$
Width of panel	$b = 12495 \text{ mm}$
Vessel depth	$D_m = 18.59 \text{ m}$
Number of cross ties	$N_s = 0$
	$z = 9.9 \text{ m}$

Calculations:

Using the above equations and tabulated data, together with unit lateral dynamic pressure of 1 kN/m^2 the following results are obtained:

$$\begin{aligned} \delta_m &= 2.699 \times 10^{-4} \text{ mm} \\ \delta &= 2.687 \times 10^{-4} \text{ mm} \\ r_\delta &= 0.8 \end{aligned}$$

Stresses due to relative deflection between the transverse bulkhead and the web frame are (equation D.3.14):

$$s_d = K \frac{m_d EI}{\ell_e^2 Z_s} r_d d = 1.467 \cdot 1.15 \frac{4.4 \cdot 2.1 \cdot 10^5 \cdot 4.23 \cdot 10^8}{4064^2 \cdot 1.44 \cdot 10^6} \cdot 0.8 \cdot 2.687 \cdot 10^{-4} = 5.960 \cdot 10^{-3} \left(\frac{\text{N}}{\text{mm}^2} \right)$$

D.3.7.3 Panel Bending – Single Hull Vessel

The equation for secondary bending stress in side shell panels at the intersection with transverse bulkheads (web frames) (see Equation C.4.4) is:

- Longitudinal Secondary Bending Stress:

$$s_{2L} = K \frac{K_b p b^2 r_a}{\sqrt{i_a i_b}} \quad (\text{panel longer than wide}) \quad (\text{D.3.15})$$

Here p is dynamic lateral pressure. In the calculation it will be taken as a unit pressure of $1 \times 10^3 \text{ N/m}^2$.

The stress concentration factor K is:

$$K = K_g \cdot 1.15$$

where K_g is taken as 1.467, and 1.15 to account for bracket/stiffener overlap.

The term K_b in Equation (D.3.15) is function of the aspect ratio ρ , panel's boundary conditions and torsion coefficient η (see Table C.4.3.2).

$$\rho = \frac{a}{b} \cdot \sqrt[4]{\frac{i_b}{i_a}} = \frac{29870}{12495} \cdot \sqrt[4]{\frac{122.10 \times 10^5}{4.626 \times 10^5}} = 2.52$$

$$\eta = \sqrt{\frac{I_{pa} I_{pb}}{I_{na} I_{nb}}} = \sqrt{\frac{1.51 \times 10^8 \cdot 1.68 \times 10^{10}}{4.23 \times 10^8 \cdot 5.21 \times 10^{10}}} = 0.34$$

here i_a , and i_b are smeared out stiffnesses calculated in accordance with section C, Table C.4.3.3, Type A panel. Terms I_{pa} and I_{pb} represent moment of inertia of effective breadth of plating alone associated with longitudinal and transverse, respectively (see Table D.3.7.2).

Table D.3.7.3 shows the values of the K_b term used in the calculations.

Table D.3.7.3: K_b values (Taken from Table C.4.3.2)

Boundary Conditions	ρ	$\eta = 0.0$	$\eta = 0.5$
Long edges simply supported	2.5	0.1263	0.1257
Short edges clamped	3.0	0.1248	0.1253
K_b @ $\rho = 2.52$ and $\eta = 0.34$	$K_b = 0.1259$ (calculated)		

In the equation D.3.15, b term is the effective depth of the panel bounded by double bottom and deck, and r_a term is the distance from point considered to the neutral axis of the panel in longitudinal direction (a – stiffener direction). This value is calculated as 292.9 mm (see figure D.3.7.1) and is distance from neutral axis to the outer fiber of the stiffener flange.

From equation (D.3.15) longitudinal secondary bending stress:

$$s_{2L} = 1.467 \cdot 1.15 \cdot \frac{0.1259 \cdot 1 \cdot 10^3 \cdot \frac{1}{10^6} \cdot 12495^2 \cdot 292.92}{\sqrt{4.626 \cdot 10^5 \cdot 122.10 \cdot 10^5}} = 4.087 \left(\frac{\text{N}}{\text{mm}^2} \right)$$

D.3.7.4 Combination of Stress Components

The above calculated stresses due to lateral pressure are combined considering the sign – positive (tension), and negative (compression).

- Stress due to stiffener bending ($\sigma_2 = 1.034 \text{ N/mm}^2$). For an external dynamic lateral pressure load, these will be in compression at the considered point (i.e., bracket termination) [D8].
- Stress due to relative deflection between web frame and transverse bulkhead ($\sigma_\delta = 5.960 \times 10^{-3} \text{ N/mm}^2$). Bending of the stiffener is resulting in compression at the considered point for external pressure loads [D.8].
- Plate/panel bending ($\sigma_{2L} = 4.087 \text{ N/mm}^2$). Bending of the stiffener results in tension at the considered point of the stiffener [D.8].

Therefore, for a unit external lateral pressure of 1 kN/m^2 exerted uniformly on the side shell panel, the equivalent stress at the intersections with transverse web (in a way of tripping brackets) is

$$-1.034 - 5.960 \cdot 10^{-3} + 4.087 = 3.047 \left(\frac{\text{N}}{\text{mm}^2} \right)$$

D.3.7.5 Calculation of Stress Ranges

This “transfer function” of 3047 kN/m^2 will be applied to both quasi-static (Equation C.3.13), and dynamic pressure (equation C.3.15). The stress range response of the structural detail was assumed to be Rayleigh distributed within each stationary condition. The stress range distribution (defined in terms of stress range bins) for a given stationary condition is (see Section C.5.3):

$$Q'(s_{mid})_{short} = \left[1 - e^{\left(\frac{-s_{upper}^2}{2m_0}\right)} \right] - \left[1 - e^{\left(\frac{-s_{lower}^2}{2m_0}\right)} \right] = F_{\gamma_s}(s_{upper}) - F_{\gamma_s}(s_{lower}) \quad (D.3.16)$$

where σ_{mid} , σ_{upper} and σ_{lower} are stress range bin's mid, upper and lower value. $Q'(\sigma_{mid})_{short}$ represents stress range bins probability distribution for a stationary condition. The m_0 is the stress response zeroth moment obtained from:

- Quasi-static Pressure:

$$m_0 = \text{RMS}_{\text{Quasi-Stat.press.}}^2 \cdot \frac{1}{10^3} \cdot 3047$$

- Dynamic Pressure:

$$m_0 = \text{RMS}_{\text{Dyn.press.}}^2 \cdot \frac{1}{10^3} \cdot 3047$$

$\text{RMS}_{\text{Quasi-Stat.Press}}$ is calculated from equation (D.3.5) where $\text{RMS}_{\text{Vert. Displ}}$ is tabulated (similar to the Table D.3.6.2, column 6) and $\text{RMS}_{\text{Dyn.Press.}}$ is also tabulated (similar to Table D.3.6.2, column 10). Division by 10^3 was necessary to bring the dimensions of RMS to kN/m^2 . Calculations using equation (D.3.18) are repeated for all stationary operational conditions. The end result is a $m \times n$ matrix with m rows, indicating number of stationary conditions, and n columns indicating the number of stress ranges (bins). In this example, there was a 175×25 matrix for the static pressure case and a 175×30 matrix for the dynamic pressure case.

The stress range response of the structural detail was assumed to be Rayleigh distributed as the excitation due to wave action is Rayleigh distributed. This is a significant approximation. In reality, the distribution of pressure peaks at POI will not be Rayleigh distributed, as the POI randomly emerges from the water and pressure drops to zero, while vessel motion continues. Thus the vertical motion response used to derive the (static) pressure will be Rayleigh distributed, but in fact the resulting pressure variation will not.

To realistically approximate the stress range distribution, the information regarding probability of POI emergence for each stationary condition (Table D.3.6.2, column 8) was employed. If the probability of emergence for a stationary condition was greater than zero, then the Rayleigh distribution for that stationary condition was “normalized” such that the “area” under the distribution (probability sum of all stress range bins) was equal to:

$$\text{Area} = \sum_1^n Q'(s_{mid})_{short} \cdot \text{prob}_{\text{emergence}} \quad (D.3.17)$$

where n is 25 for the static pressure and 30 for the dynamic pressure cases. The magnitude of $\text{prob}_{\text{emergence}}$ for that stationary condition was then added to the normalised zero stress range bin. The normalised stress range (bins other than zero) distribution for a given stationary condition then becomes:

$$[Q'(s_{\text{mid}})_{\text{Short}}]_{\text{normalized}} = \frac{\sum_1^n Q'(s_{\text{mid}})_{\text{short}} - \text{prob}_{\text{emergence}}}{\sum_1^n Q'(s_{\text{mid}})_{\text{short}}} \cdot Q'(s_{\text{mid}})_{\text{short}} \quad (\text{D.3.18})$$

For the individual stress ranges, the long-term stress probability distribution is found from (see equation C.5.6):

$$Q'(s_{\text{mid}})_{\text{long}} = \sum_{i=1}^{\text{\#of stationary cond.}} \text{prob}_i \cdot \frac{1}{1000} \cdot [Q'_i(s_{\text{mid}})_{\text{short}}]_{\text{normalized}} \cdot r_i \quad (\text{D.3.19})$$

where prob_i is a probability of a stationary condition (see Table D.3.6.2, column 4) and recall that the probability was normalized by a 1000, and the number of stationary conditions was 175. Term r_i is the ratio of zero crossing rate (frequency) to average zero crossing rate:

$$r_i = \frac{f_{Z_i}}{f_Z}$$

Zero crossing periods T_{z_i} are tabulated (similar to Table D.3.6.2, column 5 for quasi-static pressure and column 9 for dynamic pressure), and are related to frequency f_{z_i} through

$$f_{z_i} = \frac{1}{T_{z_i}}$$

The average zero-crossing frequency is calculated from:

$$f_Z = \sum_{i=1}^{\text{\#of sationatycond.}} \text{prob}_i \cdot \frac{1}{1000} \cdot f_{z_i}$$

Again, the number of stationary conditions was 175.

Quasi-static and dynamic long term stress range distribution are shown in Figure D.3.7.2a and D.3.7.2b, and mid value of stress ranges (σ_{mid}) and associated probabilities ($Q'(\sigma_{\text{mid}})_{\text{long}}$) are given in Table D.3.7.4. Note that the stress range bins are not the same for the two cases.

Table D.3.7.4: Stress Ranges and Corresponding Probabilities

Dynamic Pressure		Quasi-Static Pressure	
Stress Range (MPa) σ_{mid}	Probability of Occurrence $Q'(\sigma_{mid})_{long}$	Stress Range (MPa) σ_{mid}	Probability of Occurrence $Q'(\sigma_{mid})_{long}$
3.5	0.2107	3	0.2721
10.5	0.1721	9	0.2116
17.5	0.1470	15	0.1537
24.5	0.1108	21	0.1073
31.5	0.0853	27	0.0761
38.5	0.0669	33	0.0534
45.5	0.0511	39	0.0369
52.5	0.0380	45	0.0257
59.5	0.0281	51	0.0181
66.5	0.0211	57	0.0129
73.5	0.0160	63	0.0092
80.5	0.0123	69	0.0066
87.5	0.0095	75	0.0047
94.5	0.0072	81	0.0033
101.5	0.0055	87	0.0024
108.5	0.0042	93	0.0017
115.5	0.0032	99	0.0012
122.5	0.0024	105	0.0009
129.5	0.0019	111	0.0006
136.5	0.0014	117	0.0005
143.5	0.0011	123	0.0003
150.5	0.0009	129	0.0002
157.5	0.0007	135	0.0002
164.5	0.0005	141	0.0001
171.5	0.0004	147	0.0001
178.5	0.0003	Sum:	0.9995
185.5	0.0002		
192.5	0.0002		
199.5	0.0001		
206.5	0.0001		
Sum:	0.9993		

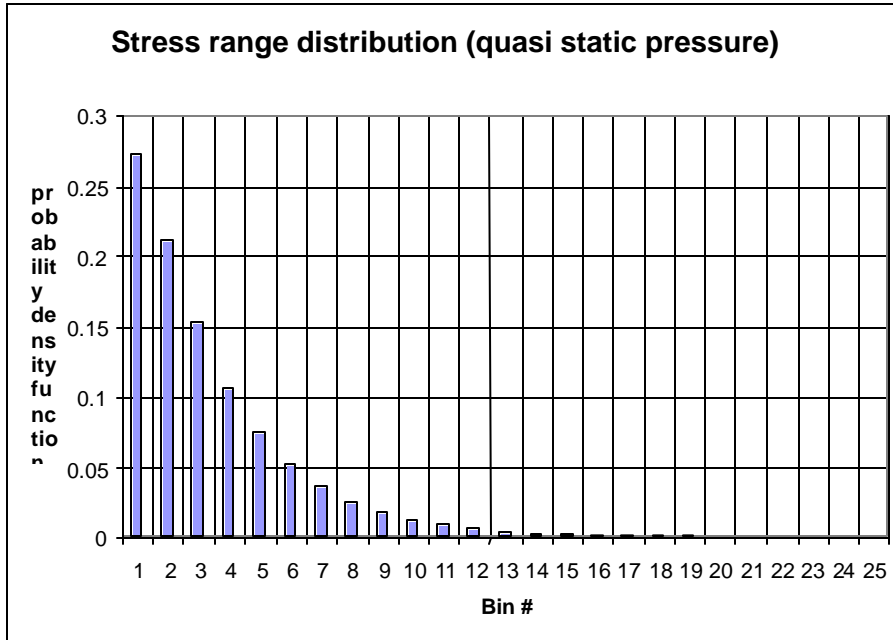


Figure D.3.7.2a: Stress Range Distribution Due to Quasi-Static Pressure

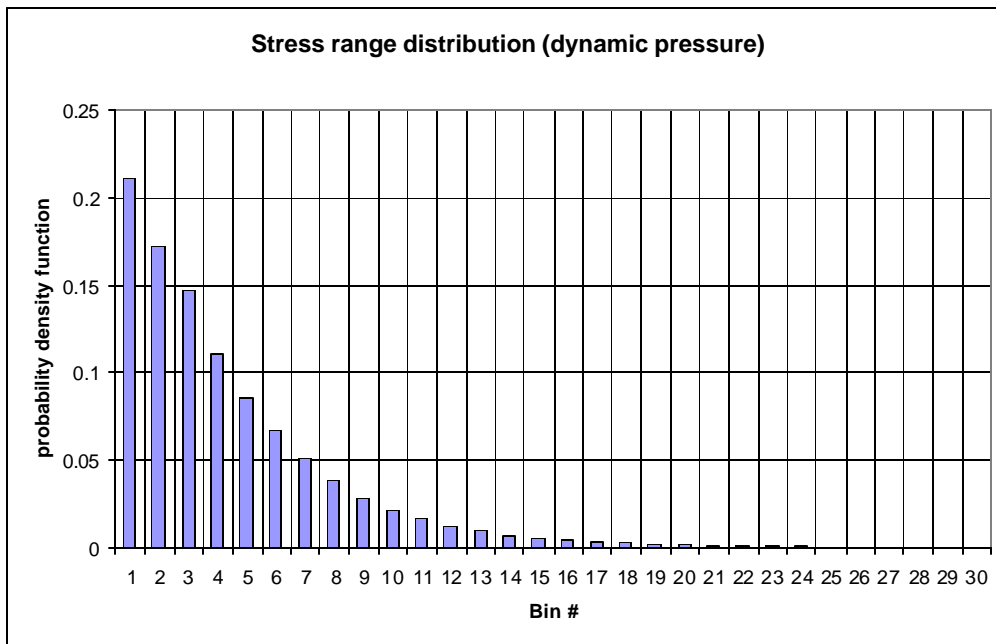


Figure D.3.7.2b: Stress Range Distribution Due to Dynamic Pressure

D.3.7.6 Damage Summation

Accumulation of fatigue damage is assumed to be described by a linear damage accumulation rule (Palmgren-Miner) (see Equation C.6.7):

$$D \cong \sum \frac{n_i}{N_i} \quad (D.3.20)$$

where D is the total damage, N_i is the number of cycles to failure for a particular stress range, and n_i is the number of cycles at that stress range. The parameter N_i was calculated from S-N curve data using:

$$\log(N) = \log(a) + m \log(\sigma) \quad (D.3.21)$$

where the parameters are defined in Table D.3.7.5. Parameters include corrosion fatigue, steel with cathodic protection and protective coating in sea water (see Equation C.6.4a and b).

Table D.3.7.5: Parameters used in Example

N	Log (a) (units in MPa)	m (units in MPa)
$N \leq 10^6$	11.784	-3
$N > 10^6$	15.637	-5

The total number of cycles ship experiences per year (N) is estimated according to:

$$N = \sum_{i=1}^{\text{\# of stationary cond.}} \frac{1 \cdot \text{num. of days at sea} \cdot \frac{\text{prob}_i}{1000} \cdot 24 \cdot 60^2}{T_{z_i}} \quad (D.3.22)$$

where the number of days at sea is given on an annual basis, and prob_i is probability of individual stationary conditions (Table D.3.6.2, column 4). Term T_{z_i} refers to the wave zero crossing period. In this example ship spends 237 days at sea per year. Thus the total number of cycles for all 175 stationary conditions was calculated to be 2.29×10^6 (dynamic pressure), and 2.78×10^6 (static pressure).

Number of cycles at stress range n_i is then estimated as:

$$n_i = F_2 (s_{\text{mid}})_{\text{long}} \cdot N \quad (D.3.23)$$

Table D.3.7.6 shows the calculations of number of cycles to failure N_i for dynamic and static pressure case using equation (D.3.21) and parameters from Table D.3.7.5. Also, Table D.3.7.6 contains the number of cycles at stress range n_i and the relative damage accumulation n_i/N_i .

Table D.3.7.6: Stress Ranges and Corresponding Fatigue Damage

Dynamic Pressure				Quasi-static pressure			
σ_{mid} MPa	n_i	N_i	n_i/N_i	σ_{mid} Mpa	n_i	N_i	n_i/N_i
3.5	4.83E+05	8.25E+12	5.85E-08	3	7.56E+05	1.78E+13	4.24E-08
10.5	3.95E+05	3.40E+10	1.16E-05	9	5.88E+05	7.34E+10	8.01E-06
17.5	3.37E+05	2.64E+09	1.28E-04	15	4.27E+05	5.71E+09	7.48E-05
24.5	2.54E+05	4.91E+08	5.17E-04	21	2.98E+05	1.06E+09	2.81E-04
31.5	1.95E+05	1.40E+08	1.40E-03	27	2.12E+05	3.02E+08	7.00E-04
38.5	1.53E+05	5.13E+07	2.99E-03	33	1.48E+05	1.11E+08	1.34E-03
45.5	1.17E+05	2.22E+07	5.27E-03	39	1.03E+05	4.80E+07	2.14E-03
52.5	8.70E+04	1.09E+07	8.01E-03	45	7.13E+04	2.35E+07	3.04E-03
59.5	6.44E+04	2.89E+06	2.23E-02	51	5.02E+04	1.26E+07	4.00E-03
66.5	4.83E+04	2.07E+06	2.34E-02	57	3.58E+04	7.20E+06	4.97E-03
73.5	3.68E+04	1.53E+06	2.40E-02	63	2.56E+04	4.37E+06	5.85E-03
80.5	2.82E+04	1.17E+06	2.42E-02	69	1.82E+04	2.77E+06	6.58E-03
87.5	2.17E+04	9.08E+05	2.39E-02	75	1.30E+04	1.83E+06	7.09E-03
94.5	1.66E+04	7.21E+05	2.30E-02	81	9.21E+03	1.24E+06	7.40E-03
101.5	1.26E+04	5.82E+05	2.17E-02	87	6.55E+03	8.70E+05	7.53E-03
108.5	9.60E+03	4.76E+05	2.02E-02	93	4.68E+03	6.23E+05	7.50E-03
115.5	7.31E+03	3.95E+05	1.85E-02	99	3.35E+03	4.56E+05	7.35E-03
122.5	5.59E+03	3.31E+05	1.69E-02	105	2.41E+03	3.40E+05	7.10E-03
129.5	4.29E+03	2.80E+05	1.53E-02	111	1.74E+03	2.57E+05	6.75E-03
136.5	3.31E+03	2.39E+05	1.39E-02	117	1.25E+03	1.98E+05	6.34E-03
143.5	2.57E+03	2.06E+05	1.25E-02	123	9.04E+02	1.54E+05	5.87E-03
150.5	1.99E+03	1.78E+05	1.12E-02	129	6.52E+02	1.21E+05	5.37E-03
157.5	1.55E+03	1.56E+05	9.95E-03	135	4.70E+02	9.67E+04	4.86E-03
164.5	1.20E+03	1.37E+05	8.80E-03	141	3.39E+02	7.78E+04	4.36E-03
171.5	9.33E+02	1.21E+05	7.74E-03	147	2.45E+02	6.32E+04	3.88E-03
178.5	7.24E+02	1.07E+05	6.77E-03			Sum:	<u>1.10E-01</u>
185.5	5.62E+02	9.53E+04	5.90E-03				
192.5	4.37E+02	8.53E+04	5.13E-03				
199.5	3.40E+02	7.66E+04	4.44E-03				
206.5	2.66E+02	6.91E+04	3.85E-03				
		Sum:	<u>3.42E-01</u>				
Total Number of cycles (N)=2.29x10⁶				Total Number of cycles (N)=2.78x10⁶			

From Table D.3.7.6 the linear cumulative damage summation given by equation (D.3.22) for two cases is:

- Dynamic Pressure $D = 0.3418$ ($1/D = 2.93$ years) $N = 2.29 \times 10^6$
- Static Pressure $D = 0.1102$ ($1/D = 9.07$ years) $N = 2.78 \times 10^6$

Fatigue life due to dynamic pressure is lower than the fatigue life due to static pressure. However, they are of the same order of magnitude, and thus an indication that both components need to be considered in the fatigue analysis.

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APPENDIX A - COMPARISON OF CURRENT CLASSIFICATION SOCIETY APPROACHES TO FATIGUE DESIGN

A.1 Introduction

One of the first steps in the design process is to determine which key challenges the design will have to overcome. This will dictate the nature of the design approach and the processes used to address each element of it. Often, fatigue is not a major issue. Many small, robust ships, such as tugs and supply vessels, do not have the levels of cyclic stresses which will lead to fatigue problems. In other ships, fatigue may be a localized concern for specific equipment foundations or for other structural components such as shaft brackets. Fatigue in machinery systems or propulsors can also affect ships where structural fatigue is highly unlikely.

All types of localized fatigue can be handled using the same general approaches used in this course. However, they will not be given any more detailed consideration. The focus of this lecture, and those which follow, will be on general ship structures where fatigue performance needs to be validated in order to assure an owner of acceptable through-life capability. This is likely to include all major warships, and larger commercial vessels including tankers, bulk carriers, and container ships.

The level of performance validation required will vary, depending on the ship's configuration and intended service. Accordingly, more or less sophisticated methods of fatigue life assessment may be utilized. Where it is anticipated that highly detailed analyses will be required at some stage in the design process, it is still probable (and highly advisable) that less complex methods be used in the earlier stages, when the overall design parameters are being selected. Detailed analyses should only lead to changes in details, rather than major reworkings of the ship design as a whole.

The designer should therefore have an understanding of the probable extent of a ship's fatigue concerns, the level to which they can be expected to influence other design considerations, and the methods which can be used to manage fatigue design most cost-effectively at each stage in the process. These methods include a variety of design codes, standards, and criteria. A number of the most commonly used of these are described in this section.

A.2 Overview

Many of the existing ship fatigue analysis codes are based on the same knowledge base, and reference (directly or in supporting documents) other non-marine or offshore standards. These other applications generated earlier interest in fatigue either because fatigue failures were more likely to prove immediately catastrophic, or because their economic consequences were more dramatic than was the case with traditional ship construction. Organizations such as the International Institute for Welding (IIW), the European Committee for Standardization, and a variety of national and industry standards bodies produced materials relevant in whole or part to ship structural design. The ship classification societies, such as ABS, DNV, and Lloyds, have applied these materials with greater or lesser degrees of customization, detail, and interpretation.

The commercial ship designer is most likely to apply one or other classification society approach, or to have to have an alternative approach approved by Class at some stage during design. It is quite possible that preliminary design may be conducted using one classification society method, and that final selection of Class will require another. It is therefore useful to have an understanding of the scope of each set of Class Rules and of their similarities and differences in approaches. For those involved in military projects, Class Rules and guidance notes can be valuable reference documents to assist in the design process. An understanding of the background information, including other codes and standards, can also assist in developing approaches to new types of design problem.

[Table A.1](#), drawn from the Proceedings of the 13th ISSC in August 1997 [[Ref C.32](#)], provides a good general summary of the procedures for fatigue assessment of ship structures which are required by the major western classification societies. Most of the Rule systems now in place have been developed and implemented quite recently, and as a result the levels of guidance available (and in some cases the requirements themselves) are still changing fairly rapidly. This table is thus only a snapshot. As can be seen, in some cases (such as the ABS and LR Rules) the use of a fatigue assessment procedure is mandatory for certain sizes and classes of ships. In others, it is always optional. Where fatigue assessments are carried out, the ship may be given a supplementary class notation which acknowledges the procedure. In the case of DNV, it may also allow the assignment of a specific design life in excess of 20 years, which is assumed to be the default. Navy standards, if added to the list, would generally show a focus on detailed analytical treatments.

As indicated in [Table A.1](#), several Rule systems can allow for the treatment of fatigue with varying degrees of design sophistication. These have been categorized by Lloyd's Register as Level 1, 2, and 3 methods, and similar terminology will be used here as a convenient shorthand. Level 1 methods involve the selection of good structural details to minimize the likelihood of problems. Level 2 performs analyses of probable performance using standardized assumptions for loading and response. A full Level 3 analysis will use ship-specific loading data, finite element modelling of global and local response, and realistic representations of any other effects expected to influence the fatigue life. As noted in the Introduction, each and all of these may be appropriate to a particular project at different stages in the design process.

[Table A.2](#), from the same source, provides some more detail on the background to each of the rule systems. Some of the differences in approach are readily apparent, but more are hidden in the details of each method. Several obvious questions could be posed, for example (a) are the resulting differences significant, and (b) if so, which approach is correct?

Table A.1: Procedures for Fatigue Assessment of Ship Structures

Class. Soc.	Reference	Brief description of the scope of the document, applicability and when required.
ABS	ABS (1996 & 1996b)	The fatigue strength assessment is performed in three steps: Step 1 is a designer oriented assessment for connections of longitudinal stiffeners to transverse webs and bulkheads. Step 2 is a simplified fatigue analysis for local hull structures. Step 3 is a comprehensive structural analysis based on spectral approach for details found inadequate in step 2. The procedure is applicable for tankers, bulk carriers and container ship.
BV	BV (1994)	The aim of the procedure is to 'provide the ship designer with relevant information to assess fatigue strength and to define the fatigue design criteria to be applied'.
DNV	DNV (1995)	General background is given the rule requirements for fatigue control of ship structure and detailed recommendations for such control. Various levels of fatigue assessment procedures defined include a simplified approach and a direct calculation approach. Its application is required for structural details 'subject to extensive dynamic loading'.
GL	GL (1997)	Rules for simplified fatigue strength analysis. Its application is required for structures which are 'predominantly subjects to cyclic loads.
LR	LR (1996)	Three Levels are given. Level 1 is based on a comparison of the structural details with recommendations derived from consolidation of available service experience. Levels 2 and 3 are a simplified and full spectral direct calculations procedures. The procedure is developed for a double hull oil tankers and bulk carriers and is under development for container and LNG/LPG ships. Mandatory for new oil tankers and bulk carriers over 190 meters in Length. Level 1 and 2 are to be applied and Level 3 at the request of the ship owner or the shipbuilder.
NK	NK (1996)	A simplified approach for ship design which has been verified for longitudinal stiffeners. Research work is under conduction for improving and revising the guidance. The procedure is applicable for longitudinal, transverse and local strength members of oil tankers, bulk carriers and container ships.
RINA	RINA (1995)	Rules for checking is the fatigue strength of ship hull structures by means of a simplified fatigue analysis. Applicable for ship structures which satisfy RINA standards for obtaining the highest class made of normal and/or high strength steels. Its application is required for the special notation FTC by RINA.
KR	KR (1995)	Guidance for simplified fatigue strength assessment is ship structures at the initial stage.

Table A.2: A Short Summary of Different Fatigue Assessment Procedures Available for Ship Design

Class. Soc.	Loads			Stress anal. guid.		Fatigue strength ¹				Corrosion method	Safety factor ⁵	Program name	Guidance on details
	Basis	Prob.	Shape	nominal	SCF	Nom.	Local ²	mean ³	thick. ⁴				
ABS	Rule	2 10 ⁻⁸	Weib.	simple	Yes	DoE	DoE	no ⁶	spec.case	net ⁷	no	SafeHull	yes
BV	Rule	10 ⁻⁵	Weib.	no	Yes	DoE	DoE ⁸	yes	25mm	time ⁹	no ¹⁰	VeriStar	no
DNV	Rule/Direct	10 ⁻⁴	Weib.	simple/FE	Yes	no	own ¹²	yes	22mm	net ¹²	no	Nauticus	yes
GL	Rule	10 ⁻⁶	Lin.	simple	Yes	IIW	IIW	yes	spec.case	no ¹³	yes ¹⁴	Poseidon	yes (in rules)
LR ¹⁵	Simple/spectral approach ¹⁶			simple/FE	Yes	no	own ¹⁷	no	22mm	net ¹⁸	no	ShipRight	yes (in prog.)
NK	Direct ¹⁹	10 ⁻⁴	Weib.	FE	Yes	BS	BS	yes	no	no ²⁰	yes ²¹	Prime Ship	Yes
RINA	Rule	10 ⁻⁸	Lin.	simple	No	IIW	IIW	yes	no	no	no	no	Yes
KR	Rule	10 ⁻⁴	Weib.	simple	Yes		DoE	yes	22mm	true ²²	no	no	No

1 The S-N data sources are given for nominal (Nom.) and local approaches. BS refers to British Standards 5400, IIW to IIW (1996) and DoE to different editions of the ref. “Offshore Installations: Guidance and Design Construction and Certification”, Health and Safety Executive (formerly Department of Energy), U.K.

2 Local approach is the hot spot method in most cases. Comparison of different local approach S-N curves is given in Figure 7.1.

3 Mean stress correction is applied on the stress range basing on the mean stress or in case of NK (1996) on S-N curve by modifying the scope.

4 The thickness effect se accounted for by a factor on stress range basing on the mean stress or in case of NK (1996) on S-N curve by modifying the slope.

5 Mean minus two standards deviations S-N curves are used in most case. Additional safety factors to the rule are referred here.

6 Not explicitly.

7 The stress calculated for net scantlings are multiplied by a factor of 0.95 to reflect a ‘mean wasted condition’.

8 Special local approach is used based on notch stress which is the structural stress multiplied by a weld factor is 1.96.

9 Corrosion is modelled by multiplying the cumulative damage with a correction factor that is a function of corrosion rate and time.

10 Mean minus one, two or three standard deviation S-N curves for non critical , critical or particular structural members.

11 Special local approach is used based on a notch stress that is structural stress multiplied by a weld factor. Default factor of the weld value is 1.5.

12 Stress are calculated using net scantlings and S-N curves for corrosive environment. A simple approach is given for partially effective corrosive protection.

13 Only implicitly for hold frames in bulk carriers.

14 For non-redundant structures and for some rounded corners with large rider plates.

15 The procedure is available through the use of the ShipRight program.

16 Loads used by voyage simulation used in Level 2, Parametric formulas for ships motions and loads in regular waves. In level 3 direct approach is used.

17 S-N curves are based on parametric formulas of the hot spot SCF’s derived from systematic FE-analysis.

18 In level 2 time invariant simulation of thickness reduction due to corrosion is used. In level 3 no corrosion modelling is applied.

19 Two approaches defined are ‘a combination’ and ‘design wave’ methods.

20 If considered, in ballast tanks for example, the stresses should be converted to appropriate values and stress safety factor of 1.1 to 1.3 should be considered.

21 Safety factors are used depending on the importance of the member. Explicit values are not given. For basic joints mean S-N curves are used.

22 The stress concentration factors are applied for stress analysis and S-N curve for corrosive environment. A simple approach is given for partially corrosive protection.

Figure A.1 provides a partial answer to the first of these. There is significant variability in the S-N fatigue design curves used by the different classification societies, and although the uses of the curves also differ, the resulting predictions can be more rather than less different. The variability can easily be increased by different interpretations of the input data requirements.

When results from any of the methods are compared with actual service experience, there is considerable scatter, some of which is inherent in the probabilistic approach and some of which may result from uncertainties in aspects such as load modelling or the actual local as-built configuration. Thus, although it can be stated that any and all of the fatigue analysis methods available are based on rational approaches to the problem, at the same time it can also be noted that following any one of them will not necessarily provide conservative predictions for a given structural component. If calculations show the possibility of fatigue problems within the intended service life of the ship, it becomes increasingly important to ensure that the fabrication quality at key details is at least as good as the default assumptions of the analysis method.

A.3 Detailed Comparisons

As there are eight classification society approaches and several other Codes referenced in Table A.1, it would be a lengthy (and confusing) task to present all the differences among the approaches. The focus of these comparisons will therefore be the approach recommended in this Guide, and used in the worked examples in Section D. The Guide approach is developed from basic data derived for British Standard 7608 [C.33], using the same principles applied in some of the more recent offshore. It is closer philosophically to the DNV ship design method [C.3] than to most of the others listed in Table A.1. Some of the most significant differences between the Guide approach and other rule systems commonly used in North America (such as ABS [C.1] and Lloyds [C.2]) will be highlighted; other approaches will only be noted where they offer significant extra assistance in the performance of fatigue analyses.

A.3.1 Load Definition

In all of the approaches to fatigue design, designers are given the option of developing loads using a spectral approach tied to the expected lifetime and operational profile of the ship. This can be utilized with response models of varying degrees of sophistication (linear strip theory, fully non-linear dynamic) and with finite element models of greater or lesser levels of refinement in order to provide the required input stresses for the fatigue damage assessments. However, it will almost always be advisable to perform more simplified load and stress calculations beforehand. These will indicate where the design may be expected to have its most severe fatigue problems, and they will also provide a valuable benchmark against which the more sophisticated results can be compared. Most fatigue (and other) design criteria have been developed using experience, calibrated against relatively simple assessments of loadings. If a sophisticated analysis produces results that are significantly different from expected values, several possibilities should be considered:

- (a) the design (or operation) have unique features which were not adequately recognized in the concept stages;
- (b) the sophistication of the analysis warrants reconsideration of traditional safety factors; or
- (c) there are errors in the analysis.

The simplified load models provided (or recommended) by codes differ in most of their provisions; the differences being smallest - in terms of outcome - when dealing with the primary stresses, and greatest for the tertiary stresses due to dynamic load components. In other areas of design practice using classification society rules, many of the differences will cancel out again due to differences in strength criteria. It is possible that this is also true for aspects of fatigue design. Designers are therefore cautioned that fatigue calculations undertaken with one Class simplified method may not be accepted by another without at least some dialogue.

A significant difference in philosophy between standards comes in using spectral methods due to the differences in the selection of reference stress exceedence probability. Different class rules and naval standards fall anywhere within the range of 10^{-4} to 10^{-8} for this. This Guide uses the 10^{-4} value. This is done for several reasons;

- (a) to highlight the fact that most fatigue crack growth (as opposed to fracture initiation) takes place at relatively low stress ranges; and
- (b) if a more extreme probability is used, the results of fatigue damage calculations will be very sensitive to the spectrum shape parameters, which are often quite uncertain.

A.3.2 Local Idealizations

The Guide is based on using the notch stress, derived from either simple or complex analyses, as the basis for fatigue damage assessment. This approach has been adopted in several codes and standards because it removes a certain amount of subjectivity from the analytical procedure. In the notch stress approach, fatigue damage accumulation is based on $\Delta\sigma_{\text{notch}}$, where:

$$\Delta\sigma_{\text{notch}} = K_w \cdot \Delta\sigma_{\text{hotspot}} = K_g \cdot K_w \cdot \Delta\sigma_{\text{nom}} \quad (\text{A.1})$$

K_w is the local weld configuration stress concentration factor, and K_g is the stress concentration resulting from the detail under consideration. Other fatigue design approaches, meanwhile, continue to use either the hotspot or nominal stress range approaches to predict fatigue. In practice, there are few inherent differences between the approaches until a Level 3 level of sophistication is applied, since tabulated correction factors have to be applied to nominal stresses under any approach.

Joint-specific S-N curves, such as those originally developed for the British Standards for various types of steel construction, are intended for use with the nominal stresses at the crack site, and incorporate K_g and K_w effects. Such curves are used in the Canadian and British naval design standards. A set of these curves is shown in [Figure A.2](#), with two examples of the 'best' and 'worst' typical joint configurations in [Figure A.3](#).

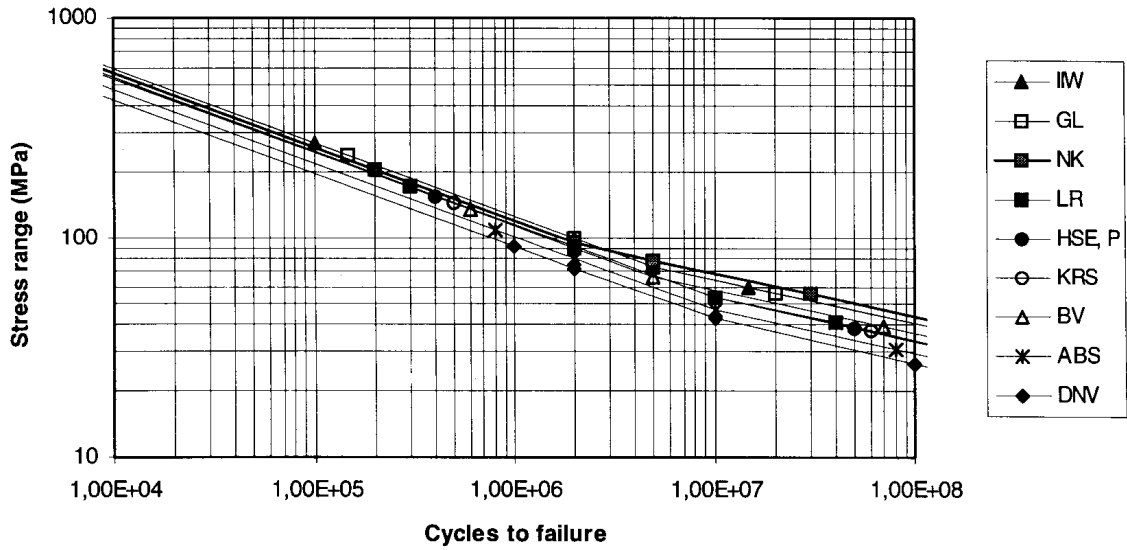


Figure A.1: Hot Spot Fatigue Design Curves (explicit or implicit)

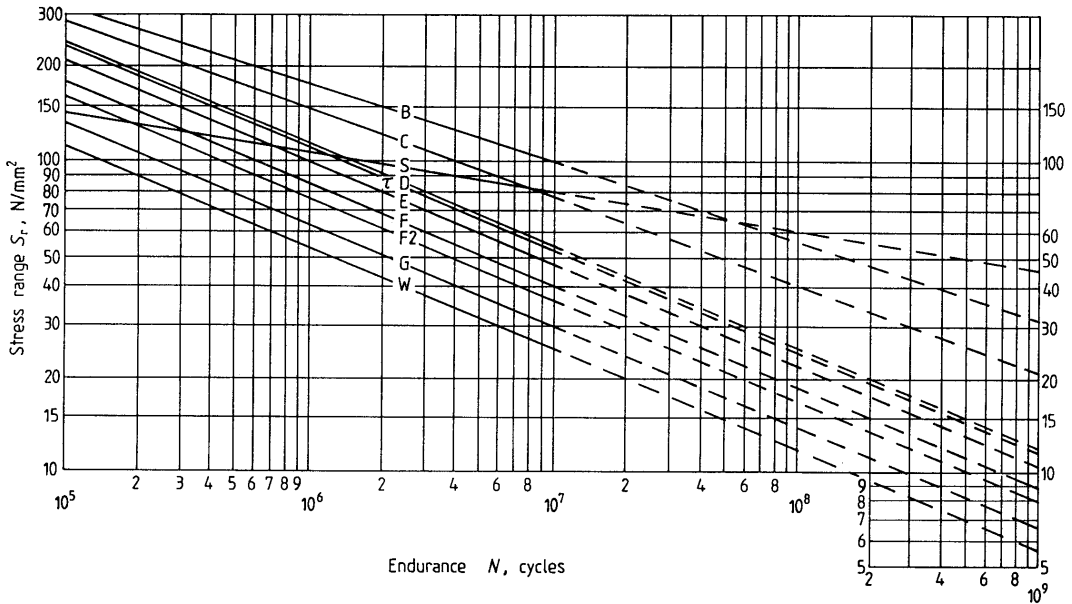


Figure A.2: Example of S-N Curve for Different Joint Configurations

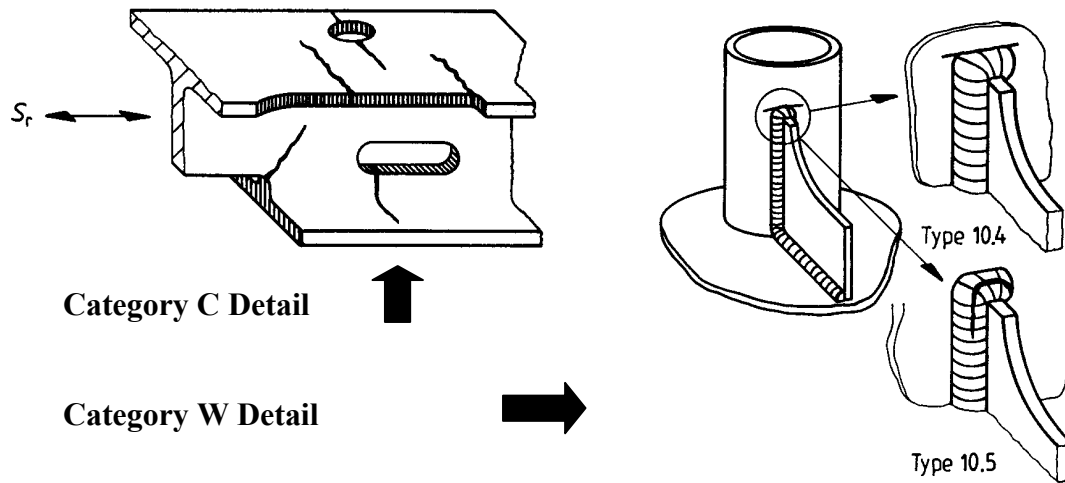


Figure A.3: “Best” and “Worst” Typical Joint Configurations

A problem with using specific curves of these types is that it is often difficult to decide (a) which nominal curve is best suited to the application, and (b) which additional stress concentration effects need to be considered. Unlike the situations shown in [Figure A.3](#), actual ship configurations will normally have multi-axial stress states, in which it is difficult to isolate the stress concentrations built into the S-N curves from those resulting from the real surrounding configuration.

A step beyond the nominal curves is to use the hotspot stresses, which take account of more general configuration effects and thus allow single fatigue design curves to be used for a range of joint types. However, there is no agreement as to which S-N curve is most appropriate to use with hotspot stresses, and this is one of the main reasons for the divergence of the curves shown in [Figure A.1](#). ABS, for example, selects the Category E curve for use with fillet welds, while others are less conservative. Lloyd's Register's Fatigue Design Assessment software incorporates its own S-N curves, which have been derived from model and finite element analyses of typical ship structural details.

Moving to the notch stress approach should allow explicit treatment of all effects except for those due to the welding process (local material properties, grain size, etc). There is still no absolute agreement as to which S-N curve best represents the base case situation, but several organisations have taken the Category D curve, corrected to remove any weld geometry stress concentration effects. The Guide has done this by applying a factor of 1.5, which is very similar to the BV and DNV approaches. This provide the formula:

$$\begin{aligned} &\text{for } N \leq 10^7 \text{ cycle} \\ &\log N = \log a_1 - m_1 \cdot \log \Delta\sigma_{\text{notch}} = 12.710 - 3.0 \log \Delta\sigma_{\text{notch}} \end{aligned} \quad (\text{A.2})$$

$$\begin{aligned} &\text{for } N > 10^7 \text{ cycle} \\ &\log N = \log a_2 - m_2 \cdot \log \Delta\sigma_{\text{notch}} = 15.637 - 5.0 \log \Delta\sigma_{\text{notch}} \end{aligned} \quad (\text{A.3})$$

This type of bilinear formulation is common to all codes, though the location of the break point varies. Codes which follow the International Institute of Welding (IIW) approach use $5 \cdot 10^6$ cycles.

A further aspect of the local idealisation, which may need to be taken into account, is the material thickness. This will affect the through-thickness stress distributions and thus the worst stress concentrations. Most of the ship design approaches use a thickness correction for steel greater than 22-25mm thickness, but for consistency with the basic data set the Guide applies a similar correction above 16mm thickness. This takes the form:

$$\log N = \log a_1 - m_1 \cdot \log \Delta\sigma_{\text{notch}} - 0.25 \cdot m_1 \cdot \log(t/16) \quad (\text{A.4})$$

i.e., the life expectancy is reduced for any thickness, t , greater than 16mm by the italicised term in the equation. Recent work in the offshore industry has suggested that this is non-conservative for very thick structures, and the exponent/coefficient should be raised from 0.25 to 0.3. However, this may only be applicable to the types of joint used in the offshore industry, and no ship rules have yet adopted the larger correction.

A.3.3 Structural Condition

Steel structures in a corrosive environment are much more prone to fatigue failures than are those in dry air. Corrosion has both overall and local effects. The overall reduction in scantlings as thickness is lost increases the global (and hence local stress levels). At the local level, fatigue and corrosion are mutually reinforcing. Stress cycling promotes corrosion, and corrosion accelerates crack formation. Coating breakdown is also most likely to occur at fatigue-prone locations, due to their geometry and also to the higher cyclic strains.

As shown in [Table A.2](#), the majority of class rules take account of both the overall and the local effects, the former by applying a correction for thickness loss through life, and the latter by modifying the S-N curve for the detail under consideration.

Thickness loss corrections are not consistent across rule systems. [Table A.3](#) highlights some of the more significant differences between the ABS and DNV rules, noting at the same time that in other areas their requirements are identical. The same would be true of most other rule comparisons.

Table A.3: Corrosion Thickness Allowances

Structural Component	t_{corr} ABS (mm)	t_{corr} DNV (mm)
Longitudinal bulkhead between cargo/fuel oil tank and dry space	1.5	0
Weather deck beam in ballast tank	2	3
Bottom plating i.w.o. ballast tank	1	1.5
Side shell frames in bulk carrier holds	1.5	1/3*

Note: * refers to upper/lower part of hold

In addition to these direct differences, ABS (uniquely) takes some account of the progressive loss of thickness through life by 're-correcting' the stress levels downwards from the values based on net thicknesses.

At the local level, several codes anticipate that a freely-corroding joint - i.e. one which is not protected by a coating or by an effective cathodic protection system - will have a design life roughly a factor of 2 shorter than the same joint in a non-corrosive environment. This is the assumption recommended in the Guide, though some judgement may still be needed in deciding when to apply it. Modern coating systems, for example in ballast tanks, may have a probable life expectancy lower than that for the ship as a whole. They can be very difficult and costly to reapply after full or partial breakdown. It is therefore realistic to assume that the structural details in such tanks will be protected for part of their life and unprotected subsequently; and the overall fatigue damage accumulation calculations should take account of both periods. DNV provides a simplified method for this that should, in principle, be applicable under any of the rule approaches.

It should be noted that some recent research has suggested that even a factor of 2 life-expectancy penalty may be non-conservative for freely corroding joints in sea water. The most recent issue of UK requirements for offshore structures applies penalties of up to 3 times, and downgrades the assumed effectiveness of cathodic protection.

A.3.4 Safety Factors

Fatigue calculations are probabilistic in nature. The load and stress level predictions are based on statistical representations of lifetime experience. The fatigue design (S-N) curves are developed from experimental data with significant levels of scatter. Source documents for S-N curves will normally quote the mean and standard deviation values for these curves, to allow safety factors to be tuned to the importance of the structural component under evaluation, or the level of structural redundancy present.

The majority of codes take failure probabilities two standard deviations below the mean curve as their default standard, giving a nominal 2.3% failure probability at a calculated damage index of 1 (normal distribution is assumed). Some of the classification society rules and guides note that higher safety factors should be used for the most important details, whose failure could hazard the ship or the safety of personnel. However, only a limited amount of guidance is offered in most cases. Bureau Veritas recommends the use of 1, 2, and 3 standard deviations for non-critical, critical, and very critical details respectively. This equates to 16, 2.3, and 0.14% failure probabilities for each class of detail respectively, or alternatively to approximately successive factors of 1.5 improvements in life expectancy.

It is possible to use the results of a fatigue damage prediction as inputs to a more general structural reliability prediction, which can use fracture mechanics approaches to predict the risk of more catastrophic failures. Such methods can be used to help categorise the different details in the ship and thus to refine the detailed design.

A.4 Summary

Fatigue life prediction is a relatively new consideration for ship design. It is an area in which the different design standards development bodies have been very active in recent years, and where codes are continuing to evolve relatively rapidly. It is still difficult for designers to identify what the criteria should be for a new design, or to assess whether they have been met successfully. Fabrication and maintenance procedures can make enormous differences to the actual performance of nominally equivalent designs.

Existing codes and their accompanying guidance documentation provide useful tools, particularly if they are used in comparative evaluations within and between designs. They can highlight potential problem areas, and guide design development which will keep the risk of fatigue damage to historically acceptable levels. They are less likely to provide accurate predictions of the onset of failure in any individual structural detail.

APPENDIX B

Level 2 Analysis Formulae for Loads

B.1 *Wave- Induced Hull Girder Bending Moments*

- (1) Where direct calculation is not required, the vertical wave induced bending moments may be calculated using the bending moment amplitudes specified as:

$$M_{ds} = -0.11k_{wm} C_w L^2 B (C_B + 0.7) \quad (\text{kNm}) \quad (\text{B.1})$$

$$M_{dh} = 0.19k_{wm} C_w L^2 B C_B \quad (\text{kNm}) \quad (\text{B.2})$$

where:

M_{ds} = wave sagging moment amplitude

M_{dh} = wave hogging moment amplitude

k_{wm} = moment distribution factor

 = 1.0 between 0.40L and 0.65L from A.P., for ships with low/moderate speed

 = 0.0 at A.P. and F.P. (Linear interpolation between these values.)

C_w = wave coefficient

 = 0.0792L; L < 100 m

 = 10.75 - [(300-L) / 100]^{3/2}; 100 m < L < 300 m

 = 10.75; 300 m < L < 350 m

 = 10.75 - [(L-350) / 150]^{3/2}; 350 m < L

and other parameters are as defined in the Nomenclature. Note that these are “permissible” bending moments, i.e. rule limits rather than the actual bending moment developed in a hull under design conditions.

In general, equations B.1 and B.2 are similar for all the major Classification Societies for longitudinal strength. This variation over the length of the vessel is accounted for in the moment distribution factor k_{wm} . If bow-slamming effects are to be considered, the individual Class Rules provide guidance on modifying vertical bending moment envelope. A general discussion is provided in Section 3.2.3 in the main text.

Alternative equations for vertical bending moment have been derived for warships and other slender hulls [Refs. C.19, C.20]. The resulting expressions for predicting the extreme lifetime bending moment including the effects of whipping were then formulated as:

$$M_{dh} = M_{sw} + 0.0006L^{2.5} B \quad (\text{B.3.a})$$

$$M_{ds} = M_{sw} + 0.0009L^{2.5} B \quad (\text{B.3.b})$$

where:

M_{dh} = design hog moment (ton-ft)

M_{ds} = design sag moment (ton-ft)

M_{sw} = stillwater bending moment (ton-ft)

The equivalent expressions for lifetime bending moment (including whipping) in SI units are:

$$M_{dh} = M_{sw} + 0.000115L^{2.5}B \quad [\text{MN}\cdot\text{m}] \quad (\text{B.3.c})$$

$$M_{ds} = M_{sw} + 0.000172L^{2.5}B \quad [\text{MN}\cdot\text{m}] \quad (\text{B.3.d})$$

Implicit in these is an operating life of 3600 days; this translates to approximately 3.888×10^7 encounters, and thus differs from the IACS standard of 10^8 encounters. Refer to Section C.3.1 for an approach to modifying the loads to meet a specific encounter probability.

(2) The horizontal wave bending moment amplitude (M_H) may be obtained from:

$$M_H = 0.22 L^{9/4} (T_{act} + 0.30 B) C_B (1 - \cos(2\pi x/L)) \quad (\text{kNm}) \quad (\text{B.4})$$

Note that the longitudinal distribution is defined in the equation, where x is the distance in metres from the A.P. to the section considered. Horizontal bending moment is of primary concern for slender vessels or open-hatch (large openings) vessels, such as container ships.

(3) Wave torsional loads and moments that may be required for analyses of open type vessels (e.g., container vessels) can be found in Appendices C and D of [Ref C.3].

B.2) Shear Loads

The formulae for shear loads at midship are derived from conventional beam theory, thus:

(1) Vertical Shear:

$$V_{ds} = -0.11k_{ws} C_w LB (C_B + 0.7) \quad (\text{kN}) \quad (\text{B.5})$$

$$V_{dh} = 0.19k_{ws} C_w LB C_B \quad (\text{kN}) \quad (\text{B.6})$$

Again, the Class Rules provide guidance on the distribution of shear over the length of the vessel using a k_{ws} factor.

(2) Horizontal Shear:

$$V_H = 0.22 L^{5/4} (T_{act} + 0.30 B) C_B (1 - \sin(2\pi x/L)) \quad (\text{kN}) \quad (\text{B.7})$$

Torsional Shear formulae can also be obtained from the References if required.

B.3 External Pressure Loads

The external pressure amplitude (half pressure range), p_e , related to the draft of the load condition considered may be taken as:

$$p_e = r_p p_d \quad (\text{kN/m}^2) \quad (\text{B.8})$$

where:

r_p = reduction of pressure amplitude in the wave zone
 = 1.0 for $z < T_{act} - z_{wl}$

$$= \frac{T_{act} + c - z}{2c} \quad \text{for } T_{act} - z_{wl} < z < T_{act} + z_{wl}$$

$$= 0.0 \quad \text{for } T_{act} + z_{wl} < z$$

where:

$$Z_{wl} = \frac{3}{4} (P_{dT}/\rho g); P_{dT} = \text{dynamic pressure at } T_{act}$$

The dynamic pressure amplitude may be taken as the largest of the combined pressure dominated by pitch motion in head/quartering seas, p_{dp} , or the combined pressure dominated by roll motion in beam/quartering seas, p_{dr} , as:

$$p_d = \max (p_{dp}, p_{dr}) \text{ (kN/m}^2\text{)} \quad (\text{B.9})$$

where:

$$p_{dp} = p_1 + 135 \bar{|y|} / (B + 75) - 1.2(T - z)$$

$$p_{dr} = 10 [|\bar{y}| \alpha / 2 + C_B \frac{(|\bar{y}| + k_f)}{16} (0.7 + 2z/T)]$$

$$p_{lp} = k_s C_w + k_f$$

$$= (k_s C_w + k_f) (0.8 + 0.15 V / \sqrt{L}) \quad \text{if } V / \sqrt{L} > 1.5$$

$$k_s = 3C_B + 2.5 / \sqrt{C_B} \quad \text{at AP and aft}$$

$$= 3C_B \quad \text{between } 0.2 L \text{ and } 0.7 L \text{ from AP}$$

$$= 3C_B + 4.0 / C_B \quad \text{at FP and forward}$$

(between specified areas k_s is to be varied linearly)

$$k_f = \text{min of } T_{act} \text{ or freeboard "f" to weatherdeck } \leq 0.8 C_w \text{ (m)}$$

$$\alpha = \text{roll angle, single amplitude (rad)} = 50 z_{wl} / (B + 75) \text{ [rad]}$$

$$\bar{y} = y \geq B/4 \text{ (m)}$$

$$V = \text{ship speed (knots)}$$

$$L = \text{ship length (m)}$$

$$Z = \text{vertical distance from baseline to POI (m)} \leq T_{act} \text{ (m)}$$

B.4 Internal Pressure Loads due to Ship Motion

The dynamic internal pressure amplitude, p_i in kN/m^2 , may be taken as the maximum pressure due to acceleration of the internal mass:

$$p_i = f_a \max \begin{cases} p_1 = r a_v h_s \\ p_2 = r a_t |y_s| \\ p_3 = r a_t |x_s| \end{cases} \text{ (kN/m}^2\text{)} \quad (\text{B.10})$$

where:

$$a_v = \text{combined vertical acceleration (m/s}^2\text{)}$$

$$a_v = \max \begin{cases} \sqrt{a_{rz}^2 + a_z^2} \\ \sqrt{a_{pz}^2 + a_z^2} \end{cases}$$

$$a_t = \text{combined transverse acceleration (m/s}^2\text{)}$$

	$a_t = \sqrt{a_y^2 + (g_0 \sin \alpha + a_{ry})^2}$
a_l	= combined longitudinal acceleration (m/s^2) $a_l = \sqrt{a_x^2 + (g_0 \sin f + a_{px})^2}$
a_p	= tangential pitch acceleration (m/s^2) $= a (2p / T_{pp})^2 R_P$
a_{px}	= longitudinal component of pitch acceleration (m/s^2) $= f (2p / T_{pp})^2 R_{PZ}$
a_{pz}	= vertical component of pitch acceleration (m/s^2) $= f (2p / T_{pp})^2 R_{PX}$
ϕ	= maximum pitch angle (rad) $= 0.25 a_o / C_B$
T_{PP}	= period of pitch (s) $= 1.80 \sqrt{L / g}$
a_x	= surge acceleration (m/s^2) $= 0.2 g a_o \sqrt{C_B}$
a_y	= acceleration due to sway and yaw (m/s^2) $= 0.3 g a_o$
a_z	= heave acceleration (m/s^2) $= 0.7 g a_o / \sqrt{C_B}$
a_o	= acceleration constant $= 3C_W/L + C_V V / \sqrt{L}$
a_{ry}	= horizontal component of roll acceleration (m/s^2) $= a (2 \pi / T_R)^2 R_{RZ}$
a_{rz}	= vertical component of roll acceleration (m/s^2) $= a (2 \pi / T_{Rp})^2 R_{RY}$
R_p	=
R_{pz}	= Vertical distance from the pitch axis of rotation to the local centre of mass or volume (m)
R_{px}	= Longitudinal distance from the pitch axis of rotation to the local centre of mass of volume (m)
C_V	= $\sqrt{L} / 50$, max 0.2
g_o	= $g A a_o$ = acceleration constant
T_R	= period of roll $= 2 k_r / \sqrt{GM}$, maximum 30 (s)
k_r	= roll radius of gyration (m) $= 0.39 B$ for ships with even distribution of mass and double hull tankers in ballast $= 0.35 B$ for single skin tankers in ballast $= 0.25 B$ for ships loaded with ore between longitudinal bulkheads
GM	= metacentric height (m) $= 0.07 B$ in general

- = 0.12 B for single skin tankers, bulk carriers and fully loaded double hull tankers
- = 0.17 B for bulk and ore carriers in the ore loading condition
- = 0.33 B for double hull tankers in the ballast loading condition
- = 0.25 for bulk carriers in ballast condition
- = 0.04 B for container carriers
- a = maximum roll angle, single amplitude (rad)
- = $50 z_{wl} / (B + 75)$
- Z_{wl} = $(1.25 - 0.25 T_R) k'$
- k' = 1.2 for ships without bilge keel
- = 1.0 for ships with bilge keel
- = 0.8 for ships with active roll damping capabilities

In a simplified analysis of bulk or ore cargoes, only p_1 need be considered. A hydrostatic distribution can be considered, although for large granular cargoes, the distribution may in fact be quite different. The appropriate density and pressure height should be specially considered.

Sloshing pressures may normally be neglected in fatigue computations. However, if sloshing is to be considered, formulae for sloshing pressures in partly filled tanks may be obtained from the Classification Society Rules.

In case of partly filled tanks on both sides of a bulkhead, the pressure range may be taken as the sum of the pressure amplitudes in the two tanks.

The above formulations are examples of what may be used for a Level 2 design approach. The designer may refer to Classification Society Rules for alternative formulae. Before doing so he may wish to review the differences by reading Appendix A.

APPENDIX C

STRESS CONCENTRATION FACTORS FOR SHIP STRUCTURAL DETAILS

This appendix provides guidance on the estimation of stress concentration factors (K_g , K_{te} , K_t) for ship structural details where: K_g is a stress concentration factor due to the gross geometry of the detail, K_{te} is an additional stress concentration factor due to eccentricity tolerance (normally used for plate connections only), and K_t is an additional stress concentration factor due to angular mismatch (normally used for plate connections only).

These stress concentration factors account for the local geometry of the detail, excluding the weld (K_w – see Appendix A). They do not account for the global stress concentration effects of the structure surrounding the detail to be analyzed (K_G). The latter should be determined by global FEA or additional published solutions. The total stress concentration factor for the location, used to determine the peak stress in the load carrying section containing the flaw, is thus defined as follows:

$$K_o = K_G \cdot K_w \cdot K_g \cdot K_{yr} \cdot K_t$$

The following SCF solutions have been adapted from Cramer et al. (1995). Alternate solutions may be found in Classification Society documents for fatigue analysis, and previous Ship Structure Committee reports.

Table B2 SCF For Stiffener Supports

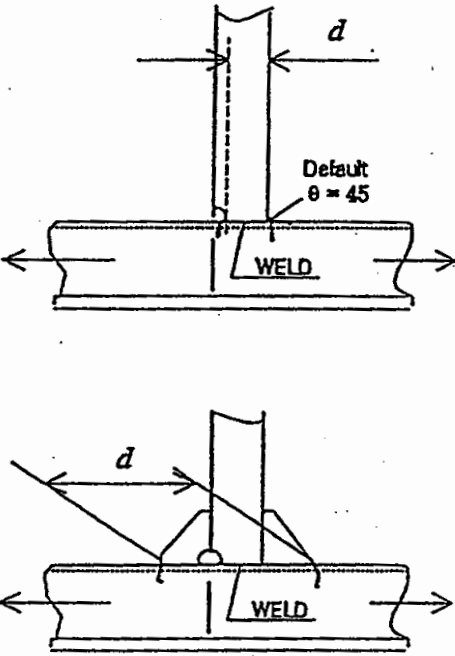
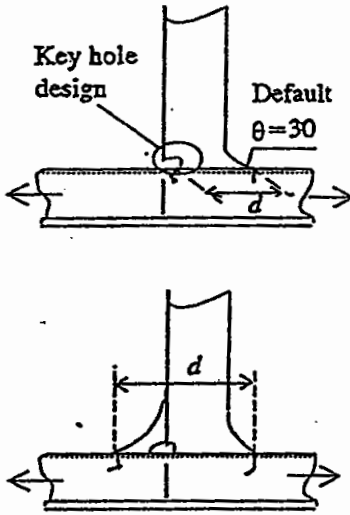
Géometry	K-factor								
<p>B.2.a</p> 	<p>For supporting members welded to stiffener flange:</p> <table border="0"> <tr> <td>$K_x \cdot K_w = 18$</td> <td>$d \leq 50$</td> </tr> <tr> <td>$K_x \cdot K_w = 19$</td> <td>$50 < d \leq 100$</td> </tr> <tr> <td>$K_x \cdot K_w = 2.0$</td> <td>$100 < d \leq 150$</td> </tr> <tr> <td>$K_x \cdot K_w = 2.2$</td> <td>$d > 150$</td> </tr> </table> <p>For supporting members welded to stiffener web by overlap with weld throat thickness as given in $\beta.5a$ (Table $\beta.5$), the above factors are to be multiplied by a factor 1.15</p> <p>Note: The weld connection area between supporting members and stiffener flange must fulfil the requirements in Rules,</p>	$K_x \cdot K_w = 18$	$d \leq 50$	$K_x \cdot K_w = 19$	$50 < d \leq 100$	$K_x \cdot K_w = 2.0$	$100 < d \leq 150$	$K_x \cdot K_w = 2.2$	$d > 150$
$K_x \cdot K_w = 18$	$d \leq 50$								
$K_x \cdot K_w = 19$	$50 < d \leq 100$								
$K_x \cdot K_w = 2.0$	$100 < d \leq 150$								
$K_x \cdot K_w = 2.2$	$d > 150$								
<p>B.2.b</p> 	<table border="0"> <tr> <td>$K_x \cdot K_w = 1.6$</td> <td>$d \leq 50$</td> </tr> <tr> <td>$K_x \cdot K_w = 1.7$</td> <td>$50 < d \leq 100$</td> </tr> <tr> <td>$K_x \cdot K_w = 1.8$</td> <td>$100 < d \leq 150$</td> </tr> <tr> <td>$K_x \cdot K_w = 2.0$</td> <td>$d > 150$</td> </tr> </table> <p>For supporting member welded to stiffener, flange only. It is assumed that the weld is kept clear of flange edge.</p> <p>Note: The weld connection area between supporting members and stiffener flange must fulfil the requirements in Rules.</p>	$K_x \cdot K_w = 1.6$	$d \leq 50$	$K_x \cdot K_w = 1.7$	$50 < d \leq 100$	$K_x \cdot K_w = 1.8$	$100 < d \leq 150$	$K_x \cdot K_w = 2.0$	$d > 150$
$K_x \cdot K_w = 1.6$	$d \leq 50$								
$K_x \cdot K_w = 1.7$	$50 < d \leq 100$								
$K_x \cdot K_w = 1.8$	$100 < d \leq 150$								
$K_x \cdot K_w = 2.0$	$d > 150$								

Table B3 SCF For Termination of Stiffeners on Plates

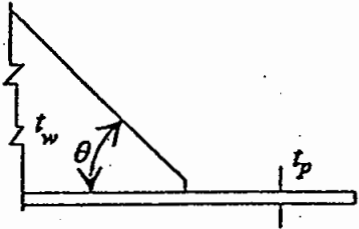
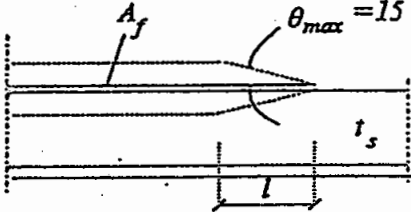
Geometry	K-factor
<p>B.3.a Local elements and stiffeners welded to plates</p> 	$K_s \cdot K_w = 2 \left(1 + \frac{t_w \theta}{t_p 160} \right)$ <p>θ = angle in degrees of sloping termination</p>
<p>B.3.b Sniping of top flanges:</p> 	$K_s \cdot K_w = \frac{3A_f}{lt_s}$ <p>and $K_s \cdot K_w = \min 3.0$</p>

Table B4 SCF For Butt Welds

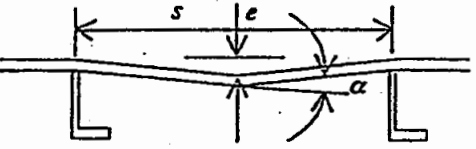
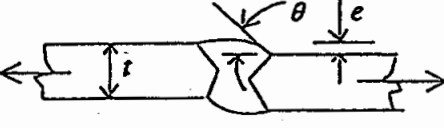
Geometry	K-factor
<p>B.4.a</p>  <p>Default: $e = 6 \text{ mm}$</p>	<p>Angular mismatch in joints between flat plates results in additional stresses at the butt weld and the stiffener</p> $K_{\alpha} = 1 + \frac{\lambda}{4} \alpha \frac{s}{t}$ <p>where:</p> <p>$\lambda = 6$ for pinned ends $\lambda = 3$ for fixed ends $\alpha =$ angular mismatch in radians $s =$ plate width $t =$ plate thickness</p>
<p>B.4.b Welding from both sides</p>  <p>Default: $e = 0.15 t$</p>	<p>$K_g = 1.0$</p> $K_w = 1.0 + 0.5(\tan \theta)^{1/4}$ <p>Default value $K_w = 1.5$ for $\theta = 45 \text{ deg.}$</p> <p>K_{α} from 7.4.a</p> $K_{\alpha} = 1 + \frac{3e}{t}$

Table B4 SCF For Butt Welds (Continued)

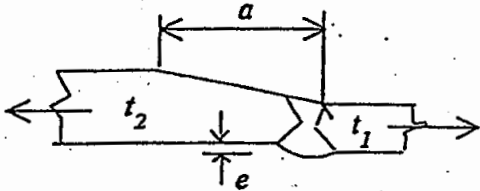
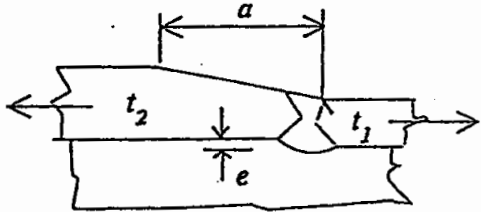
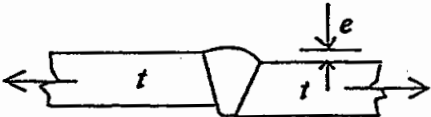
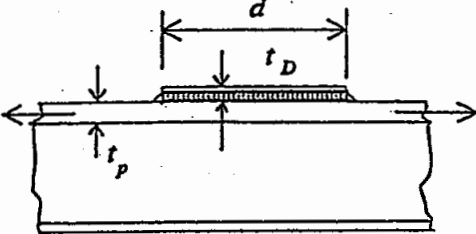
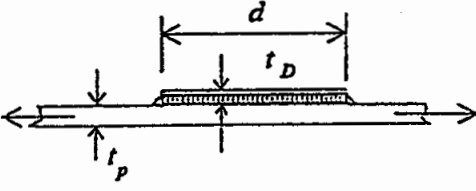
<p>B.4.c Plate not restricted in out-of-plane movement</p>  <p>$e = 0.15 t_1$</p>	$\Delta t = t_2 - t_1$ $K_g = 1 + \frac{3 \frac{\Delta t}{t_1}}{1 + \left(1 + \frac{\Delta t}{t_1}\right)^3}$ $K_w = 1.4 \left(1 + \frac{\Delta t + e}{2a}\right)$ $K_u = 1 + \frac{6 \frac{e}{t}}{1 + \left(1 + \frac{\Delta t}{t_1}\right)^3}$ <p>K_{ta} from B.4.a</p>
<p>B.4.d Plate restricted in out-of-plane movement (e.g. flanges)</p>  <p>Default: $e = 0.15 t_1$</p>	$\Delta t = t_2 - t_1$ $K_g = 1.0$ $K_w = 1.4 \left(1 + \frac{\Delta t + e}{2a}\right)$ $K_u = 1.0$ $K_{ta} = 1.0$
<p>B.4.e Welding from one side</p>  <p>Default: $e = 0.15 t$</p>	<p>Welding from one side is not recommended in areas prone to fatigue due to sensitivity of workmanship and fabrication</p> $K_g = 1.0$ <p>Default value: $K_w = 2.2$</p> $K_u = 1 + \frac{3e}{t}$ <p>K_{ta} from B.4.a</p>

Table B5 SCF For Doubling Plates

Geometry	K-factor
<p>B.5.a</p> <p>Cover plates on beams</p> 	<p>Welded at its end with throat thickness a</p> <p>For $a \geq \frac{t_p \cdot t_D}{t_p + t_D}$</p> <p>$K_s \cdot K_w = 18$ $d \leq 50$</p> <p>$K_s \cdot K_w = 19$ $50 < d \leq 100$</p> <p>$K_s \cdot K_w = 2.0$ $100 < d \leq 150$</p> <p>$K_s \cdot K_w = 2.2$ $d > 150$</p>
<p>B.5.b</p> <p>Doubling plates welded to plates</p> 	<p>For $a \geq \frac{t_p \cdot t_D}{t_p + t_D}$</p> <p>$K_s \cdot K_w = 18$ $d \leq 50$</p> <p>$K_s \cdot K_w = 19$ $50 < d \leq 100$</p> <p>$K_s \cdot K_w = 2.0$ $100 < d \leq 150$</p> <p>For $l > 150$: $K_s \cdot K_w = 2.5 \left(1 + \frac{t_D}{2t_p} \right)$</p> <p>if a more detailed analysis is not performed.</p>

Note: If the welds of the doubling plates are placed closer to the member (flange, plate) edges than 10 mm, the K-factors in Table B.5 should be increased by a factor 1.3

Table B6 SCF For Cruciform Joints

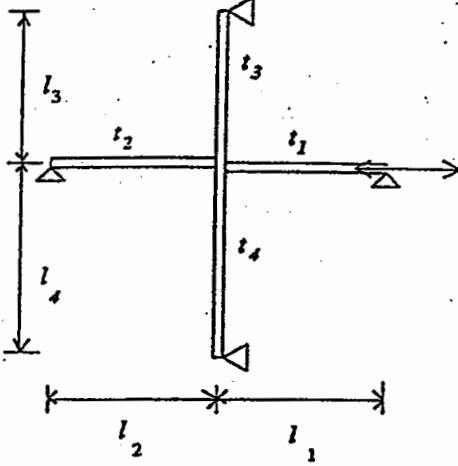
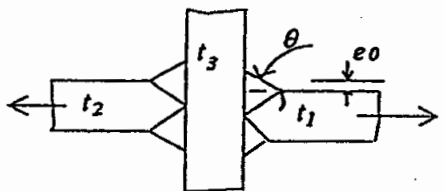
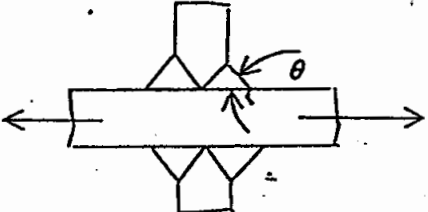
Geometry	K-factor
<p>B.6.a</p> 	$K_u = 1 + \frac{6t^2 \cdot e}{l_1 \left(\frac{t_1^3}{l_1} + \frac{t_2^3}{l_2} + \frac{t_3^3}{l_3} + \frac{t_4^3}{l_4} \right)}$
<p>B.6.b</p>  $e = \frac{t_1}{2} + e_0 - \frac{t_2}{2}$ $t_1 \leq t_2$ $e_0 \leq 0.3t_1$	$K_f = 1.0$ $K_w = 0.90 + 0.90(\tan \theta)^{1/4}$ <p>Default value: $K_w = 1.8$</p> $K_u \text{ from B.6.a}$ $K_{tc} = 1.0$
<p>B.6.c</p>  <p>Applicable also for fillet welds</p>	$K_f = 1.0$ $K_w = 0.90 + 0.90(\tan \theta)^{1/4}$ <p>Default value: $K_w = 1.8$</p> $K_u = 1.0$ $K_{tc} = 1.0$

Table B6 SCF For Cruciform Joints (Continued)

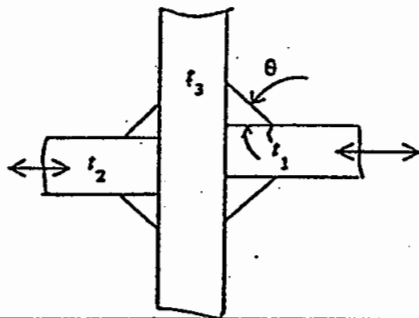
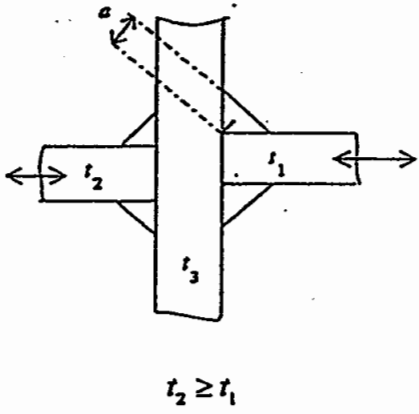
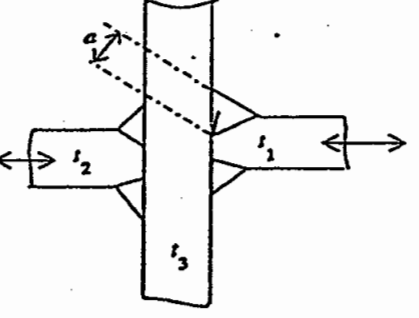
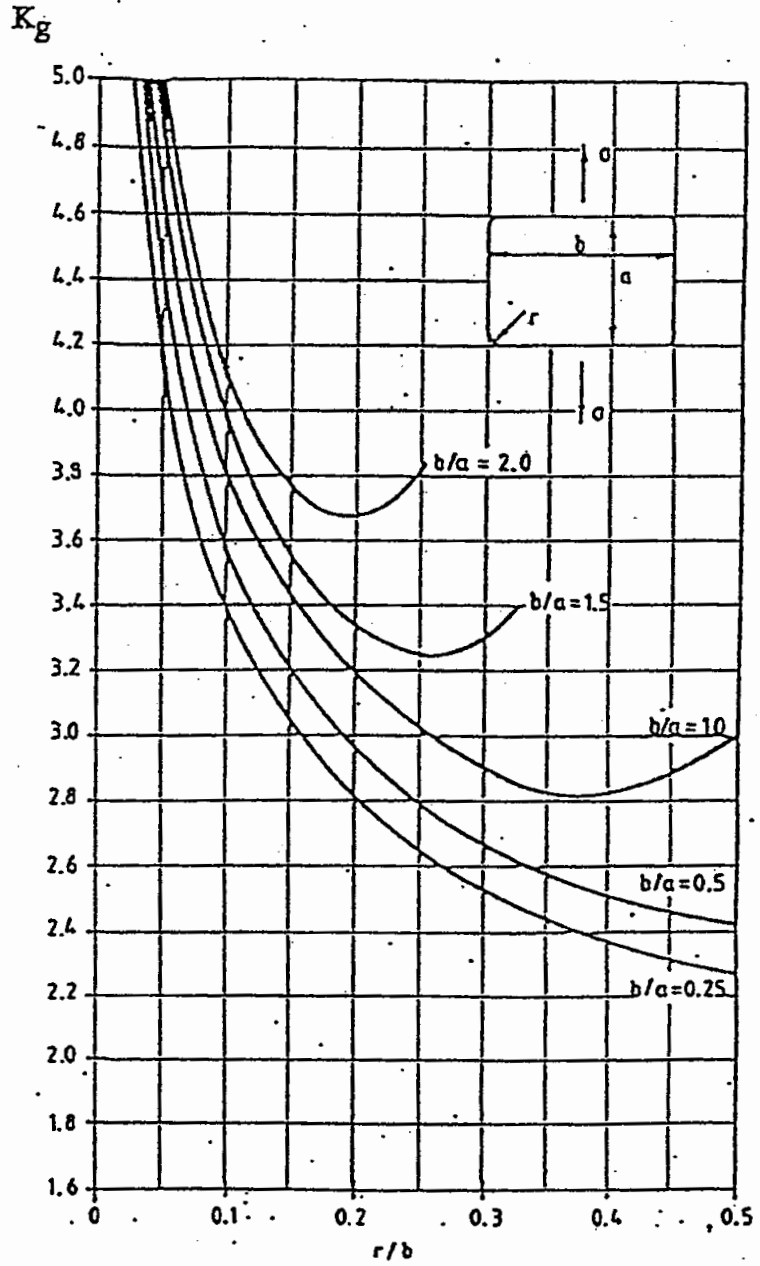
<p>B.6.d</p> 	$K_z \cdot K_w = 1.2 + 1.3(\tan \theta)^{1/4}$ <p>Default value : $K_z \cdot K_w = 2.5$</p> <p>K_w from $\beta.6.a$ with e as given in $\beta.6.b$</p> $K_{tz} = 1.0$
<p>B.6.e</p>  <p>$t_2 \geq t_1$</p>	<p>Based on nominal stress in member with thickness t_1</p> $K_z \cdot K_w = 12 \frac{t_1}{a}$ <p>K_w from $\beta.6.a$ with e as given in $\beta.6.b$</p> $K_{tz} = 1.0$
<p>B.6.f</p> 	$K_z \cdot K_w = 12 \frac{t_1}{a}$ <p>K_w from $\beta.6.a$ with e as given in $\beta.6.b$</p> $K_{tz} = 1.0$

Table B7 SCF For Cut Outs



PROJECT TECHNICAL COMMITTEE MEMBERS

The following persons were members of the Project Technical Committee that represented the Ship Structure Committee to the Contractor as the resident subject matter experts. As such they performed technical review of the initial project proposals to select the contractor, advised the contractor in cognizant matters pertaining to the contract of which the agencies were aware, performed technical review of the work in progress, and edited the final report.

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Carderock Division, Naval Surface Warfare Center

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U. S. Coast Guard

Mr. Jack Waldman
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Mr. Michael Johnson
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