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A REVIEW OF MONOHULL FSUs AND FPSUs

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A Review of Monohull FSUs and FPSUs

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1. INTRODUCTION

This is a wide ranging review of structural and some associated hydrodynamic aspects of monohulls used for hydrocarbon storage or production.

The purpose of the document is to introduce modern methods of analysis and assessment and to discuss some of the more important results obtained from these analyses. In general these methods are only beginning to be used for design and assessment purposes. In order to cover a lot in a relatively small document the detail has been limited to that required to explain the underlying principles and the more significant results.

The document is based on experience obtained by WS Atkins, from work done over the last fifteen years on semi-submersibles and TLPS and the last five years on ships, for various oil and shipping companies.

1.1 REVIEW OF CURRENT USE OF F(P)SUs

Monohull vessels are used as floating storage units (FSUs) in many parts of the world. The monohull can be specially built for the job but is often a tanker which has been converted from trading to being permanently moored. The mooring system usually has to allow the tanker to weathervane as the wind, current and wave directions change. The tanker is filled as the field produces the oil. At intervals which depend on the rate at which the tanker is being filled a shuttle tanker moors to the storage unit and off loads the oil.

Floating production and storage units (FPSUs) have production facilities on the tanker; eg, the crude oil from the well is separated from water and gas on the tanker rather than on an adjacent fixed or floating platform.

The FSU and shuttle tanker used in the Fulmar field are shown in Figure 1.1. The mooring arrangement in this case is a buoyant column which acts as an upside down pendulum to provide a horizontal stiffness which keeps the FSU tanker in the mean position when there are no environmental forces acting and controls the surge motion in waves. Mooring system designs vary considerably and the design of the mooring will affect the motion and stress response of the tanker. This effect of mooring systems is discussed in the parallel document "A Review of Single Point Moorings".

![Figure 1.1](image)

Fulmar FSU and shuttle tanker
BP's SWOPS vessel (Figure 1.2) is an FPSU intended for small fields. It is dynamically positioned instead of being moored and returns to port to discharge its cargo as opposed to utilizing a shuttle tanker.

Early production and well testing vessels are essentially FPSUs but they are intended to operate on any particular well for a short time in order to test the well and justify a permanent installation. The existing and proposed Petrojarl ships are of this type. They are typically capable of keeping position either with a turret mooring system or using dynamic positioning. They can discharge oil to a shuttle tanker or discharge directly in port.

![Figure 1.2 SWOPS](image)

### 1.2 Description of Hull Construction

Tankers used for offshore storage usually have displacements in the region of 1-300000 tonnes and are about 250-330m long with a beam of about 50m and a draught of 7 to 17m.

The usual layout of existing tankers is shown in Figure 1.3. It includes fore and aft peak tanks used for water ballast and five or seven groups of three tanks down the length of the hull. Each group comprises a centre and two wing tanks. All centre tanks and typically alternate wing tanks are used for oil. The remaining wing tanks are used for water ballast. The proportions of the tanks are arranged to give reasonable ballast and loaded draughts. Before the Marine Pollution (Marpol) regulations of 1973/1978 the same tanks could be used for cargo when loaded and ballast water on the unloaded trip. Pre Marpol tankers used a smaller number of tanks for ballast and generally therefore had larger wing tanks. Additional smaller tanks are located in the stern. These are used for fuel, potable water, lubrication oil etc.

#### 1.2.1 Structural form

In global terms a tanker is a long beam which is subject to biaxial bending, biaxial shear, axial forces and torsion from the combination of deadweight, cargo buoyancy and environmental loads. However the pressure loading is applied to a relatively thin stiffened plate shell structure and the manner in which the loading is transferred into the overall beam action is quite complicated as shown in Figure 1.4. Essentially the pressure is transmitted by plate bending to the longitudinal stiffeners. These transfer the load into the transverse frames which themselves bend and transfer the load as a shear into the side shell and longitudinal bulkheads. Finally the shear causes bending of the hull girder. The interaction of the load transfer mechanisms and the overall hull girder bending is also quite complex. For checking the strength under extreme loading this interaction is not usually specifically taken into account. However for checking fatigue strength this interaction must be taken into account.
Figure 1.3
Typical tanker layout
Figure 1.4
Transfer of pressure forces into hull structure

Note:
(1) External and internal water pressure is applied to plating.
(2) Plating spans between and transfers differential pressure loading to longitudinal stiffeners.
(3) Longitudinal stiffeners span between transverse frames or transverse bulkheads.
(4) Transverse frames span onto longitudinal bulkheads and side shell.
(5) Longitudinal bulkheads and shell transfer carry out of balance forces along the length of the ship.

1.2.2 Details
Connections between the various components of a ship structure are essential parts of the load path described above. The performance of a detail may be quite acceptable when the ship is used for ordinary world wide trading yet quite unacceptable when the ship is moored continuously in a relatively severe environment such as the N. Sea. These details are a common source of fatigue cracking problems, as will be discussed in Section 1.3.

1.2.3 Steel strength
Typical shipbuilding steels are:

a) Mild steel with a yield strength of about 245N/mm².

b) Higher yield steels with strengths of about 315-355N/mm².

Whilst high yield strength steel is about 30-40% stronger than mild steel only a small proportion of this benefit may be of use in practice because:

a) The fatigue resistance of a high yield steel is generally no better than of a mild steel.

b) The design of many components is strongly influenced by buckling. The critical buckling load does not increase with yield stress and drops if a member is made more slender. Therefore if buckling dominated the design there would be no benefit from using higher strength steel. In practice members are designed on the basis of an interaction between yield and buckling and in this case there is some benefit from the higher yield strength.

c) An equal amount of corrosion is more serious in a thinner high yield plate than a thicker mild steel plate of equivalent strength.

d) Brittle fracture may result in a limitation on stress.
1.2.4 Steel toughness
The Second World War Liberty ships demonstrated the phenomenon of brittle fracture caused by a combination of stress concentrations, a continuous welded structure and inadequate resistance to relatively small cracks before the cracks became unstable. As a result of these failures toughness requirements have been used for steel selection which have largely overcome the problem of brittle fracture. However, for the severe fatigue prone environments in which F(P)SUs are sometimes placed the rule requirements need to be checked.

Over the last 3-4 years there has been much discussion about the adequacy of the toughness, particularly of grade A steels where no tests have been performed or where commonly the steel has failed the Charpy test for a higher grade. The practical significance of the steel toughness is discussed further in Sections 6 and 7.

1.3 WORLDWIDE APPLICATION

Floating storage units are used in offshore oil fields all over the world. Most other locations are however less fatigue prone than the North Sea.

1.4 SERVICE RECORDS

The Tanker Co-operative Forum publication (1986) show the types of detail that have caused fatigue problems. A survey reported by Jordan and Krumpen (1984) found that in 437514 tanker connections inspected 3635 were found to be cracked.

1.4.1 General results of confidential studies made by Atkins
Atkins have been involved with a large number of tankers and the general conclusion of these studies are:

a) Fatigue is not properly accounted for by design to classification society rules.
b) Fatigue is usually more of an inconvenience to the operator than a major safety problem, owing to the preferred orientation of the most common fatigue cracks being parallel rather than perpendicular to the applied hull girder bending stresses.
c) In some designs the preferred orientation of the more common fatigue cracks is unfortunately perpendicular to the hull girder bending stresses and in these cases fatigue may lead to a catastrophic brittle fracture of the hull girder.
d) Residual stresses may play an important part in fracture. However they probably shake down during the life of the structure so that the worst combination of fatigue crack and maximum residual stress does not usually coincide.
e) Maintenance repair welding, especially if this involves high restraint (eg. plating replacement) can provide this worst combination of high residual stress and existing fatigue defects. It must therefore be treated with caution.
f) Large hull girder bending stresses can result from the combination of wave frequency bending stress, slamming and welding residual stress. Critical defect sizes in these cases depend on the steel but may commonly be in the region 20-200mm.
g) Inspection access and method is not sufficient to reasonably guarantee that near critical defects will be found in the lower grade ship building steels.
h) Better steel grades eg. E/EH would very significantly improve the reliability of ship hulls by making them less likely to suffer a brittle fracture as a result of quite small fatigue cracks. The larger critical defect sizes are also less affected by the generally localized residual stresses.
i) Decreasing the inspections intervals as the ship becomes older should help to maintain adequate reliability.
j) A draught range will distribute the side shell fatigue damage, i.e. more positions may become fatigue sensitive but the worst location will have a better fatigue life.

k) Considerable benefits accrue from operating at a draught sufficient to avoid most incidences of slamming.

l) Fatigue in trading ships can be more acceptable than in an F(P)SU because repairs in port are easier to perform than repairs on station.
2. NEW CONCEPTS

2.1 DOUBLE SKIN DESIGNS

The US Government has brought in legislation that will require double skins for tankers operating in US waters. The requirement is for two skins on the bottom and side separated by at least 2 metres.

A typical configuration is shown in Figure 2.1.

These requirements are still the subject of considerable controversy.

The advantages are seen to be:

a) Less likelihood of a major oil spill in a grounding, collision or hull shell fracture.
b) Potential for better, more fatigue resistant structural form.

The disadvantages are:

a) Danger of gas accumulation in the double skin compartments.
b) Difficulty of cleaning double bottom in the event of oil leaking into it.
c) Difficulty of gas freeing and checking for gas prior to inspection.
d) Difficulty of rescue in the event of a person being injured in the double bottom space.
e) Radio communication will probably not work in the proposed highly cellular structure of the double skin.
f) More difficult and expensive to construct.
g) More steel area for painting and inspection
h) It has been suggested that it is more difficult to refloat a grounded double skin ship

With careful design the structure can be significantly better than single skin designs because the transverse framing can be much stiffer and so provide a better support to the longitudinal stiffeners. Maximum benefit is obtained with both horizontal flats and transverse webs between the inner and outer plating.

Another benefit is that the double skin can be designed to allow good inspection access, at least to the side shells and bottoms. Access to the side shell is much easier if horizontal flats are provided at a spacing of about 2.5m. Deck head access is still a problem.

2.2 DOUBLE SIDE DESIGN

Because the probability of grounding a permanently moored a F(P)SU is small, the main danger of accidental damage to the outer shell is from collision. It may therefore be argued that a double bottom is unnecessary and that only a double sideshell is then required.
2.3 MID DECK DESIGN

The mid-deck design (from Mitsubishi) is shown in Figure 2.2. It relies on the oil pressure in the lower tanks being less than the external pressure so that in the event of damage to the bottom the oil is retained in the tank. This design has the following advantages/disadvantages compared with the double skin design:

Advantages:

a) Possibly less steel required;
   b) Possibly easier to gas free a lower tank than a double bottom.

Disadvantages:

a) Not so easy to inspect the bottom or the bottom of the mid deck, since both will be oil fouled;
   b) Difficult to remove sludge;
   c) Larger number of tanks and therefore fittings, pipe work etc.

Pollution risk of a mid deck design may be greater or less than from a double skin design depending on the type of incident envisaged.
2.4 OTHER CONCEPTS FOR REDUCING MAJOR SPILLS

A number of proposals have been made for both new designs and retrofits that should reduce pollution in the event of structural damage. It has been proposed that a low pressure be maintained above the oil. This pressure however would be difficult to maintain, especially in the aftermath of an accident.

The POLIS (Pollution Limitation System) proposal also uses a vacuum but in this design special tanks are maintained at a low pressure and in the event of damage to an oil tank a valve is opened which permits oil to flow from this damaged tank to the low pressure tank. The flow is driven partly by the low pressure and partly by gravity.

The SCOL (System for Control of Oil Leakage) proposal is for large diameter valves to be fitted near the ship's bottom and between the oil tanks and the ballast tanks. In the event of damage to an oil tank a valve is opened and the oil flows into the ballast tank. The level drops in the oil tank and the internal oil pressure is then less than the external water pressure so that the outflow of oil should be reduced.
3. STABILITY

Some double skin designs have no longitudinal bulkheads other than those forming the inner skin. The intact and damaged stability of this type of design may be compared with that of a conventional tanker of similar overall dimensions.

Calculations show that:

a) Owing to free surface effects in the very wide tank of the double skin design, it has less intact stability than a conventional tanker. However the stability is still quite acceptable.

b) If damaged such that both inner and outer tanks are flooded then the damaged stability is still adequate and the heel is less for the double skinned than the single skinned design.

c) If both skins are damaged and oil is released until hydrostatic equilibrium is reached then the double skin design could release 3 times the amount of oil of a single skinned design. So although the probability of an oil spill should be significantly reduced by the double skin it is possible that in some types of accident the spill could be worse.
4. ENVIRONMENTAL LOADING
ON HULL GIRDER

The calculation of the environmental loading on a ship's hull may be performed using simplified methods and formulae or by a first principles analysis. In order to perform sensible fatigue and fracture calculations an understanding of the structural behaviour is required. This in turn means that analytical methods are preferable to the simpler rules. Nevertheless there is still considerable choice in the type of analyses that are performed. A fundamental decision is whether to attempt to model all the various effects in one complex analysis or whether to try to separate all the effects. The Author's preference is to calculate them separately and then to estimate the manner and consequences of them acting together.

4.1 RULE APPROACH

Rule approaches were originally based on balancing a ship on a wave of length equal to the ship length and of steepness about 1:20 as shown in Figure 4.1. Shear forces and bending moments are introduced because although the ship's dead and live load is balanced in total by "hydrostatic" pressure, the dead and live load distribution is different to the pressure distribution.

Rules now tend to be based on experiments and hydrodynamic analysis and give the wave vertical bending moment from a simple equation eg.:

\[ M = C \cdot H \cdot B \cdot L \]

where

- C is a constant dependent on the block coefficient
- H is the wave height specified in the rules: typically L/20
- B is the beam
- L is the ship length (between perpendiculars).

For ordinary design purposes the rule approach is satisfactory. It is intended for use as an extreme loading estimate. Although it can also be used as a basis for fatigue calculations a direct calculation method is preferable. This needs to account for local pressure loading, horizontal bending and dynamic (acceleration) effects, as well as vertical bending. First principles analysis is described in the following sections.

![Figure 4.1](image)

Figure 4.1
Early approach to calculation of wave loading
4.2 HYDRODYNAMIC ANALYSIS OF PRIMARY WAVE LOADING

First principles analysis is usually either based on 2-D strip or a full 3-D diffraction analysis. These are equivalent to a linear wave theory with the ship hull as an extra boundary condition and the ship motion in 6 degrees of freedom as an additional unknown. It is usual for these analyses to make the same linearising assumptions as linear wave theory. Higher order diffraction theories, corresponding to higher order wave theories have been and are being derived but are presently still subject to a great deal of theoretical uncertainty and debate. This is because the derivations require an assessment of the important terms in series expansions and it is not always clear which these are. Some higher order effects are important because they cause steady and slowly varying drift forces. These are of primary importance for the mooring design. They are discussed in section 4.4.

4.2.1 Three dimensional linear diffraction analysis

This type of analysis takes full advantage of the linear superposition of solutions that is valid at small wave heights. The calculation procedure (using the source-sink method) is:

a) Discretise the hull into n small areas or ‘facets’. Typically the facet dimensions should be no greater than wavelength/5.

b) Calculate the flow velocities and pressures in incident regular waves, at the facet locations corresponding to the mean position of the hull, but making no allowance for the interaction between the hull and the waves.

c) Calculate the strength of pulsating sources of water which if placed, for example, at the facet centroids would result in no net flow through each facet. The source strengths are calculated to allow for both the incident wave flow and the flow from all the other facet sources. This requires the solution of a set of complex
simultaneous equations of number equal to the number of sources n. The equations are complex because the unknowns represent the amplitude and phase of each of the sources.

Typically this solution requires significant computer time (eg. 20 minutes on a SUN workstation). However, multiple wave directions at a single wave period can be solved with little more computer time than 1 wave direction. Multiple wave periods/frequencies unfortunately do not lead to a similar run time benefit.

d) The combination of (b) and (c) yields the loading on the hull which would occur if the hull was restrained from moving. This cyclic force F can be used to calculate the motion of the hull by solving 6 differential simultaneous equations of the form:

$$M \ddot{x} + C \dot{x} + Kx = F$$  \hspace{1cm} (4.1)

Note that M, C and K in general have structural and hydrostatic/hydrodynamic components.

The mass M is composed of a structural mass matrix and a hydrodynamic added mass matrix.

The damping matrix C is composed of viscous damping and wave making radiation damping which allows for the energy radiated away from an oscillatory hull.

The stiffness matrix K is essentially the hydrostatic stiffness of the hull in the water.

e) The added mass and radiation damping can be calculated by oscillating the facet model, in each of its 6 degrees of freedom, in initially still water and calculating this pressure on the hull using the same technique described in (c). The forces and moments in phase with the hull motion can be interpreted as added mass. The forces 90° out of phase with the hull motion can be interpreted as hydrodynamic radiation damping.

f) Having solved equation 4.1 for the hull motion x, which is a complex vector 6 rows long, the total pressures acting on the hull, allowing for the actual hull motion, are found by adding the results of (d) and (c) to the results of (e) factoried by x from (d).

The diffraction analysis procedure is summarised in Figure 4.3. Note that the procedure works in terms of amplitudes and phases for a particular wave frequency (or wave period). It is therefore a frequency domain method.

4.2.2 Strip theory linear diffraction analysis
This approximate method is applied in a similar manner to the full 3-D theory but the ship is divided longitudinally into a series of transverse strips. Each strip is assumed to be part of an infinitely long prism so that it’s properties can be calculated from a 2-D analysis instead of a 3-D analysis. The diffraction loading, added mass and added damping are therefore calculated by summing the effects on each strip. The final calculation of the rigid body motion is calculated in a manner identical to that used for the 3-D analysis.
4.2.3 Non-linear analysis
Non-linearities occur in real waves partly because of the surface boundary conditions which are non-linear for the larger waves. Usually once non-linearities are considered in the analysis, frequency domain methods have to be replaced by time domain methods. (Frequency domain methods are preferred when possible because they are quicker to perform and give a better insight into the physics). Although, as described above, non-linear diffraction theories have been developed these are not yet satisfactory for routine use. As a partial alternative certain approximate methods may be used. One such method (AQWANAUT/DRIFT) makes the following approximations:

Figure 4.3
Diffraction analysis of hull loading and motion
a) The incident wave pressures are calculated using a higher order wave theory with the pressure loading up to the instantaneous water level on the structure.
b) The diffraction, added mass and damping forces are calculated as for the linear theory and applied according to the instantaneous heading angle.
c) For a sea with a spread of frequencies a single dominant response frequency is chosen for the calculation of added mass and hydrodynamic damping.

This method allows reasonably for the effect of intermittent wetting, which can be an important non-linear effect. It will not fully allow for other non-linear effects such as the second order velocities and accelerations. However the surface non-linearity is quite important and this simple improvement over linear wave theory is useful in practice.

4.2.4 Effect of forward speed
It is not straightforward to incorporate the effect of forward speed because, in addition to the harmonic effect of the waves, the solution involves a steady (relative to the ship) Kelvin wave pattern (Figure 4.4), which would exist even without incident waves. The Kelvin waves have a relatively small wavelength and therefore would require a very fine mesh to generate them properly in the model (see Section 4.2.1a). To avoid this problem it is common to semi-empirically modify the zero forward speed analysis to incorporate forward speed but to ignore the Kelvin wave pattern which is mainly important for resistance calculations.

The simplest approximation is to assume that the forward speed can be assumed to change the apparent wave frequency from the true wave frequency to the encounter frequency.

A better approximation considers each component of the wave loading separately and assigns incident wave frequency or encounter wave frequency as appropriate. A suitable analysis would then use:

a) Incident wave loading: Incident wave length but applied at the encounter frequency.
b) Diffraction wave loading: Encounter frequency wave length and frequency.
c) Added mass and damping: Calculated for the encounter frequency.

The latter method is possibly theoretically rather better but requires more computer time than the first method because different headings result in different encounter frequencies and, as discussed above in Section 4.2.1c, each wave frequency needs a separate set of simultaneous equations to be solved.

Figure 4.4
Kelvin wave pattern
4.2.5 Structural loading and quasi-static structural analysis
For structural analysis purposes a structural model of the ship is required. This model may be a simple stick model (Figure 4.5) or a complex shell model. For tanker types of structures the simple stick model provides very useful information and is recommended, even if a more complicated model is also to be built. (Figure 4.6 shows a shell model of a tanker stern).

Clearly the pressure loading calculated by the diffraction analysis can be applied directly to the shell model but the pressures will have to be integrated to forces and moments for the stick model.

4.2.6 Quasi-static structural analysis
The dynamic response of a ship has two parts:

a) The rigid body dynamics of the whole ship oscillating in the water.
b) The structural oscillation of the hull as a vibrating beam.

The longest structural natural period of a tanker is usually about 2 seconds and in extreme sea states there may not be much dynamic excitation of these natural frequencies. In these circumstances the structure can be analyzed as if the wave loading were applied statically. This type of analysis is referred to as quasi-static. The wave loading is reacted by rigid body accelerations of the whole ship structure.
The hull girder stick model is used to determine the distribution of still water, wave and slam bending moments and shear forces.

The structural properties of the hull girder are calculated allowing for shear lag effects. The beam element represents the hull stiffness and runs along the centroid of area of the effective hull cross section.

The correct position of the centre of mass and the correct roll inertia are obtained by offsetting the mass on rigid stalks as shown in the detail.

TOTAL MASS = 2m/2 = m
TOTAL ROLL INERTIA = m/2I = MI

Figure 4.5
Hull girder stick model
Figure 4.6
Shell model of a tanker stern
The balancing effect of the accelerations is therefore modeled as a set of forces, of size $-m\ddot{x}$, applied to the structural model ($m$ is the mass at some point and $\ddot{x}$ the acceleration of that point). Many structural analysis systems allow the user to specify the mass distribution and the accelerations so that the balancing forces are then calculated automatically. Care is required to determine whether the structural analysis program calculates the reacting force: $-m\ddot{x}$ or the opposite force given by $+m\ddot{x}$.

Although the model is analyzed in quasi-static equilibrium it is still necessary to apply a set of supports to the model. This support system should be the minimum required to restrain rigid body displacement and rotation of the ship. Because the resultant force and moment of the loading (including equilibrating $-m\ddot{x}$ forces) is zero there should not be any forces acting on the supports; however the support system is necessary in order to define a datum for the calculation of deflections. In case the loading is not perfectly in equilibrium the reactions should be chosen to take out the forces on strong parts of the structure and a wide spacing will reduce the forces associated with any residual moment in-balance.

A minor difficulty is that the forces calculated by the diffraction analysis are in an axis system which is effectively fixed to the ground and if the ship rolls, pitches or yaws the forces remain in the global system. However the quasi-static analysis has the rigid body rotations removed and so the axis system is not quite the same: see Figure 4.7. This effect becomes increasingly important as the roll, pitch or yaw angle increases and the differences, including the effects of the accelerating axes, must then be taken into account in the analysis.

![Axis system for hydrodynamic analysis](image1)

![Axis system for quasi-static analysis](image2)

Figure 4.7
Fixed and moving axis systems
4.2.7 Hydro-structural dynamic analysis (for springing)

As discussed in 4.2.5, we can distinguish between two fundamentally different types of dynamic response of a ship:

a) The rigid body dynamics of the whole ship oscillating in the water.

b) The oscillation of the hull as vibrating beam.

The total response is the sum of (a) and (b). So far we have only been concerned with the former but, in some instances the hull girder dynamics must be taken into account.

Two of the most important mechanisms of excitation of commercial ships are slamming (described in Section 4.3) and springing. Springing occurs at forward speed into head seas when the encounter period of the waves equals the natural hull girder period. In this circumstance a wave of say 5 seconds may be encountered every 2 seconds by the ship.

For a ship with no forward speed a 2 second wave (of wavelength $1.56T^2 = 6.24m$) causes neither:

a) sufficient in phase loading along the length of the ship;

b) sufficient pressure on the bottom;

to produce a significant dynamic response.

However, a 5 second wave: (length = 39m) encountered at 2 seconds can produce a significant response.

The analysis has to be based on a proper dynamic model of the hull girder including the hydrodynamic effects of the water that are induced by the hull vibration. This can in principle be performed in several ways. A common and reasonably simple method is to treat the hydrodynamic forces caused by the hull vibration as added stiffness, mass and damping. Care has to be taken because the structural and rigid body motions are also dependent on hydrodynamic added stiffness, mass and damping. However, the values of the latter two are generally different for the dynamic structural response and the dynamic rigid body hull response. It may therefore convenient to separate the calculation of the rigid body and hull girder responses. This is also better numerically because the large rigid body deflections reduce the accuracy of the calculation of the small relative deflections which determine hull girder stresses. In practice this is not a problem with coarse beam type models but could become a problem with fine detail shell models.

Typical results, presented in the form of a transfer function, of midship bending moment, in regular waves with and without springing are shown in Figure 4.8. When combined with a sea state spectrum the result can be considerable additional excitation of the hull girder in relatively short seas, as shown in Figure 4.9. The importance of the resonant peak in the transfer function is dependent on the amount of exciting energy in the sea that is encountered at that frequency. Stresses in extreme seastates are dominated by long period effects. Excitation at resonant frequencies occurs in less extreme sea states and therefore makes more of a contribution to fatigue than the total extreme hull girder bending response. The resonant dynamic response is controlled by structural and hydrodynamic damping. Damping is always difficult to quantify and results from full scale measurements are the most reliable way of estimating the amount to use in the analysis.
Figure 4.8
Transfer function of hull bending moment per unit wave height with and without springing.
Figure 4.9
Hull girder response in various zero crossing period seas with and without springing.
4.3 SLAMMING

4.3.1 Impact slamming
Impact slamming occurs when the forefoot (the forward part of the ship's bottom), or an appendage (such as a mooring arm) emerges from the water and then re-impacts with the water surface. This can lead to local damage to the forefoot but, possibly of greater importance, generates a large dynamic oscillation of the hull girder in its first few modes.

The risk of slamming is greatest when:

- The wave length is ≈ ship length;
- When the draught is small (say less than 10m);
- At forward speed.

A simple calculation suggests that the pressure on re-impact should have the following form:

$$ P = \frac{1}{2} C_p \rho v^2 $$

where

- $C_p$ is a non-dimensional coefficient
- $\rho$ is the mass density of sea water (1025 kg/m$^3$)
- $v$ is the impact speed

Sometimes the equation is written as:

$$ P = kv^2 $$

clearly: $C_p = 2k/\rho$

where $\rho$ must be in the same units as $k$.

In practice $C_p$ is found to be dependent on several factors:

a) the angle between the water surface and the bottom: the slam angle;

b) the condition of the water surface.

There appears to be a minimum slam velocity below which no slam occurs. This is perhaps associated with a wave, generated by the ship bottom, filling up the air space under the bow and so preventing a slam occurring.

Values of $C_p$ have been determined theoretically in various model and some full scale tests. Some results are shown in Figure 4.10. The value of $C_p$ is highly dependent on the precise conditions of angles and water surface shape as the impact occurs. Full scale measurements, with the results handled in a statistical way, is the probable way forward for analysis. As a slam occurs the transverse line over which the force is applied moves towards the bow and the slam velocity and angle constantly changes. The analysis is therefore most conveniently undertaken in the time domain. Often it is reasonable to assume that the slamming forces do not effect the rigid body response of the ship. The slamming response can then be calculated as a post-processing operation. In some cases, particularly small fast craft, this is not possible and the slamming effect on the rigid body response has to be taken into account.

In addition to the peak value of pressure during a slam it is necessary to have a model of the rise and decay of pressure at any location. Again tests can provide this information. Some typical results are shown in Figure 4.11.
4.3.2 Flare slamming

Flare or momentum slamming is physically similar to bottom slamming but it occurs on more steeply sloping surfaces, such as the flared bow of a ship. The local pressures are less than for bottom slamming but the pressures are applied for longer periods over larger areas and so the resulting impulse applied to the hull, and the overall hull girder dynamic response can be similar. Tankers often have very little bow flare and therefore flare slamming is usually of less importance than bottom slamming for these types of vessel. There would seem to be a possibility of flare slamming at the stern but relative ship-water motion are often less there and as far as we are aware this has not been investigated.

Figure 4.10
Bottom slamming coefficients (Rask 1986)
\[ F(x)_t = F(x)_0 \cdot 80t \cdot \exp(1 - 80t) \]

where \[ F(x)_0 = \rho k y^2 \]

\( \rho \) = water density

\( k \) = slam coefficient (= 50)

\( V \) = entry velocity

**Figure 4.11**

Slam pressure time history

4.3.3 Analysis of structural response

The slam time history in terms of relative position and velocity of the ship and water surface may be calculated from either a frequency domain linear analysis or a time history analysis with some element of non-linearity as described in Section 4.2.3. Owing to slamming being primarily of interest in the larger waves, where the other non-linear effects are also becoming greater it is best to use the non-linear time history method.

An overall time-history for the hull slamming force can be calculated using the time history of the forefoot wetting, the pressure coefficient equation and the experimentally determined shape of the local pressure time history. This can be applied to the structural model in order to obtain the slamming response. Some typical midship results are shown in Figure 4.12. Note that the first mode response is most important at midships. The second mode is most important at the quarter points. The amplitude of the second mode response deflection is less than that of the first mode but the higher curvature: deflection ratio in the second mode make the moments as important.
4.4 WAVE DRIFT FORCES

In waves of very small height and a given frequency all loading effects are linearly proportional to the wave height and the sinusoidal nature of the waves means that:

a) There is no time averaged or mean force from the waves.

b) The ship response to each incident wave frequency is independent of the response to all other wave frequencies and the total response can therefore be calculated using the superposition assumption of linear spectral analysis.

In finite height waves the waves become non sinusoidal and the intermittent pressure loading around the water line is clearly non sinusoidal. The result is that in addition to the linear zero mean response at the wave frequency there is also:

a) a mean force (described in Section 4.4.1)

b) fluctuating forces at frequencies other than the wave frequencies (described in Section 4.4.2).
4.4.1 Mean drift force
The mean drift force is mainly associated with wave reflection and can be calculated on
energy principles for the special case of pure reflection of an incident wave by a wall:

\[ F = \frac{1}{8} \rho \, g \left[ 1 + \frac{2kd}{\sinh(2kd)} \right] H^2 \]

where
\[ k = \frac{2\pi}{L} \]
\[ L = \text{wave length} \]
\[ d = \text{water depth} \]
\[ H = \text{wave height} \]
\[ \rho = \text{water mass density} \]

In practice for a ship the wave is only partially reflected and so the steady force is lower
than given above. The relationship between the incident, reflected and transmitted wave
height and the steady drift force is then:

\[ F = \frac{1}{16} \rho \, g \left[ 1 + \frac{2kd}{\sinh(2kd)} \right] (H_i^2 + H_r^2 - H_t^2) \]

This relationship does not immediately help calculation of the drift force because the
reflected wave height is not known. However \( H_i, H_r \) and \( H_t \) are implied by the results of
linear diffraction theory. Therefore, the usual method of calculation is to use a linear
diffraction analysis in conjunction with certain relationships that conveniently give the 2nd
order surge, sway and yaw force. Note that the force is proportional to the square of the
wave height so that, as discussed above, in very small waves the steady drift force is
negligible.

4.4.2 Non-wave frequency excitation and slowly varying drift force

Regular waves
In regular waves the time varying second order force (ie that part which is proportional
to wave height squared and fluctuates about the mean value described in Section 4.4.1) is
small in comparison with the first order or linear wave force. The frequency of the force
is primarily at the wave frequency but it will also contain high frequency but low
amplitude components at for instance twice the wave frequency. Because of their low
amplitude these high frequency components are only important if there are any natural
frequencies in their region. For a free, conventionally moored or single point moored ship
the rigid body frequencies are significantly lower than the wave frequencies so the higher
frequency components cannot excite them. (Note however, for TLPs the heave, pitch and
roll frequencies may be excited by the high frequency harmonics).

Irregular waves
In irregular waves the sea can be considered to be composed of a range of frequencies.
The non-linear interaction of the second order force between any two of the different
frequency components result in a force having sinusoidal components at the sum and
difference frequencies of the two interacting component waves. This can be seen from the
mathematical identity associated with the product of two sine waves:

\[ a \cos(\alpha) b \cos(\beta) = ab \left[ \sin(\alpha+\beta) + \cos(\alpha-\beta) \right] \]

The frequency difference term results in a low frequency force and some pairs of
frequencies of waves in a seastate will have a frequency difference equal to the natural
surge period of the moored structure. The natural surge period will effectively tune into
and respond to this available forcing and the typically low surge damping will result in a large response (see Section 4.6.1). Indeed as the wave height increases the response becomes much greater than the first order linear response. Therefore the second order slow drift response is of primary importance in the design of the mooring system and the hull region in the vicinity of the mooring. A typical combined first and second order response is shown in Figure 4.13. The long period is the slow drift response, the short period is the wave frequency response.

Vessel motions and resultant mooring forces under wind and irregular wave loading as predicted by AQWA-DRIFT

Figure 4.13
Slow drift and wave frequency response of a moored tanker

4.5 WIND FORCES

4.5.1 Nature of the wind
Wind forces and dynamic response are complicated because:

a) The wind velocity averaged over some short time period increases as the time period reduces. So a 3 second gust has a higher wind speed than a 15 second gust. It is not immediately obvious which is appropriate for a given design.
b) The wind structure is such that over a given structure the wind speed will not be uniform, even before the structure disturbs the wind.
c) The structure will respond dynamically to the fluctuating wind so a dynamic amplification effect will need to be taken into account.

4.5.2 Traditional approach
Traditionally the approach has been to use some combination of gust period and dynamic amplification factor that is based on experience. A mooring system might therefore be designed for a 15 second gust and no dynamic amplification.
However it is a very approximate design method that takes little account of the precise
details of the given system. A more logical analysis method is given below.

4.5.3 Mean wind force
The wind force is conveniently split into a steady and dynamic component. Again it is not
immediately obvious how to define the steady wind speed because it depends, as discussed
for gusts above, on the time period over which the mean is taken. However analysis of
wind data shows that using an hourly mean is a reasonable basis for further analysis. This
is because it falls between the time taken for the weather system to pass (10 - 1000 hours)
and the periods of turbulence (0.1 to 0.001 hours).

4.5.4 Gust response spectrum
The fluctuation from this hourly mean can now be regarded as turbulence carried along
by the mean wind speed. The turbulence itself can be analyzed using spectral methods to
define both a single point turbulence spectrum and cross-spectra between all pairs of points
and directions within the wind. The dynamic response to the wind gusts can be obtained
by combining the turbulence, single point and cross spectra with the dynamic response of
the structure to sinusoidal loading. This is a straightforward procedure, although it is
computer intensive. This is quite practical and is a method used for analysing the response
of towers to wind turbulence. However for floating structures the long natural periods
likely to be excited by the wind allow a simplification of the problem. The simplification
is based on the knowledge that long period gusts are of a large size so that they envelop
the complete structure. This allows the double summation to be replaced by a single
summation in the region of the natural frequency of the structure. The dynamic response
of the structure can then be analysed using spectral methods which are very similar to
those used for the calculation of the response to waves.

4.6 DAMPING

4.6.1 Slow drifting damping
An assessment of slow drift damping is important as it governs the amplitude of the slow
drift response of a moored tanker to waves and wind turbulence.

The slow drift damping which limits the slow drift oscillations has components due to:

- wind damping
- current damping
- wave making damping
- skin friction on hull in still water
- additional damping in waves
- damping due to effect of mooring

The predominant wind and current damping takes the form of the linear damping which
is proportional to the mean wind and mean current speeds.

Since the slow drift oscillations have long periods, typically 200 seconds, the damping
caused by radiated waves is relatively small.

Model tests show that the damping in waves is much greater than in still water. This effect
has been traced to a variation in the wave drift forces with forward speed. It has been
shown that as the vessel oscillates, the drift forces on the hull increase with velocity in a
way that acts to dampen the slow drift oscillations. Much work has concentrated on
determining this analytically using diffraction methods. (eg. Newcastle University;
Hearn, Tong and Lau 1987)

For some types of structures the damping forces on the mooring form a considerable part
of the total damping, eg. drag forces on catenaries.

4.6.2 Springing/slamming damping
Damping is important both for slamming and springing. It determines the sizes of the
decaying cycles following a slam and the amount of resonant springing response. In both
cases the primary effect is on fatigue rather than extreme loading. The damping of the
structural modes is practically impossible to assess theoretically. Instead it is necessary to
rely on full scale measurements. These suggest that the damping of the first mode of a
typical tanker is about 0.5% - 2% critical.

4.7 STRENGTH ANALYSIS OF HULL GIRDER

4.7.1 Extreme load calculation
The extreme loading on the hull girder has to take into account:

- First order wave forces resulting in bending, shear, torsion etc, (Section 4.2).
- First and second order mooring forces, (Section 4.2, 4.4 and 4.5).
- Slamming (Section 4.3) and springing (Section 4.2).

It is possible to calculate (or at least to approximate) all the responses in time history
simulations of the total behaviour. However the results of this type of all embracing
analysis allow only a limited understanding of the underlying behaviour and, as discussed
earlier, it may therefore be considered preferable to analyze the separate responses and to
add them together for the combined effect. The problem then is to decide how the various
values of individual extremes should be combined. It can be argued that if the effects are
uncorrelated a square root sum of squares addition of the separate rms values might be
appropriate. In practice the various effects are driven by the same waves and so some
underlying correlation may affect the results. This problem is presently under investigation
at Southampton University as part of the Bofcos program.

4.7.2 Strength of hull girder
The strength of the hull girder is usually based on an elastic calculation and a factor of
safety against first yield. However, failure will occur not when the yield stress is reached
but when either:

a) the plastic capacity of the cross section is reached in bending;
b) the plastic capacity of the cross section is reached in shear;
c) a brittle fracture originates from a defect or fatigue crack;
d) buckling of plates or stiffeners occurs to the extent that the overall strength
   reduces.

or:
a) The plastic behaviour implies a small factor of safety on the first yield criterion.
b) Shear yield is also checked by beam type calculations.
c) Brittle fracture is not usually directly calculated but is assumed to be avoided by
   a combination of material selection and inspection. This is an area of interest now
   that tankers are commonly being used for long periods in harsh environments
   such as the North Sea where fatigue cracking is much more of a problem. There
   is some question as to whether the materials acceptable to the classification
   societies are in fact sufficiently tough to be able to withstand likely fatigue
   cracking with sufficient safety against brittle fracture.

Design for fatigue and the corresponding inspection requirements will have to be
more carefully thought out to keep the risk of fracture to an acceptable level. A
high risk period may also follow repair eg. to replace corroded plating. The
welding in of new material will induce residual tensile stresses. If these affect a
nearby fatigue crack a brittle fracture may result.

This is discussed in greater detail in Section 7.

d) Buckling is taken into account by direct calculation or by rule sizing requirements which aim to ensure that the yield strength can be achieved prior to buckling. If the additional benefit of a fully plastic section strength is required then the buckling requirement must allow for extensive strain beyond the yield strain, as the plasticity extends towards the neutral axis.

4.7.3 Extreme stresses
See Section 6.3.3 for a discussion of likely extreme stresses in the hull structure.

4.8 FATIGUE ANALYSIS OF HULL GIRDER

Fatigue crack growth in a ship's hull is often caused partly by the global hull girder flexure and partly by the local effect of external pressure fluctuation and internal sloshing in tanks. Filling and emptying of tanks may also be an important fatigue mechanism for F(P)SU's. The global effects are considered in this section and the local effects in Section 5. However to make a fatigue assessment the non-linear interaction of the two types of loading needs to be considered.

4.8.1 Details subject to fatigue from overall bending
The details potentially subject to fatigue from hull girder bending are welding defects and discontinuous or stress concentrating details in the longitudinal structure. Possible sites of stress concentrations include cut-outs for pipes to pass through decks, hatches, attachments welded to the decks, connections between transverse and longitudinal frames and stiffeners. Large "rogue" welding defects in butt welds may also cause unexpected fatigue cracks.

4.8.2 Phasing of loading effects
A small additional stress range can significantly reduce the fatigue life, this is reflected by fatigue damage being considered proportional to the cubic power of the stress range. For this reason it is important to allow for effects such as bending in the horizontal plane, shear and local pressure loads. The relative phasing of the loads has to be accounted for in the analysis and this may be done using complex number techniques.

4.8.3 Deterministic fatigue analysis
In order to assess the amount of damage caused by fatigue it is necessary to estimate the number of cycles of stress ranges of various sizes. The simplest method to use is the deterministic method. However, the label deterministic fatigue analysis is applied to many different types of fatigue analysis. Commonly the methods involve large approximations but are quite simple to apply and are useful for that reason. Simple deterministic methods are in general not satisfactory for springing but can be sensibly applied to quasi-static wave frequency response and slamming.

One of the simplest deterministic methods is based on an extreme (say 20 year) dynamic response. This, in conjunction with an assumed number (say $5 \times 10^4$ waves per year) is used to define a log-linear exceedence diagram of stresses. Having thus obtained an estimate of the number of cycles of the various stress ranges, a fatigue calculation can then be performed.

4.8.4 Semi-probabilistic fatigue analysis
A more sophisticated deterministic method breaks up each sea state into a number of constituent waves of various heights and periods. These are summed for all sea states and the response is assumed to be characterized by the response to the constituent waves as
individual waves. This method was proposed for offshore structures by Holmes et al. (1978) which named it the semi-probabilistic method. It is reasonable when the springing response is negligible. The method generally seems to give conservative but more reliable results than the simple deterministic methods.

4.8.5 Random sea time history analysis

Other deterministic methods pay less attention to the wave period and these are less reliable than the semi-probabilistic method.

A deterministic method which does not involve the large approximations is the time history analysis in irregular waves. This is however computationally very demanding and is not usually used in practice at present for tanker structures.

The method has the advantage that the interactions between effects such as slamming and wave frequency response can be automatically included. However, the method does not necessarily lead to an understanding of the behaviour that is modelled.

4.8.6 Spectral fatigue analysis

Spectral (frequency domain) fatigue analysis is very useful for the analysis of the fatigue damage from the wave frequency loading and it handles a number of situations which are impractical to analyse with simple deterministic/semi-probabilistic methods:

- The springing response of a structure which is responding at its natural frequency to some part of the spectrum of exciting forces;
- It can be used to predict the number of slam events of various severities;
- Spectral analysis is also an integral part of the drift response calculation.
- Combinations of seas and swells and directional spreading effects are also best handled using spectral analysis.
5. THE LOCAL EFFECT OF LOADING

5.1 INTRODUCTION

5.1.1 Loading mechanisms
The loading on the side shell plating is associated with:

a) External pressure loading:
   - The incident wave pressure fields;
   - The changes in them caused by diffraction effects (which for beam seas may double the pressure ranges associated with the smaller waves which reflect from the side of the ship);
   - The ship motions, primarily pitch roll and heave, in the essentially hydrostatic pressure field and the ‘radiation’ pressures caused by waves generated by these motions;
   - Impact slam forces on the bottom as it cuts the water surface;
   - Flare slam forces on the bow as it submerges;
   - Slap forces from the crests of steep waves;

b) Acceleration reaction forces from ship and cargo mass;
c) Cargo sloshing;
d) Global bending, shear, torsion etc;
e) Welding residual stress.

5.1.2 Typical loading patterns
The overall pressure loading from those items identified in (1) above is dependent on wave height, wave period, ship speed and direction relative to the waves. An example “fatigue averaged” pressure distribution is shown in Figures 5.1 (Note: these results will be route specific). The wave forward side is subject to higher pressure loads and this is particularly pronounced on the bow in bow seas. In short waves the majority of the wave energy is reflected so the wave pressure on one side of the hull is considerably increased whilst the other is decreased. In longer waves less reflection occurs as the wave can pass under the ship. Further increases in wavelength result in the ship riding the waves with very little wave induced cyclic pressure.

Cargo sloshing may be more important for an F(P)SU than a trading tanker because certain tanks will commonly operate at variable level as the F(P)SU is filled. The analysis of cargo sloshing requires a solution of the Navier Stokes equations because viscous and non-linear effects are important. An indication of potential problem areas can be obtained from hand calculations or diffractions analysis programs.
Figure 5.1
Example of external pressure loading on side shell

Notes:

a) Contours show relative pressure ranges averaged for fatigue calculations for a particular route and tanker.
b) Actual values will be dependent on route, tanker and speed.
c) Port and starboard values will be different, often significantly different, because of effect of sea state directionality.
d) Ballast draught values will have a similar pattern but displaced downwards.
e) This diagram requires second order effects associated with finite height waves to be taken into account. It cannot be determined from linear diffraction analysis alone.

5.2 OUTER SHELL CONSTRUCTION

5.2.1 Typical connections
The outer shell, in the tank region, is typically longitudinally stiffened between transverse bulkheads and frames. The longitudinals on the shell are usually fabricated angles, although sometimes bulb flats are used. The stiffeners on the bottom are usually tees. The connections between the longitudinals and the frames and bulkheads are where the fatigue damage is most common. These connections are a compromise between ease of fabrication and fatigue durability and arguably their design should be influenced by the expected use of the tanker. Ideally a tanker used for long periods in a fatigue hostile environment such as the North Sea should have better quality details than a tanker intended to be used primarily in the Indian Ocean. Unfortunately the design rules do not yet take this into account and the shipyards have no reason to change from their cheaper details because they are only responsible for defects which occur in the first year.

5.2.2 Historical data
There has been a gradual evolution of connection details as Classification Societies have responded to fatigue problems. Figure 5.2 shows a typical detail and the cracks that have caused problems in the past. Figure 5.3 shows the common present arrangement. This is better but similar locations are still prone to fatigue cracking. Figure 5.4 shows further, but expensive improvements which may be used to modify existing problem ships or to obtain a fatigue resistant new build. These are discussed further in Section 5.5. Highway bridge decks have suffered similar problems. The usual approach to improving the bridge check connections has been to avoid welding to the flange of the longitudinal stiffener.
Figure 5.2
Old design of side shell connection detail showing typical cracks

Figure 5.3
Common present design or repair of side shell connection

Figure 5.4
Improved but expensive connections design, shown for bottom shell
Note soft toes on bracket, flat bar stiffener and frame web cut out
5.3 STRENGTH ANALYSIS OF CONNECTIONS

Design for ultimate strength is much easier than design for fatigue. This is because, providing the structure performs in a ductile manner, ie without buckling or fracturing, the ultimate strength is determined by the overall capacity of the steel to resist the applied loads and not by localized stress concentrations. Therefore a detailed stress analysis is not required for strength calculations. All that is necessary is to check that there is sufficient material to resist the applied loading.

5.3.1 Strength requirement with depth
The requirement for strength of side shell and connections with depth is determined by the difference between internal and external pressures. In large waves the external pressure will increase approximately linearly with depth and the requirement is therefore determined on the basis of the maximum likely crest position of a wave relative to the hull.

5.3.2 Strength assessment
The rules require a design pressure equivalent to a head of 0.9m above the deck level. On the basis of this the stiffeners can be designed for the encastre moment of:

\[ p.a.b^2/12 \]

where:
- \( p \) is the pressure
- \( a \) is the spacing of longitudinals
- \( b \) is the length of the longitudinals between supports

The connections themselves can be designed to transfer the force of \( p.a.b \) and providing the sum of the capacities of the various load paths is satisfactory the strength can be considered sufficient.

5.4 FATIGUE ANALYSIS OF CONNECTIONS

Fatigue behaviour is very dependent on the local stresses, details and the crack-like defects built into the structure, eg. by welding. Fatigue analysis therefore has to account for both the local stress concentrations and the likely defects at that location. This is achieved either:

- with stress analysis in conjunction with experimental results for similar details: the S-N approach; or
- by using fracture mechanics which may involve a detailed stress analysis with defects modelled in the analysis.

It should be noted that the commonly used C to W S-N classes can give misleadingly optimistic results when applied to ship details.

5.4.1 Fatigue strength requirement with depth and position
Fatigue damage is approximately proportional to the number of cycles times their stress range cubed. For any particular sea conditions it is therefore possible to calculate a cubic weighted mean stress response such that if all the stress ranges were of constant amplitude they would cause the same amount of damage as the actual random stresses. For the side shell the local stresses are proportional to the pressure so an interesting property of the pressure loading is a contour diagram of cubic weighted mean pressure on the shell. This demonstrates the "fatigueness" of the applied loading and how the strength of the shell needs to be distributed. Figure 5.1 was obtained in this way. The diagram will vary from one offshore location to another and is dependent on the typical heading of the ship to the waves. However the overall characteristic will remain the same. A diagram, comparing the extreme and average fatigue pressure at one cross section, for a reasonably severe offshore environment, is shown in Figure 5.5. Note that this does not indicate the relative
importance of strength and fatigue, which is discussed in Section 5.5.

5.4.2 Effect of tank loading (dynamic and static effects)
A cargo in a tank has three effects.

a) If the tank is only partially full the cargo may slosh from side to side of the tank and induce additional fluctuating and possibly slam pressures on the inside of the tank and stiffeners.

b) The inertia of the cargo results in a stress range on the shell as the cargo accelerates and causes a pressure gradient in the cargo.

c) The static cargo pressure changes the static stress acting in the connections and this changes crack growth rates. This is difficult to take into account because welding induces very large tensile residual stresses in the welds. The S-N curves used for welded steel assume the worst tensile mean stress compatible with the stress range. This is a pessimistic assumption and in reality an external loading pattern that results in a mean compression will have a better fatigue life than one which results in a mean tension.

The external pressure acting on the shell of an empty tank results in beneficial compressions which may extend the fatigue life of ballast tank longitudinal-transverse connections near the loaded water line and cargo tank connections near the ballast draught water line.

5.4.3 Interaction with transverse frame deformation
The side shell longitudinals span from transverse frame to the next frame or bulkhead. The bulkheads are very stiff whereas the frames can deflect considerably under the effect of shell pressure loading. For strength calculation it matters little what the support is since the frame deflection moments induced in the stiffeners can be plastically redistributed. However for fatigue calculation these deflection-induced stresses are as important as any primary stress and should be taken into account. An example of the change in bending moments in the longitudinal and the effect on the forces transferred at the connections is given in Figure 5.6. The precise effect will depend on the construction of the ship. Those without horizontal struts in the transverse frames may be more susceptible to this deflection effect than those with struts. However the strutting action is not perfect because the struts are usually only placed in the wing tanks and are not triangulated. They therefore make both the inner and outer parts of the transverse frame deflect together, so perhaps doubling the stiffness but do not cause the order of magnitude change that could be beneficial.
5.4.4 Interaction with hull girder bending and shear
The stiffeners will also be subject to axial stressing from hull girder bending and axial forces. The shell plating will also be stressed by hull girder shear and torsion. The fatigue calculations should take the combined effect into account. The highest bending moments occur around the midship region. The highest shears are at 25% and 75% along the length. The combined effect is to give high stresses along a large part of the length of the ship (see Figure 5.7).

5.4.5 Fatigue strength assessment of typical notched details
Fatigue strength assessment is complicated for many ship details because they contain right angled notches. The theoretical stress concentration factor for these notches is infinity (∞). An attempt to perform a conventional finite element analysis can result in an answer which simply approaches ∞ as the mesh is refined. Clearly this is of little use for fatigue analysis purposes. There are two methods of handling this problem:

a) Test results; results from a similar detail which is also subject to finite element analysis with the same coarseness of mesh as the detail of interest. The results of the tests are extrapolated to the detail of interest by using the ratio the results of the two analyses. This method is of doubtful accuracy because the tests of sufficient similarity will not usually be available and the accuracy of the ratio will
always be in doubt.

b) Fracture mechanics methods: A small crack can be modeled at the notch and the
fracture mechanics stress intensity factor calculated. This can be used for the
direct calculation of crack growth rates. For a given initial defect size a fatigue
crack can be calculated and the results are also useful as a guide to understanding
the preferred crack propagation directions and for inspection planning, (see
Section 7).

5.4.6 Crack growth rate calculation (including residual stresses)
The crack growth equation is known as Paris' Law:

da = C.ΔK*dn

where: C and m are constants which define the crack growth rate
da/dn is the crack growth per cycle
ΔK is the cyclic range of stress intensity factor K

In certain simple geometries:

K = YΔσ \sqrt{πa}

where: Δσ is the applied stress range at some distance from the crack
a is the crack length (or half length)
Y is a factor which allows for the geometry of the specimen and
the crack

In many of the geometries of interest there is no obvious definition of Δσ and no tabulated
Y values but the fracture mechanics method in conjunction with the finite element method
calculates ΔK directly and so avoids the problem.

Static and residual stresses complicate the fracture mechanics analysis in a similar way to
the S-N calculation. They can be taken into account in the parameters A and m or in the
proportion of the stress range that is assumed to act on the opened crack. (Note this is not
necessarily the tensile part of the applied plus welding residual stress range since plastic
effects at the crack tip induce an additional local pattern of residual stress.)

Typical mean values of C and m, in units of MPa and m, are:

C = 5.2 \times 10^{-12} \, , \, m = 3 \, (Maddock, 1974)
C = 7.1 \times 10^{-12} \, , \, m = 3 \, (Austin, 1978)

5.5 SELECTION OF CONNECTION DETAILS

5.5.1 Typical details
Figures 5.2 and 5.3 showed some typical details used for longitudinal stiffener-transverse
frame connections. There is a tendency for these details to suffer from fatigue cracking
as was discussed in Section 5.2.2. The critical position at which cracks grow in any detail
is dependent on the loading and the geometry of the detail. It should be noticed that some
crack directions are inherently preferable to others. In particular cracks growing through
the longitudinal stiffener to the shell may result in pollution or a catastrophic hull fracture
whereas cracks growing through the frame stiffener are, for some time at least, not too
much of a problem either for pollution or overall vessel safety.

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5.5.2 Primary member sizing

a) Longitudinal stiffeners

The longitudinal stiffeners need to be sized for the cyclic pressure and hull girder forces. As a rough guide the cubic weighted mean pressure range profile down the side of a tanker is as shown in Figure 5.5. It is possible to design for an F2 weld classification but for many designs the worse G classification is more appropriate. In 20 years there are about 108 stress cycles. The acceptable stress range for 108 cycles of a G detail is 13.5 N/mm². To compare this with a conventional design requirement we first convert to an effective extreme pressure that would also satisfy the fatigue design requirement:

For steel the extreme stress is usually limited to 0.6fy so the extreme pressure that if resulting in 0.6fy would satisfy the fatigue requirement is simply the profile shown in Figure 5.5 × (0.6fy/13.5).

For comparison this and a typical classification society rule requirement are shown in Figure 5.8. This clearly demonstrates the lack of stiffener section modulus that may result from simply using the extreme-load-based class rules.

This calculation does not take into account:

- Stress concentrations effects that may make certain parts of the joints behave in a worse manner than a G class detail.
- Initial stiffener lack of straightness effects which may be more severe for the extreme loading case than for fatigue. Often these effects are ignored completely for fatigue. This is partly justifiable because the buckling mode for extreme load design will probably have an effective length equal to the stiffener length, whereas the buckling mode that magnifies the fatiguing stresses will have 0.5-0.7 the effective length and 4-2 times the buckling load. However the straightness imperfections will still cause an increase in stress range. We have found cases of fatigue damage resulting from this effect.
- Overall hull girder bending which will often considerably expand the area of shell plating which is dominated by fatigue.

Fatigue cracking through a longitudinal stiffener is a severe problem because it can lead to brittle fracture or pollution.

![Diagram of fatigue and strength](image)

**Figure 5.8**
Relative importance of fatigue and strength for longitudinal stiffener section modulus from local bending effects

*Note: High stress concentrations in certain locations of the longitudinal stiffener to transverse frame connection can, in these locations, make fatigue dominant over the wetted area of the hull.*
b) Shell plate
   A similar calculation can be performed for the shell plate. The S-N classification would
   normally be F but could be worse, depending on the cut-out around the side shell -
   transverse frame connection or the presence of an attachment. Double lugs connecting the
   longitudinal stiffener probably reduce the probability of shell plate cracking at the cut-out.
   Avoiding too wide a cut-out and providing soft toes (as shown in Figure 5.4) will also be
   beneficial.

c) Transverse frame stiffener
   In the majority of connection designs this is a weak point. As discussed earlier bridge
   designers have responded to this problem by not connecting the transverse frame stiffener
   to the longitudinal. This has been tried on some tankers but there is some concern that the
   resulting structure may be less tolerant of berthing forces or minor collisions.

   At first sight the stresses in the frame stiffener (see Figure 5.3) should be quite easy to
   estimate and the average stress is typically approximately:

   \[ \sigma = \frac{P \cdot a \cdot b/A}{2} \]

   where
   P is the pressure on the side shell
   a is the spacing of longitudinal stiffeners
   b is the length of longitudinal stiffeners between supporting frames
   A is the area of the frame stiffener

   The division by two is to allow for approximately half the load being carried out of the
   longitudinal stiffener by the frame stiffener and half by the lugs which typically connect
   the longitudinal stiffener web to the frame web. (A more accurate estimate can be made).

   Unfortunately this simple calculation does not allow for the high SCF which occurs in the
   connection. The SCF seems to be caused by several factors including:

   - The right angled corner between the frame and longitudinal stiffeners.
   - The eccentric support of the frame stiffener by the frame, which results in the
     frame stiffener bending and redistributing the applied force.
   - Compatibility of strain in the longitudinal and transverse stiffeners.

   Whilst we have started to make some effort to write semi-empirical formulae to describe
   this behaviour, convincing and generally useful results are still some way off and in the
   meantime it is usually necessary to perform finite element/fracture mechanics analyses on
   the specific details of any ship. This is reasonably practical since although there are
   thousands of details in any ship there are only a limited number of types.

   Fortunately although fatigue problems in this area are a nuisance, and can eventually
   propagate into the shell this requires extensive crack growth and a reasonable inspection
   procedure can usually be expected to find these cracks before they become a safety
   problem.

5.5.3 Brackets
   To improve the fatigue behaviour it is common to add backing brackets and/or lugs (see
   Figure 5.3). These help both the longitudinal and frame stiffener. Brackets are beneficial
   because they reduce the span of the stiffener and they can be designed for a low SCF in
   conjunction with a lower average stress. However the most important effect is probably
   that they move the likely critical fatigue location from the mouse hole which is very
difficult to weld, to the outer toe of the bracket where it is much easier to obtain a
reasonable weld. Also as a retrofit to a damaged structure the addition of brackets may be
very beneficial since, if fitted to either:
a) the opposite side of the frame web to the stiffener; or
b) both sides of the frame web;

They reduce the eccentricity and hence the bending in the connection.

5.5.4 Lugs
Recent tankers have had lugs fitted during construction. Some earlier designs did not have lugs and suffered from fatigue damage in early trading. Lugs should significantly improve the fatigue life but even designs which are lugged on both sides of the longitudinal stiffener web and bracketed may, in a harsh environment, still be unsatisfactory from a fatigue point of view.

5.5.5 Options
New designs can be sized to avoid problems, however this requires more detailed analysis than has been common at the design stage. Using the analysis methods outlined above a satisfactory design should be achievable. In many cases the fatigue resistant design will be significantly more expensive than typical poorly performing existing details.

For example, in the column-pontoon nodes of a floating production platform it was calculated that the stress ranges in the areas of the connections were such that ordinary ship-type connections were unacceptable. After considerable discussion a relatively expensive detail was selected. This involved fully plating the frame web and using T stiffeners on the frames with continuity of flanges through the joints.

The continuity of flanges may be expected to considerably reduce the SCF at the critical location on the frame stiffener and the fully plated arrangement avoids the SCF associated with the cut-outs in the frame web. Although a G classification for the stiffener is obtained, because of the weld around the edge of the longitudinal stiffener flange, shear lag effects make this not quite so unbefeficial.

However for ordinary ship shape FSUs a cheaper arrangement should be satisfactory and it is likely, given more time and some of the analysis techniques developed since, that a cheaper detail could now be designed.

Improving troublesome details in existing ships is expensive because of the numbers involved. Additional lugs and brackets are often added and will usually give a significant additional life. The design of the additional steelwork is more difficult than for a new design where all the details can be planned to fit together. Even a relatively poor bracket tends to have an advantage because, as described above, the highest stress concentration is usually moved to an area of better weld quality. However, care is necessary since the new welding will induce residual stresses which may cause nearby undetected defects to fracture.
6. STEEL TYPE

In order to accept the safety factors, used for present day design, it is important that the steel should be sufficiently tough and ductile. Toughness implies that small cracks will not result in a failure before the design strains are reached. Sufficient ductility will allow redistribution of loading from any part which is over stressed from some combination of internal stresses (eg. welding residual, temperature) and external loading (eg. still water and wave). For ductile behaviour in the presence of fatigue cracks a high level of material toughness is necessary. The benefits of toughness and ductility are therefore:

a) A tolerance of crack like defects.
b) The capability to redistribute stresses and to absorb energy in plastic deformation, which adds significantly to the safety of the structure.
c) The structure is much more amenable to approximate analysis methods since a ductile structure will deform and if at all possible find some pattern of internal stresses which equilibrates the applied loads. In contrast a brittle structure may fracture and so damage other parts of the structure which in principle could otherwise carry the applied loads.
d) Slamming forces, which on occasion may result in moments which are very high in comparison with the ordinary wave bending moments, are more likely to be absorbed by plastic deformation, providing a fracture does not occur first.

6.1 Strength and toughness characteristics
The fracture resisting property of a material containing crack like defects is referred to as its toughness. In principle the material toughness and inspection should be selected after taking the following into account:

a) Avoidance of fracture initiation from a defect which is too small to be located by practical visual inspection.
b) Arrest of a fracture, from a defect in a poor low toughness weld, by the surrounding plate.

It would also be convenient to be able to:

c) Arrest a large running crack.

Toughness is dependent on the thickness of the steel, the temperature, the nature of the cracks and the loading rate. Ideally therefore tests to determine toughness should be conducted under conditions as close as possible to those likely to be found in practice. Unfortunately this requires full thickness specimens with sharp fatigue cracks and relatively slow loading rates.

Tests such as the crack tip opening displacement (CTOD) test have these characteristics. They give useful toughness properties in the form of Kc and CTOD values, but are expensive to perform.

The commonly used Charpy test uses a test piece which is usually thinner than the plate of interest, a blunt notch and a high loading rate, none of which are representative of the service conditions. However it is reasonably cheap and has served most industries well since its introduction as a quality control procedure.

Because the correlation of Charpy values with the toughness properties of real interest (eg CTOD and Kc values) is only partial, the data obtainable from the Charpy test is difficult to use directly in any calculations that try to determine acceptable defect sizes or inspection requirements. However various correlations between Charpy and Kc values are available. A particularly useful correlation is that produced by Sanz, see Section 6.3.2 below.
The specification for strength and toughness of Lloyds ship grade steels are summarized in Table 6.1:

<table>
<thead>
<tr>
<th>Grade</th>
<th>Yield Stress</th>
<th>Tensile Strength</th>
<th>Charpy Test Temperature</th>
<th>Energy</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>235</td>
<td>400-490</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>B</td>
<td>0</td>
<td>-10</td>
<td>27</td>
<td></td>
</tr>
<tr>
<td>D</td>
<td></td>
<td>-40</td>
<td></td>
<td></td>
</tr>
<tr>
<td>E</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>AH32</td>
<td>315</td>
<td>440-590</td>
<td>0</td>
<td>31</td>
</tr>
<tr>
<td>DH32</td>
<td></td>
<td>-20</td>
<td></td>
<td></td>
</tr>
<tr>
<td>EH32</td>
<td></td>
<td>-40</td>
<td></td>
<td></td>
</tr>
<tr>
<td>AH34S</td>
<td>340</td>
<td>460-610</td>
<td>0</td>
<td>34</td>
</tr>
<tr>
<td>DH34S</td>
<td></td>
<td>-20</td>
<td></td>
<td></td>
</tr>
<tr>
<td>EH34S</td>
<td></td>
<td>-40</td>
<td></td>
<td></td>
</tr>
<tr>
<td>AH36</td>
<td>355</td>
<td>490-620</td>
<td>0</td>
<td>34</td>
</tr>
<tr>
<td>DH36</td>
<td></td>
<td>-20</td>
<td></td>
<td></td>
</tr>
<tr>
<td>EH36</td>
<td></td>
<td>-40</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Note: See original document for additional notes.

6.2 RULE SELECTION OF STEEL FOR A GIVEN SERVICE

The selection of a steel for a given service is based on rule requirements: example requirements are shown in Figure 6.1.

These rules are based on experience and an assessment of the likely stresses and temperatures that different parts of a ship are likely to see. They also allow for greater Charpy toughness for thicker plating. This is because the Charpy specimens are always of constant thickness yet increasing plate thickness results in decreased toughness.
Figure 6.1
Steel grades required by Lloyds rules
example cases for various positions along a vessel

Note:
1) \( L = \text{ship length}, L > 250 \text{m} \)
2) beyond 0.6L A or AH steel is acceptable
3) Longitudinal stiffeners may be grade A or AH
4) based on Lloyds rules
   and regulations for the classification of ships
   1989, part 3. Table
   2.2.1.

6.3 CALCULATION OF CRITICAL DEFECT SIZE

6.3.1 PD6493 methods
PD6493 provides methods for assessing critical defect sizes in various stress fields. The latest issue (1991) contains two basic methods:

The first is the old PD6493 method which is useful for certain simple geometries.

The second is the CEGB R-6 approach which is particularly useful in conjunction with linear elastic fracture mechanics. This method can deal with more complicated geometries providing a linear elastic fracture mechanics \( K_c \) solution is available from a reference or from a suitable (eg. finite element) analysis. This technique is therefore very useful for handling the complicated details often found in ship construction.
6.3.2 Material toughness estimates
Because the specification of the toughness of ship steels is based on Charpy testing, it is difficult to perform fracture mechanics calculations for specific grades. However, Sanz (1981) has proposed a correlation between Charpy values and $K_c$ values. This can be used to estimate fracture toughness requirements for steel types for F(P)SUUs.

Preliminary calculations suggest that:

Estimated lower bound (±2 standard deviation values) $K_c$ for steels with the minimum Charpy requirements for the various ship steel grades are (based on Sanz 1981 except * which is based on Sumpter et al., 1988) are given in Table 6.2:

<table>
<thead>
<tr>
<th>Low loading rate:</th>
<th>High loading rate:</th>
</tr>
</thead>
<tbody>
<tr>
<td>initiation toughness</td>
<td>'arrest' toughness</td>
</tr>
<tr>
<td>Mild steels $\sigma_y = 235$ MPa</td>
<td></td>
</tr>
<tr>
<td>A 1500*</td>
<td>700*</td>
</tr>
<tr>
<td>B 3500</td>
<td>1600</td>
</tr>
<tr>
<td>D 4200</td>
<td>2000</td>
</tr>
<tr>
<td>E 6900</td>
<td>3000</td>
</tr>
<tr>
<td>High yield strength steel $\sigma_y = 355$ MPa</td>
<td></td>
</tr>
<tr>
<td>AH 4000</td>
<td>2000</td>
</tr>
<tr>
<td>DH 5900</td>
<td>2600</td>
</tr>
<tr>
<td>EH 7200</td>
<td>3900</td>
</tr>
</tbody>
</table>

Better estimates could be made from data on ship steels. Charpy results show considerable scatter and can be considerably above the minimum specified value which these $K_c$ values have been based upon.

6.3.3 Extreme hull girder stresses and critical defect sizes
A potentially dangerous time occurs after repair to an old ship because the high residual stresses occur at the same time as larger, fatigue grown, cracks. This combination, with commonly occurring wave conditions, may lead to a relatively high probability of a brittle fracture occurring. Therefore particular attention has to be taken to the estimation of the stresses.

The extreme (1 in 20 year) stresses in the dock and bottom of tankers are roughly estimated in Table 6.3 below. Clearly the location of the tanker and the manner of operation have a large effect on these values. These estimates should be checked against full scale measurements which some operators are now performing.
Table 6.3  
Estimated extreme stresses

<table>
<thead>
<tr>
<th>Residual compression in plates:</th>
<th>initial value</th>
<th>15% yield</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>shaken down values</td>
<td>10% yield</td>
</tr>
<tr>
<td>Welding residual tensile stress along welds:</td>
<td>initial value</td>
<td>100% yield</td>
</tr>
<tr>
<td></td>
<td>shaken down values</td>
<td>50% yield</td>
</tr>
<tr>
<td>Still water + wave bending:</td>
<td>20 year values</td>
<td>60% yield</td>
</tr>
<tr>
<td>Slam vibration at midships:</td>
<td>highly variable</td>
<td>50% yield</td>
</tr>
</tbody>
</table>

Notes:
Excessive slamming is in principle avoidable by maintaining adequate draught at the bow.

Slamming is generally much less serious for a stationed FSU than for a trading ship with forward speed.

Peak still water + wave + slam bending moments are unlikely to occur simultaneously.

From a CEGB R-6 method (see 6.3.1) fracture assessment viewpoint the slamming can be considered as a self equilibrating stress. This is because, if the structure is sufficiently ductile, slamming will simply cause a little plastic deformation. However if it does not have sufficient ductility slamming can cause a brittle fracture. The higher strain rates associated with slamming (eg. 5 x the wave bending strain rates at midships) are also disbeneficial for fracture.

Tensile residual stresses act over a band much wider than the weld itself. An estimate of the width can be made from research done to estimate residual compressive stresses for buckling calculations in the bridge code B5400. This has found that residual compressive stresses are about 15% of yield. For a plate of typical b/h of about 40 equilibrium requires a tensile stress band width x, where:

\[ 0.15\sigma_y (40 - x/h) t^2 = \sigma_y (x/h) t^2 \]

or:

\[ x/h = (0.15 \times 40)/(1 + 0.15) = 5.2 \]

for a typical thickness (t) of 20mm x = 100mm

ie. the band of residual stress has a width of about 100mm centred on a weld. This is a rather rough calculation. Fortunately the general conclusions are not too sensitive to the width although the precise values of critical defect sizes are dependent on this assumption.

Tensile peaks in the cyclic stressing of a ship leads to further yielding of the yield tensile areas in the welds. As the additional tensile load is removed so the resulting overall compressive strain causes the stress in the weld to drop from the original tensile value to a lower value. This is shown in figure 6.2. This shakedown is a highly beneficial effect because it means that it is unlikely that the full effect of original tensile residual stress and external loading stress and fatigue crack can occur at the same time.

In an old ship the residual stresses should have shaken down to perhaps half their original values except in the vicinity of repairs. In a new ship the shakedown should occur reasonably rapidly during larger storms.
After welding full yield stress occurs as a residual stress in the welds.

An external tensile load is applied, strain but no stress in the weld increases.

External tension is removed, strain and stress in the weld decreases. 'Shake down' has occurred.

Figure 6.2
Shake down of Residual Stress

The combinations and factors to be used for structural design and material selection purposes need to be calibrated by full scale measurement (to determine joint probability effects and loading sequence for shakedown estimation) in conjunction with reliability analysis. Clearly it is in practice impossible for a ship to meet the design wave in a 50 year storm without some shakedown of residual stresses having occurred. At present we can only estimate what the design combinations and safety factors should be for material selection in order to avoid fracture. Preliminary rough estimates of stress (or more precisely Young's modulus times strain), which could perhaps be used in conjunction with the -2 standard deviation fracture toughness values, are:

New or newly repaired condition:

<table>
<thead>
<tr>
<th>% yield</th>
<th>residual stress</th>
</tr>
</thead>
<tbody>
<tr>
<td>80%</td>
<td>25%</td>
</tr>
<tr>
<td>25%</td>
<td>still water bending moment</td>
</tr>
<tr>
<td>15%</td>
<td>wave bending moment</td>
</tr>
<tr>
<td>10%</td>
<td>slam</td>
</tr>
</tbody>
</table>

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Shaken down extreme condition:

- 50% yield  residual stress
- 25% yield  still water bending moment
- 25% yield  wave bending moment
- 10% yield  slam

The peak stress is 130% of yield for the new condition and 110% of yield for the shaken down condition. Although the stresses are higher in the new condition it will generally not be as critical for fracture as the shaken down extreme condition, because the defects may be expected to be relatively small.

6.3.4 Critical defect sizes

It is now possible to combine the toughness of the steel as given in Section 6.3.2 above with the stresses given in Section 6.3.3 in order to estimate critical defect sizes which might result in a brittle fracture of the hull girder. Typical results are given in Table 6.2.

<table>
<thead>
<tr>
<th>Steel Grade</th>
<th>Critical defect size</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>In weld</td>
</tr>
<tr>
<td>A</td>
<td>25 mm</td>
</tr>
<tr>
<td>B</td>
<td>100 mm</td>
</tr>
<tr>
<td>D</td>
<td>150 mm</td>
</tr>
<tr>
<td>E</td>
<td>1500 mm</td>
</tr>
<tr>
<td>AH36</td>
<td>75 mm</td>
</tr>
<tr>
<td>DH36</td>
<td>150 mm</td>
</tr>
<tr>
<td>EH36</td>
<td>500-1000 mm</td>
</tr>
</tbody>
</table>

These results are based on 0°C, 20mm plate thickness and relatively slow loading rates. The sensitivity to various factors (based on Sanz, 1981) is shown in Table 6.3.
Table 6.3
Sensitivity of critical defect size

<table>
<thead>
<tr>
<th>Change</th>
<th>Typical % Change</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Toughness</td>
</tr>
<tr>
<td>+ 10°C</td>
<td>+12%</td>
</tr>
<tr>
<td>+ 5 mm plate thickness</td>
<td>-15%</td>
</tr>
<tr>
<td>Strain rate, 10 times increase</td>
<td>-33%</td>
</tr>
<tr>
<td>1 standard deviation (of Charpy - $K_c$ correlation)</td>
<td>+10%</td>
</tr>
</tbody>
</table>

The above table gives some indication of the size of changes in critical defect size with effects that change material toughness. Clearly allowance must be made for actual temperatures, thickness and strain rates in any calculations. The standard deviation is used in reliability analysis either directly or for setting safety factors.

6.3.5 Crack arrest

In some circumstances brittle fracture may start but then be arrested because the crack runs into an area of low stress or high toughness steel. A running crack has a very high strain rate, perhaps two orders of magnitude higher than for ordinary wave loading. This (see Table 6.3) results in a large reduction in toughness and therefore makes the crack more difficult to arrest.

It is important that a small crack running from a very localized area of low toughness (e.g. a bad weld) should be arrested by the surrounding plate.

It would be convenient if a large crack running in low toughness plate could be arrested by high toughness and sometimes referred to as "crack arrest strakes" which are typically found on the corners of a ship's hull (see Figure 6.1).

Crack arrest calculations can be performed on running cracks in several ways:

The most accurate way is to perform a finite element dynamic analysis of the running crack. Whilst possible this is at present used more for research than for practical engineering calculation.

A good approximation in many cases is to simply consider the energy balance. The crack initiates approximately when the applied stress intensity factor is greater than the material toughness. The crack then runs and releases elastic strain energy from the structure as it extends. It also absorbs energy at the crack tip as it fractures the material ahead of the crack.

The crack will arrest if the total energy absorbed becomes greater than the energy released.

(Crack initiation can also be defined in energy times: a crack will initiate if the energy released for a small crack tip extension is greater than the energy absorbed).
Arrest calculations applied to ship structures suggest that:

a) E or EH grades have a high probability of arresting small (say 25mm) fractures running from very localized areas of brittle material. Lower grade steels have a significantly lower probability of arresting these defects. However, stress history effects may be important and beneficial. For example a defect in an area of very low toughness may initiate at a relatively low stress and then be arrested by surrounding plate of only moderate toughness. The arrested defect may then be stable with the crack tips in the moderate toughness plate at relatively high stress.

b) Welded crack arrest strakes are not likely to be effective in arresting a crack of any significant length. They will be beneficial for resisting any defects growing from brittle welds to themselves.

Riveted crack arrest strakes were used on the Liberty ships, apparently with some success. This is because:

- The additional material reduced the working stress levels.
- The riveted strake has much greater capability of resisting a running crack because the crack cannot run directly into the riveted plate.
7. INSPECTION

7.1 INTRODUCTION

Inspection is required to identify the following forms of structural damage before they result in a catastrophic, polluting or expensive failure:

a) Corrosion
b) Fatigue cracks
c) Indications of inadequate design or fabrication
d) Impact damage
e) Coating (if any) breakdown

The following sections discuss inspection methods and how a justifiable inspection strategy can be determined for an F(P)SU. They also demonstrate the significance of the steel toughness in inspection planning calculations.

7.2 INSPECTION STRATEGIES ADOPTED FOR TRADING TANKERS AND RELEVANCE TO F(P)SU's

The inspection necessary for an F(P)SU to operate safely is not easy to determine. Regulations for trading tankers require either:

Special Surveys, which include a significant amount of close up visual inspection every four years or:

Continuous Survey which results in some inspection of every tank on a 5 year cycle.

These regulations are not necessarily relevant to avoiding excessive fatigue cracking in a permanently moored F(P)SU because:

a) The amount of time spent at sea by a trading tanker is typically about 75% of the total available time.
b) About half the time is spent at the loaded and half at the ballast draught. This spreads the fatigue damage from side shell loading over two areas whereas, to simplify the mooring design, F(P)SUs often operate at approximately constant draught so concentrating the damage at a particular level.
c) Many tankers have spent long periods laid up in calm environments.
d) The weather conditions in certain oil field areas can be significantly more severe than usually seen by a trading tanker. A trading tanker can sometimes "weather route" to avoid particularly severe weather.
e) Unless thrusters are fitted, the heading a moored tanker makes with the sea is not controllable. In contrast a trading tanker can change course to reduce motions and thus will change the fatigue loading (probably but not necessarily for the better).

7.3 INSPECTION ACCESS

Inspection access is always difficult because:

a) The tanks need to be gas freed prior to entry and this can take many hours.
b) In oil tanks, even after crude oil washing, the structure is covered with oil which is slippery to walk on and covers up even moderate cracks.

c) In ballast tanks the steel is sometimes covered with a thin slime which is again slippery and conceals small cracks.

d) There is no easy method of inspecting any of the side shell higher than about 2m above the level of the bottom stiffener flanges:

Climbing the stiffeners is possible but dangerous.

Ladders can be used to gain access to about 10m above the bottom but they are not really practical for reaching above this and the deck head (underside of the deck), will typically be about 20m above the bottom.

Scaffolding can be erected inside the tanks to be inspected. This is time consuming but provides a stable base for inspection and can allow good access to the deck head.

A common method of inspecting the side shell and deck head is by rafting. This usually involves floating in a small rubber dingy with the tank filled to various levels. Access to the deck head is still difficult because the water level cannot be above the clearance to the transverse frames. This in practice means that close inspection of the deck head is not possible by this method either.

For new designs we would recommend that inspection access is considered at an early stage and that the amount of access provided is justified as part of the overall inspection strategy. One tanker, which has just been converted to an FSU for a North Sea field, is unusual in having small holes in the deck just away from the transverse frames. It is believed that these were intended for fitting wire ropes through so that a bosun’s chair could be used for the side shell connection inspections. It is doubtful however that these have ever been used.

Other possibilities for inspection include:

The use of “cherry picker” hydraulic platforms. These would require larger hatches in the deck for the entry of these devices but would seem to be a practical possibility for new designs.

Building ladders and walkways into every transverse frame. This would be expensive and there would also be concern about the safety of the walkways which would inevitably corrode with time.

Careful inspection is vital to the continuing integrity of the hull structure. Good access is important if a satisfactory inspection is to be performed.

7.4 INSPECTION METHODS

7.4.1 Inspection for fatigue cracks

When inspecting for cracks, it is usual to rely on visual inspection. As there are so many structural connections in an oil tanker that it is not considered practical to use the more sensitive Non-Destructive Testing (NDT) methods. An internal visual inspection will largely reveal the state of the structure and areas which require closer examination can then be targeted.

NDT tests such as ultrasonics or magnetic particle can be used to assess the size of defects. They can also be used to check the more highly stressed and fatigue prone areas.
It will be necessary to consider the probability of detecting cracks of various sizes in different details. Consideration has to be given to accessibility and surface condition, e.g. painted/ corroded/ oil covered/ slime covered. An example probability of detection curve is given in Figure 7.1.

7.4.2 Inspection for corrosion
This will usually be performed visually for pitting with a limited number of thickness measurements made with an ultrasonic thickness gauge. Drilling can also be used but although this is accurate, it is destructive and therefore may not be convenient.

![Probability of detection curve](image)

**Figure 7.1**
Probability of detection curve

7.5 Deterministic inspection modelling
This approach sets the inspection interval for each location on the basis that the next inspection should occur before the largest undetected defect reaches a critical size. This method is deterministic in the sense that the failure assessment is made from crack growth and fracture mechanics calculations which actually determine the inspection intervals based on calculated values. An inspection procedure will put different time intervals for different areas of the structure and the inspections can therefore be optimised. Typically safety is built into the calculation by using an upper estimate of the crack growth (e.g. mean + two standard deviations) and applying a safety factor in the critical crack size calculation.

Example critical defect sizes are estimated in Section 6.3.4, crack growth rates in Section 7.5.1 and the inspection interval required is developed in Section 7.5.2.

7.5.1 Crack growth rates
Crack growth rates can be calculated from the Paris equation described in Section 5.4.6.

It is necessary to have the following information:
a) Overall stress range data - from analysis or measurement.
b) Stress intensity factors (K - see section 5.4.6) describing the stress in the vicinity of the crack and allowing for the local details of construction. These stress intensity factors can sometimes be obtained from handbooks but frequently the geometries of interest are more complicated than can be found in handbooks and finite element analysis, using a program capable of computing K, will be required.
c) material data - obtained from correlations with Charpy tests, specific fracture toughness tests on the ship material or published results of laboratory tests on similar material.

7.5.2 Inspection timing
A typical crack growth curve for a defect in the bottom shell plating is shown in Figure 7.2. Time 0 corresponds to a surface crack of about 0.1mm deep and about 10mm long. This grows through the 20mm plate thickness and then extends across the width of the plate as a though thickness crack. The unfortunate characteristic of this curve is that crack has very little chance of being found by inspection when it is less than say 50mm long. However, the crack growth rate is very high once the crack exceeds that length and so it quickly reaches the critical lengths shown in Table 6.2. Therefore, if an extreme stress were to be applied in the few years after a 50mm defect was just missed by an inspection, a brittle fracture might occur.

In some industries the likely locations of fatigue cracks are well known and the inspections allow much smaller defects to be found. In these cases where sensitive and well targeted inspection is possible, a deterministic inspection procedure as described at the beginning of Section 7.3 is practical. However, for ship inspection we have a high uncertainty of finding potentially critical defects but the probability of the extreme load and worst material properties occurring is relatively low and this to some extent counteracts the poor inspection. In order to assess this situation a probabilistic method of estimating the reliability of inspection is preferable.

![Crack growth curve](image)

**Figure 7.2**
Crack growth curve
Transverse butt weld in ship's bottom.

Note:
Allowance made:
- a) for residual stresses
- b) for beneficial effect of longitudinal stiffeners.
7.6 PROBABILISTIC INSPECTION MODELLING

The deterministic method described in Section 7.4 is quite good for planning the inspection of a limited number of known problem areas but is sensitive to the loading and response assumptions made by the analyst and does not take into account the probability of missing a defect, which is an important consideration when large areas are being inspected.

Also, when a lot of inspection is being undertaken, the inspection results provide useful information about the rate of crack growth which can be fed back into the calculation using Bayesian probability techniques (ref Thoft-Christensen and Baker). The probabilistic method can allow for many of the uncertainties involved in selecting inspection strategies. It can therefore determine the most cost-effective options with acceptable safety. Whilst these methods are mainly being applied to jacket structures, their extension to ship structures is attracting increasing interest. The calculation must take into account the very large number of potential initiation sites and the low redundancy with regard to brittle fracture. An approach used by WS Atkins is to select an acceptable probability of failure of the overall structure and to divide this between the various potential failure locations.

Note that the implication of this is that a non-redundant ship structure will be less reliable than the individual components from which it is built. In contrast, a redundant jacket type of structure may be more reliable than its components.
From knowledge of a: wave climate
b: ship response
c: fracture mechanics

From assessment of inspection procedures and tests (tests presently more relevant to offshore structures)

From crack growth data and fracture mechanics analysis of detail

(method for 1 location shown. Many typical locations need to be analysed in practice).

Figure 7.3
Probabilistic Inspection planning
The target reliability for each detail is assessed as follows:

Suppose the overall required probability of failure P is $10^{-4}$/year and the corresponding acceptable probability of failure of any of n details is Q, where Q is to be determined.

Then the probability of survival of the structure is $1 - P$.

If it is assumed that the probability of failure of each detail is independent of the other details, the probability of survival of n details is then $(1 - Q)^n$.

For the two probabilities to be equal:

\[
(1 - Q)^n = 1 - P  \\
1 - Q = (1 - P)^{1/n}  \\
Q = 1 - (1 - P)^{1/n} 
\]

For the range of interest the highest acceptable failure probability for each of n details is essentially $\approx P/n$.

In practice the individual detail failure probabilities are not independent. This is mainly because they have highly correlated loading. However for the low acceptable failure probabilities of interest the correlation makes the $P/n$ estimate slightly conservative. (Calculations performed to date have shown $P/n$ overestimates the required detail reliability but by less than 25%.) An interesting result of the correlated loading is that the probability of a small defect causing a failure at an unusually high load level is reduced. This is because it is likely that a larger defect will cause a failure before the unusually high load event occurs.

Initially it might be assumed that the reliability of the structure will largely be determined by about 1000 critical details or metres of weld. Therefore the initial target reliability for each detail, after allowing for the overall inspection plan, should be about $10^{-7}$ per year. The assumption of 1000 critical details is only an approximation and when an inspection plan has been developed the actual overall reliability can be compared with the $10^{-4}$/year overall requirement and the local $10^{-7}$/year target adjusted if necessary.

Figure 7.4 shows example results for various inspection scenarios, defined fatigue lives and steel grade assuming defects in 1000m of butt weld. The examples are for a particular case and the assumptions would need to be adjusted for any other application. Also the basic input assumptions will benefit from further research. However, the general trends are expected to be valid. The preliminary conclusions are:

a) It is difficult, even with frequent visual inspection, to maintain a low probability of failure of a ship type of structure as the actual life approaches the design - two standard deviation design fatigue life. It would be preferable for the actual life to be no greater than about one third of the two standard deviation fatigue life.

b) Good steel toughness is very important for fatigue reliability. More work is required in order to obtain realistic requirements but it would appear that for decks and bottoms mild steels should be D grade or better and high yield strength steels should be EH or better. For the side shell between the "crack arrest" strakes D or DH may be acceptable owing to the lower stresses closer to the neutral axis in vertical bending.

c) Longitudinal stiffeners should have a similar grade steel to the plating.

d) For details in areas of essentially low fatigue life (e.g. 20 years design fatigue life and 20 year actual life) the fabrication defects affecting long term reliability are typically 2.5mm - 5mm long. (Design fatigue life is the -2 standard deviation value).
NOTES:
1) Fatigue life is -2 standard deviation design value (mean = 2.5 x this value) for an E curve.
2) A 30% reduction in peak stress is equivalent to a doubling of critical crack size (equivalent to a movement of along crack length axis).
3) Steel grade values are indicative for highly stressed areas.
4) Defect distributions based on surveys. Gross fabrication defects not included – effect is to decrease high reliabilities and to make inspection more worthwhile.
5) Failure is by fracture.
6) Temperature is taken as 0°C mean (but could be a variable).
7) Steel grades based on Charpy correlations to fracture toughness with 100 year return stress taken as 60% of yield.
8) Residual tensile stress assumed only local to the stiffener welds.

Example results of probabilistic inspection / steel grade / fatigue life

Figure 7.4

<table>
<thead>
<tr>
<th>Mean 100 year critical crack length (mm)</th>
<th>Steel Grade</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>A</td>
</tr>
<tr>
<td>25</td>
<td>A</td>
</tr>
<tr>
<td>50</td>
<td>A</td>
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</tr>
<tr>
<td>800</td>
<td>A</td>
</tr>
<tr>
<td>1600</td>
<td>A</td>
</tr>
</tbody>
</table>

TARGET 10^{-4}

Annual failure probability in 20 years

10^{-8} 10^{-7} 10^{-6} 10^{-5} 10^{-4} 10^{-3} 10^{-2} 10^{-1} 10^{0}

60 yr fatigue life annual insp
60 yr fatigue life 4 yrly insp
60 yr fatigue life no insp
20 yr fatigue life annual insp
20 yr fatigue life 4 yrly insp + yrs 14 & 18
20 yr fatigue life 4 yrly insp + 80% p.o.d.
20 yr fatigue life no insp
When the design fatigue life is high (e.g. 60 years fatigue life for 20 years actual life) the fabrication defects leading to low reliability are much bigger.

e) When the design fatigue life is much greater than the actual life then the fatigue test results have to be treated with care. Our calculations suggest that the low reliability starts to be dominated by "large rogue" defects and that the likelihood of rogue defects may be much less in some, usually considered, poor fatigue details. This implies that ordinary fatigue lives, based on S-N curve data, may be a poor indications of where to look for fatigue cracks! In particular it would seem quite possible for an E classification transverse built weld to be less reliable than a stiffener penetrating a bulkhead with a G classification detail. Statistics based on laboratory tests must be handled carefully in this region since rogue defects will have generally been filtered out.

f) Inspection significantly improves the reliability of low fatigue life structures. In certain cases there is demonstrable benefit in decreasing inspection intervals, from say four to two years in the later 1/3 of the hulls life.

g) There is some indication that the LRS classification requirements for steel grades may be a little lax for ships used continuously in more hostile environment such as the North Sea.

h) Crack arrest characteristics of plate are of importance in providing some safety against a brittle fracture initiating from a very small defect in a brittle weld. Typically a crack of say 20mm length in a brittle longitudinal weld would be arrested if surrounded by E grade plate. However, E grade "crack arrest" strakes at the top and bottom of the side shell are not capable of arresting a crack of any significant length.

7.7 Stress monitoring
There has been a recent upsurge of interest in monitoring stresses and presenting the results in real time on the bridge. This has been caused by the introduction of the Lloyd's 'Sea' notation which has encouraged ship owners to install stress monitoring. Although not directly relevant to F(P)SU's, as the ability to change speed or headings to reduce stress levels is not available, it can be useful for determining inspection intervals. Where a trading tanker is deployed which has had stress monitoring equipment fitted, there would be the added benefit of having a "stress history". This would give an assessment of the present fatigue exposure and would help to indicate which areas might be prone to fatigue. Stress monitoring an F(P)SU has the benefit of providing input to inspection planning calculations and allows long term behaviour to be assessed. This should eventually lead to better design guidance.
8. CONCLUSION

8.1 IMPORTANT FACTORS FOR F(P)SU DESIGN

In order to obtain satisfactory structural performance a mixture of rule design, to take into account the considerable experience already in the industry, and analysis based design, to allow for special factors and especially for fatigue, is recommended.

Rule based strength design is probably adequate in the majority of cases although harsh environments in combination with the lack of heading and storm avoidance options open to an F(P)SU might impose more severe loading than for a typical tanker.

Fatigue design should be based on first principles analysis and should take into account the proposed steel grades and the amount and type of inspection that is planned. It should not be assumed that the overall section modulus that is required for extreme loads will also be satisfactory for fatigue.

Fatigue design should account for the stress concentrations implicit in many details. Care must be taken to allow properly for notched details where the stress may be infinite and a fracture mechanics approach should be used to estimate crack growth rates from initial defects. Trying to match conventional C-W S-N classes to ship details can give over optimistic results.

Many standard shipbuilding details are not very satisfactory on trading tankers and would be very unsatisfactory on an F(P)SU moored in the North Sea. In these cases special details involving for example soft toed brackets will be required.

Side shell design is likely to be dominated by fatigue requirements, especially in the areas just below the loaded and ballast water lines.

The detailed design of the connections between shell longitudinal stiffeners and transverse frames requires special care since these areas cause a lot of trouble in practice.

Design rules for ships in fatiguing environments could be produced as an alternative to requiring first principles analysis for each ship. The rules would provide guidance on the required hull, stiffener and plate section properties and details. Standard details would be provided with different fatigue gradings.

Inspection access is difficult. Provision should be made for improving access perhaps by providing openings through which a "cherry picker" hydraulic platform could be lowered.

Inspection planning should be performed based on statistical models of the fatigue decay, the inspection procedure and brittle fracture failures.

Steel grade should be selected with an understanding of it's significance for the reliability of the structure. Significantly more high grade (E/EH) steel should be used in Hulls working in environments which are fatigue severe.

It may be necessary to reduce inspection intervals as a ship ages. It may also be advisable to reduce the allowable stresses as the ship ages. For a moored F(P)SU the environmental loading can be reduced by keeping a deeper draught to avoid slamming. The still water bending moments and shear forces can be reduced by selection of loading and ballasting arrangements.

Some ships may not be suitable for duty in a fatigue-harsh environment.

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On-board stress monitoring with real-time displays on the bridge can improve the structural reliability of trading ships by prompting the master to change course or speed. The benefits for a moored F(P)SU are that the fatigue damage rate and extreme stresses can be monitored and used as input to the inspection planning calculations and safety cases and for redeployed trading tankers their fatigue exposure could be assessed.

Patterns of tank filling should be established which benefit the ship structure by:

- Avoiding excessive sloshing.
- Keeping the side shell connections in compression when possible, to reduce fatigue damage.
- Reducing longitudinal bending and shear stresses as far as practical and necessary.

8.2 RESEARCH REQUIRED TO IMPROVE F(P)SU DESIGN

The most important areas of research required to improve the structural safety of F(P)SUs are:

a) Assessment of the slamming behaviour of real ships. Laboratory and very limited full scale experimental work suggests that slamming forces are very variable but can be very large. These forces are probably not too much of a problem for a new ship because they can be resisted by elasto-plastic deformation of the hull. However in an old ship the fatigue cracks may cause a brittle fracture before the full deformation has occurred. A statistical treatment of the probability of slamming and the probability of the response would be much more useful than the existing deterministic data available to analysts.

b) Assessment of the toughness properties of ship steels in terms of $K_t$, $J$ or $\delta$, as well as Charpy values should be obtained from the tests. These results will allow a better assessment of resistance to fracture.

c) The significance of residual stresses on fracture is worth further investigation. (The results given in this document are purely theoretical.)

d) Further analysis and testing of ship type details should be performed in order to provide rules to avoid the need for complex structural analysis for each ship detail.

e) Studies, possibly in conjunction with full scale measurements could provide useful data on peak stresses and stress sequences. The latter is important in relation to shake down of welding residual stresses.

f) Ship inspection planning is an important area, which will benefit from further development work. In particular the sensitivities to the input assumptions, the as-built defect sizes (including rogues) typical of ship construction and the probability of detection curves need further investigation.

g) Full scale measurements taken on ships should be checked against the predictions from analysis.
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