4. LOADS AND DESIGN STANDARDS

4.1 Quasi-Static Wave Load Analysis

Two linear methods of calculating wave forces are normally used; Diffraction Theory and the Wave Force Equation. In their normal form neither are applicable to the Clam since both assume a linearised free surface. The forces on the SEA Clam have been shown in experimental studies to be sensitive to non-linear free surface effects because of the very small freeboard. Therefore a modified Wave Force Equation analysis has been used which takes account of the variable depth of immersion across the device.

The complex interaction of the device with the incident wave system when generating power has not been considered since the peak structural forces occur when the bags are either fully inflated or fully deflated and the device is inactive. Similarly device motions are not included since the peak internal forces will occur with the device stationary and the applied external forces in equilibrium.

The wave load analysis method used is a development of the method used by RPT for the 1986 Report. The loads are calculated at two parts of the wave cycle; with the device poised symmetrically across the wave trough and across the wave crest. The vertical position of the device is adjusted until it is in static equilibrium. Vertical hydrodynamic forces due to water particle acceleration are conservatively ignored. Vertical buoyancy forces and horizontal inertia forces are then calculated around the perimeter.

To calculate the bending moment about the horizontal axis (heave moment) the first moment of the vertical force distribution is calculated by numerical integration.

The ring is one degree indeterminate for symmetrical bending about the vertical axis (surge bending) so a release has to be introduced on the axis of symmetry. The surge moment distribution is then calculated using the flexibility method.

To allow the loading from a number of different wave heights and wavelengths to be rapidly assessed the calculation is programmed into two spreadsheets; one for the crest poised case and one for the trough poised case.

The calculations give bending moments about the vertical and horizontal axes but since the section is unsymmetrical these are not the principal axes. Heave bending for example gives a secondary surge moment because of the inclined principal axes. Because of the indeterminacy of the structure in surge bending it was not immediately clear how this secondary moment would be distributed around the perimeter of the device. A simple finite element analysis was therefore carried out to examine the behaviour of the ring under unsymmetrical bending.

Table 4.1 shows the distribution of forces around the structure as derived from the finite element model for an extreme wave load case. The moment results have been normalised to the maximum value (at the centreline) calculated by resolving the applied moment into the principal axes. There is a negligible error at the centreline, where the ratio is effectively unity, which is caused by the redistribution of surge bending. This shows that it is only necessary to resolve the applied moments into the principal axes for the section.

Table 4.1 Distribution of Forces Calculated by Finite Element Analysis (See Fig 4.1 for definition of model).

Member	Joint	$\frac{M}{Mmax}$	$\frac{T}{Mxmax}$
1	1	1.00011	0
	2	0.91	0
2	2	0.79	0.46
	3	0.14	0.46
3	3	- 0.11	0.47
	4	- 0.81	0.47
4	4	- 0.93	0.0
	5	- 0.93	0.0
5	5	- 0.81	0.47
	6	- 0.11	0.47
6	6	0.14	0.46
	7	0.79	0.46
7	7	0.91	0
	8	1.00011	0

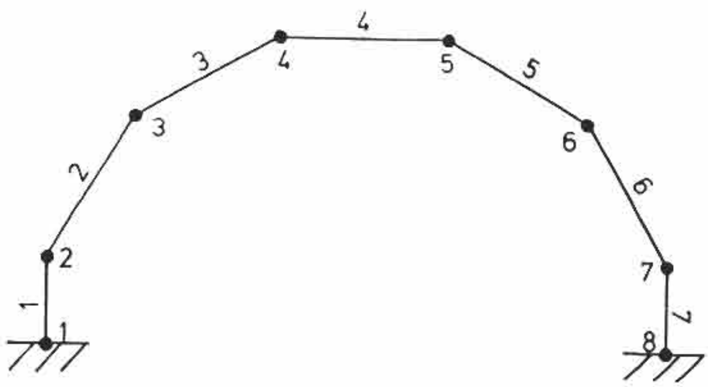

FIGURE 4.1 F.E MODEL DEFINITION

Table 4.1 also shows the distribution of torsion around the structure normalised to the resolved heave moment, to which it is closely related. This shows that the maximum torsion is approximately 50% of the heave moment for this case. This relationship has also been assumed to apply for all other load cases.

4.2 Calculated Wave Loads

4.2.1 Parametric Study

A parametric study was carried out to determine the critical wave length for a range of wave heights and the relationship between bending moment and wave heights for these critical wavelengths. The maximum moments were invariably found to occur with the device in the wave trough and the critical wavelength was generally found to be just less than twice the device diameter (taken as being 60m). The limiting wave steepness was taken as wavelength/height < 7.0 (Ref 3). The crest length was taken as being infinite.

Figure 4.2 shows the relationships between peak moments and wavelength for a monochromatic sea with a waveheight of 6m. In this case the peak surge moment occurs with a wavelength of about 90m and the peak heave moment occurs with shorter wavelengths. Wavelengths of less than 40m have been ruled out for this wave height by the maximum steepness criterion.

Figure 4.3 shows the relationships between peak heave and surge moments and significant wave height for the operating case with bags partially inflated. From this it can be seen that the heave moment reaches a plateau at relatively modest wave heights but that the surge moment keeps on increasing with wave height. It was found, in fact, that the magnitude of the surge moment was limited by wave steepness rather than wave height.

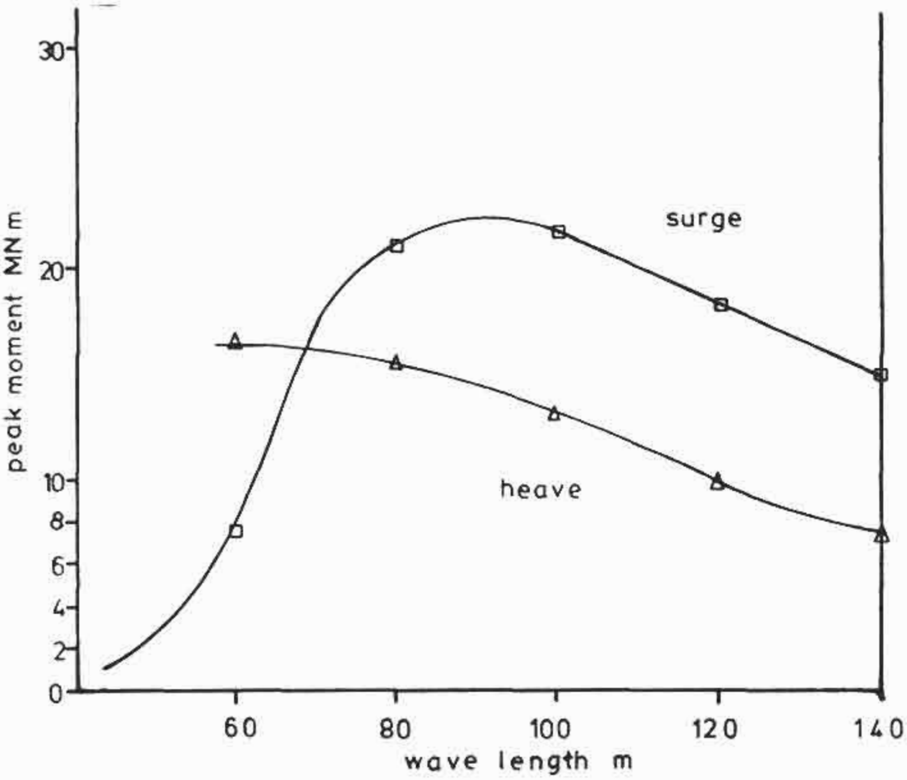

FIGURE 4·2 RELATIONSHIPS BETWEEN PEAK MOMENTS AND WAVE LENGTH FOR $H_{max} = 6m$.

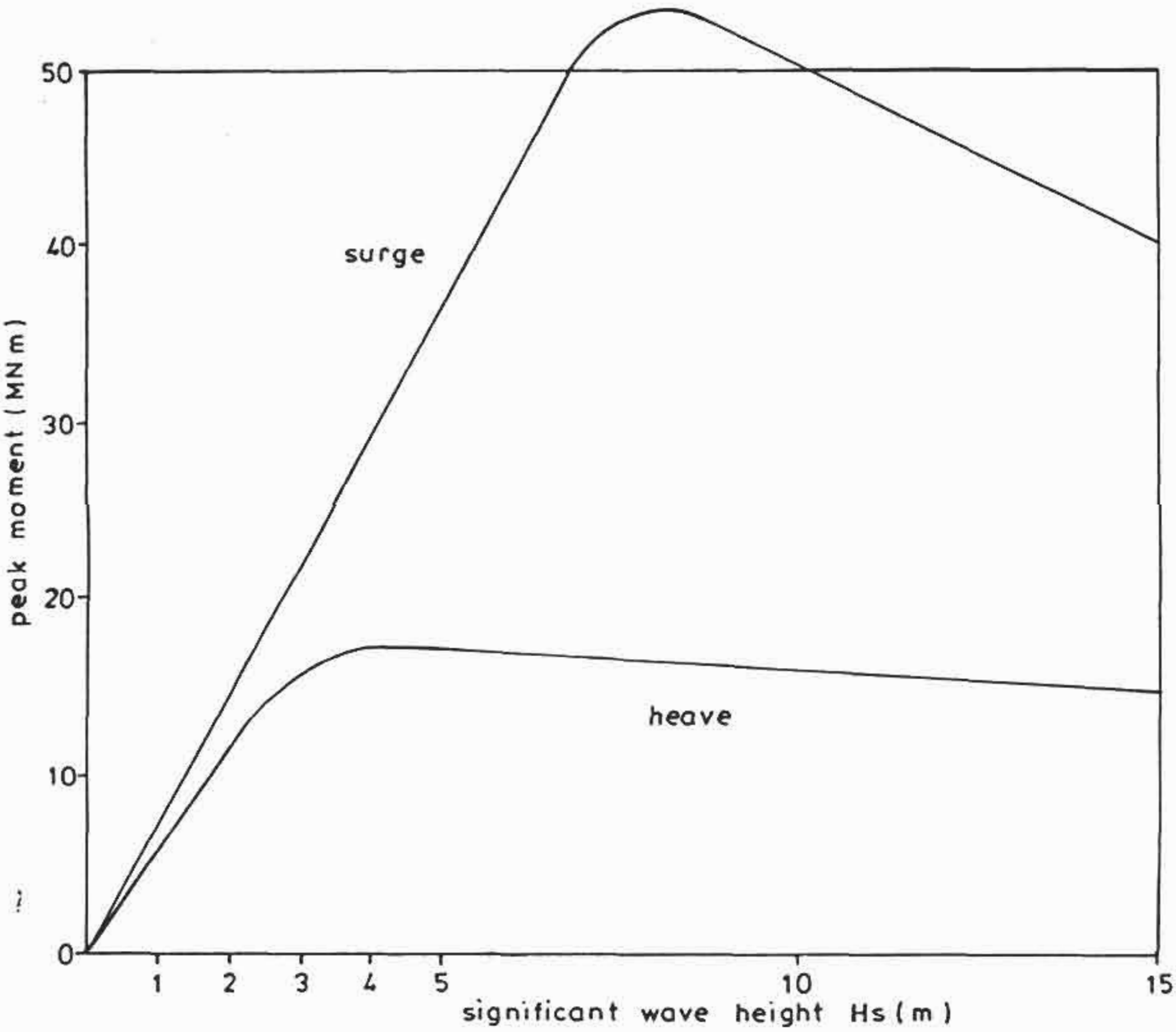

FIGURE 4·3 RELATIONSHIPS BETWEEN PEAK MOMENTS AND SIGNIFICANT WAVE HEIGHT FOR CRITICAL WAVE LENGTH.

4.2.2 Design Operating and Extreme Wave Loads

Since the actual site for the Clam is not defined the wave environment has been selected rather arbitrarily. The design wave has been assumed to have a wavelength of 120m and a limiting steepness of 7 giving a height of 17m.

A typical site might be in an offshore location at South Uist where wave data exists and where the one in 100 year return extreme conditions are represented by a sea state with a significant wave height of about 17m. The one year return conditions at this site have a significant wave height of about 12.5m. It has therefore been assumed that an individual wave with a height of 17.0m will occur quite regularly and the design wave has been used for both normal and extreme wave loads. However, whereas in the extreme case the device has been considered either fully inflated or deflated, whichever gives the most severe loading, the normal wave load case has only been applied to a device with the bags 50% inflated.

Table 4.2 gives the design wave loads used in the analysis of the hull sections.

Table 4.2 Design Wave Loads

		Extreme	Operating
Heave Bending Moment	(MNm)	23	16
Surge Bending Moment	(MNm)	53	53
Torsion	(MNm)	11.5	8

4.2.3 Fatigue Wave Environment

In the absence of site specific information the wave spectrum used in the 1986 report was re-used in this study. The quasi-static wave load spreadsheets were run for all the fatigue waves giving the bending moment results shown in Table 4.3.

Table 4.3 Fatigue Wave Loads

(m)	Range* (m)		Range* (MNm)	Occurrences (MNm) per annum
1	60	5.2	1.1	3 618 981
3	104.9	13.6	15.8	828 085
5	149.8	12.2	17.4	192 849
7	185.3	11.7	17.5	45 618
9	209.9	11.9	18.2	10 942
11	232.2	12.0	18.6	2 653
13	247.6	12.5	19.5	657
15	267.7	12.4	19.5	162

* The Bending Moment (BM) range is defined as the algebraic difference between the moments at the wave peak and the wave trough.

Because of the sensitivity of the fatigue calculations to the spectrum used, a sensitivity study was also carried out with three different wavelengths for each wave height band as shown in Table 4.4.

Table 4.4 Fatigue Wave Spectrum (Sensitivity Study)

Wave Height m	Wave Length 1 (m)	2 (m)	3 (m)	No. of Occurrences per annum for each wavelength
1	48.0	60.0	72.0	1206327
3	83.9	104.9	125.9	276028
5	119.8	149.8	179.8	64283
7	148.2	185.3	222.4	15206
9	167.9	209.9	251.9	3647
11	185.5	232.2	278.6	885
13	198.1	247.6	297.1	219
15	214.2	267.7	321.2	54

4.3 Other Loads

Wave slam loads have not been assessed in this study since the only device susceptible to wave slam, the tubular device, was ruled out at an early stage. Considerations governing wave slam are discussed in RPT's 1986 report (ref 2).

Pressure differentials across panels have been assessed on the assumption that there will be a design external hydrostatic head of upto 7m above the top deck of the device in addition to a transverse hydrodynamic pressure due to wave inertia forces of up to 40 kN/m^2. The design head has only been applied across members surrounding unballasted compartments.

The flexible membrane anchorage force has been assumed to be 60 kN/m in the design of local attachments.

4.4 Design Standards

This study has not been sufficiently detailed to be able to confirm that the outline designs presented for costing meet the relevant standards in all respects. However it is intended that the designs shown will generally comply with the requirements of one or more of the following:-

- Lloyds Rules for the classification of ships - steel plated structures generally.
- BS 5400 - Part 10 - Design of Steel Concrete and Composite Bridges - for fatigue of steel structure.
- FIP - Recommendations for Design and Construction of Concrete Sea Structures.
- BS 8110 - Structural use of Concrete.

- BS 8007 - Design of Concrete to Retain Aqueous Liquids - Crack control.

- BS 6349 - Code of Practice for Maritime Structures - Sea keeping.

- Offshore Installations: Guidance on design construction and installation, 4th Edition, Department of Energy.

5. REVIEW OF 1986 DESIGN

The 1986 concept design had two faces of the cross section (i.e top and outside face) which were shell plated, and 2 sides only (inside face and bottom) which were tubular trusses. There were also tubular members vertically on the outside face to take vertical forces, with shear being carried in the shell plate.

Shells and tubular space frames do not mix well. One resists loads in a distributed sense, the other concentrates loads and carried them from node to node. Therefore a mixed design is necessarily inefficient as the loads need to be distributed then concentrated and vice versa as they are carried round the structure.

RPT initially proposed an alternative cross-section (figure 5.1) which is in fact intermediate between the original tubular design and a shell plated "hull" and has the following advantages:

- fewer tubular members.

- provision for venting slam pressures.

- better continuity of stiffening at the top.

- fewer tubular nodes.

However this scheme still has the disadvantage of the tubular space frame structure of:

- complex welded joints in the splash zone (considered very bad practice in offshore structures for inspection, corrosion and maintenance).

- potentially fatigue prone details at points of concentrated load.

- premium rates will apply to the cost of fabrication.

As far as the fabrication of the tubular units is concerned, there will be no difficulties that cannot be overcome by good detailing but the fabrication will inevitably be complex - i.e:

- mixed welding types with limited scope for automatic welding.

- sensitivity to dimensional accuracy/fit up/distortion.

- through thickness stressing at nodes needing high grade steel (Z quality).

It was subsequently found that the alternative shell plated "hull" concept of conventional ship construction was easily adapted to the shape required and from preliminary estimates was not significantly heavier. The hull concept does not suffer from any of the above problems and offers several potential advantages for maintenance of structure and replacement of plant.

Thus RPT considered that the disadvantages of the tubular concept would be outweighed only if there were very considerable (say 30-50%) gains in device energy efficiency. As no model tests with a tubular frame concept have been carried out it was not considered worth pursuing the tubular option further. It was agreed at the first progress meeting to concentrate resources instead on the design of the hull structures and to introduce a composite hull design to replace the tubular truss option.

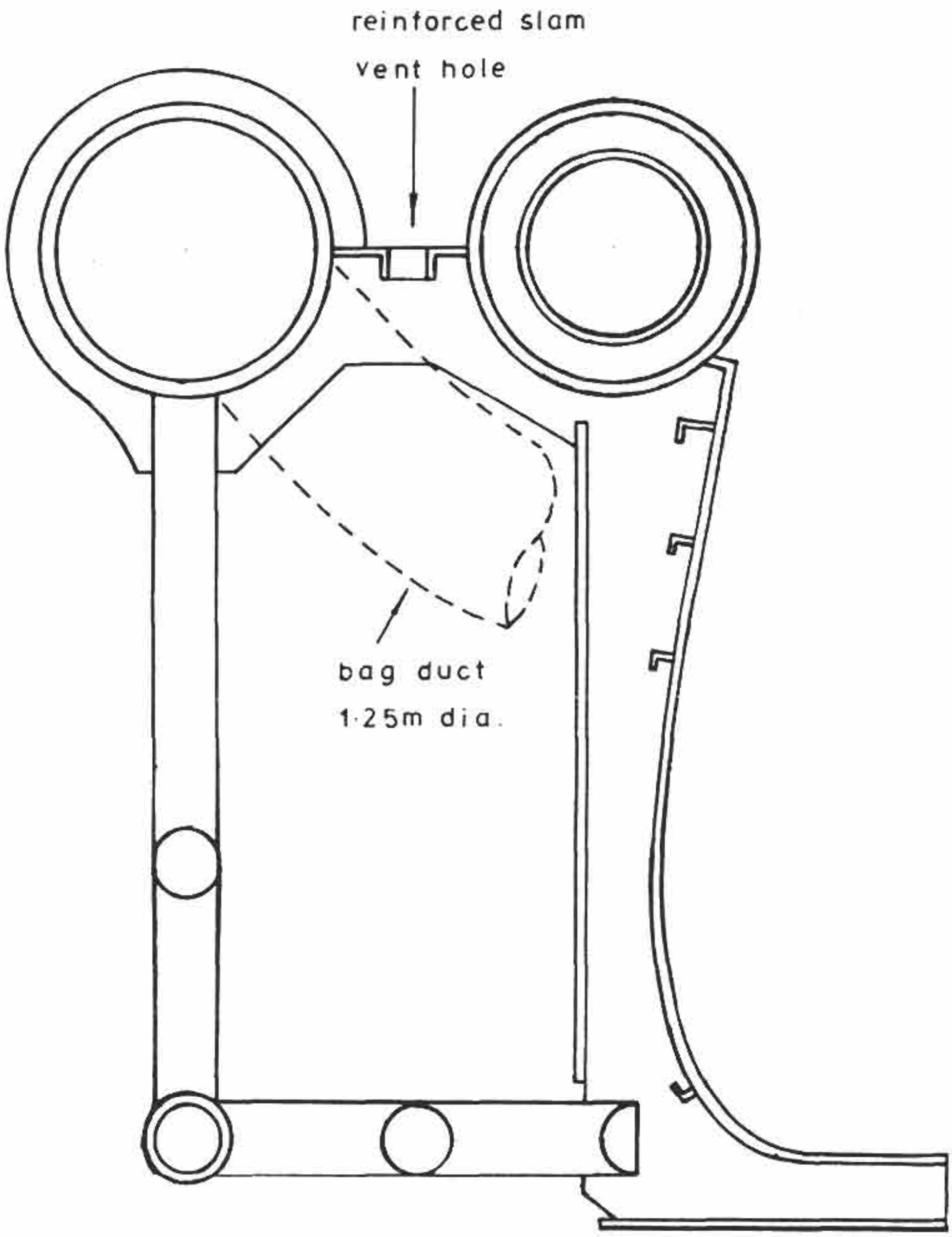

FIG 5·1 REVISED TUBULAR STEEL DESIGN