Analysis of the SEA-OWC-Clam Wave Energy Device

Part B: Structural Integrity Analysis

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Abstract

In this paper the dynamic loads determined for the SEA-OWC-Clam wave energy device, treated as a floating offshore structure with six degrees-of-freedom with partial internal sealed-off channels, are applied to assess its structural integrity. This task necessitates the matching of the boundary element determined dynamic pressures to the corresponding Finite Element Analysis (FEA) to determine the effective von Mises stress levels. A decomposition of the total load to permit attribution to different aspects of radiation and diffraction loading is presented. Possible modification of structural details is identified by undertaking a more local stiffened plate analysis to improve structural details.

Keywords: wave-energy, oscillating water column (OWC), fluid structure interaction, linear FEA, stress reduction, corrugated plate

1. Introduction

The historical development of the SEA-OWC-Clam wave energy device and the choices concerning the detailed hydrodynamic and motion analysis to determine the resulting loads was detailed in Part A [1]. Within [1] it was argued that a 6 degrees-of-freedom system, rather a 78 degrees-of-freedom system was sufficient for initial structural integrity assessment. Here the deliverables of that conventional 3D hydrodynamic and motion response analysis of a floating offshore structure are now applied using FEA. The configurations previously designated the ‘open’ device and the ‘closed’ device, are examined to appreciate how the required openings accommodate the water ingress and the partial construction of the J- and N-shaped OWCs affect the structural response characteristics.

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There are two anti-sloshing plates within each OWC, for each module, and they are included in the structural model as they provide some reinforcement to the structure.

This companion paper will assess the structural integrity of the SEA-OWC-Clam to full scale dynamic stress induced by waves consistent with the observed South Uist wave climate [1].

1.1. Paper structure

Having indicated variation of representative 3D dynamic pressure distributions over the wetted surface of the device, the transference of this dynamic load from hydrodynamic boundary elements to structural finite element is addressed. With structural modelling details presented, finite element formulations of the ‘closed’ and ‘open’ device forms are discussed prior to explaining the FEA approach adopted. Phasing of the wave loading relative to the incident wave of frequency, ω , is captured by using real and imaginary time independent descriptions, with time dependence represented by $e^{-i\omega t}$. Hence the von Mises effective stress levels will be presented as an envelope of magnitudes. Having discussed the presented stress levels a more local structural investigation is undertaken prior to paper closure.

2. Pressure distribution

A quarter of the closed and open forms of the device are explicitly modelled using 810 and 1440 boundary elements respectively. Thereafter representative pressure distributions are generated from summation of incident waves, diffracted waves and radiation waves for selected wave frequencies in accordance with equations (14) of [1].

Within the hydrodynamics, the solution technique provides the radiation velocity potentials on the explicitly defined quarter structure (treated as a solution domain) and Table 2 of [1] indicates how to generate values on its images; the images are not explicitly generated, but their influences are addressed within the solution technique applied. Diffraction velocity potentials are similarly determined on input structure and its image structures automatically.

The fluid loading of the structure and the rigid body motion responses of the structure are recorded as complex numbers. This provides both the amplitude and the phasing of the

hydrodynamic excitation loads and the motion responses of the structure relative to the incident wave frequency. An alternative and more pragmatic way of looking at the complex number representation is that the maximum responses do not coincide with the arrival of the crest of the wave at the centre of the structure.

2.1. Matching BEM & FEA meshes

The Fredholm integral equation description of the rigid-body based fluid-structure-interaction [2–4] leads to a fully populated matrix formulation when the boundary element approach is adopted. The fully coupled nature of the hydrodynamic analysis can lead to relatively fewer boundary elements than the corresponding structural modelling. Hence the mesh requirement for a structural analysis performed by finite elements must be more refined to capture the physics. Furthermore to facilitate easy transfer of hydrodynamic loads to the structural model each boundary element centroid (where hydrodynamic data is determined) is matched with the common node of four structural finite elements as depicted in Figure 1. Thereafter the complex pressure acting on each panel of the hydrodynamic model is transferred as a distributed load to the structural model; the finite element method transforms the distributed load into nodal loads via the shape function of the selected structural element. Undertaking the steady-state harmonic structural analysis the elastic displacements and hence the associated strain and stress distributions over the structure are determined.

Fig. 1 Matching of hydrodynamic and structural meshes.

Different arrangements would be necessary if the less common hydrodynamic higher order isoparametric boundary elements [5–7] were exploited.

3. Structural modelling of SEA-OWC-Clam

The SEA-OWC-Clam is a regular dodecagon with an external side length of 20.710 m and an internal side length of 16.424 m; from a geometrical point of view the device can be inscribed in a circle of diameter $D_0 = 80.02$ m. The dodecagon has vertical sides of height 8 m and a draught of 6 m [1]. For a structural integrity assessment, the device is partitioned into representative structural elements for the subsequent FEA. The ‘essential’ structural members of the transverse cross section of the closed and open idealised forms are simultaneously illustrated in Figure 2a with dimensions specified in Figure 2b. Within Figure 2a the still water level (SWL) is indicated on the closed and open transverse cross sections.

Unlike the hydrodynamic analysis, the whole structure has been explicitly defined in the FEA structural model since the loading in each quadrant is not necessarily symmetric and exploitation of planes of geometric symmetry invoking symmetric and anti-symmetric boundary conditions would have led to further unnecessary complexity to the set of performed simulations at different frequencies.

The SEA-OWC-Clam consists primarily of thin plates of thickness t . Here t is relatively small compared to finite element length, L , and width, W . Hence, a shell element is used to model each SEA-OWC-Clam structure. Of the different possible ABAQUS[®] shell elements available is the four node ABAQUS[®] S4 element [8]; hereafter foreshortened to S4.

This element can accommodate either thin shell or thick shell theory. In the former case the Kirchhoff [9] assumption of negligible transverse shear deformation applies, whereas a thick shell deforms more generally in accordance with Mindlin-Reissner behaviour [9]. For the length to thickness ratios used in modelling the device the thin shell theory is automatically selected by the software since $L/t \gg 15$. This means that the normal sections to the middle (neutral) plane of the undeformed plate remain straight and normal during the deformation. Upon assuming the 1-2 plane is the middle plane of the shell with the third axis perpendicular to this plane, thin shell theory implies that the transverse shear strains ε_{13} & ε_{23} vanish, but not necessarily the transverse shear stress components σ_{13} & σ_{23} . The only component of stress that is neglected is σ_{33} .

Fig. 2a Structural modelling of SEA-OWC-Clam module.

Fig. 2b Plan view dimensions of the SEA-OWC-Clam module.

The structural steel grade S355, with a yield stress of 355MPa, was considered as the most practical construction material for the preliminary design analysis undertaken. S355 Grade Steel has a density of $\rho_{Steel} = 7850 \text{ kg/m}^3$, a Young's modulus of $E = 213 \text{ GPa}$ and a Poisson ratio of $\nu = 0.3$. These material properties are applied together with a nominal

thickness of 0.015 m. The artificial closure plates in the closed form of the device will have their thickness varied as discussed in Section 4.1.1.

In the closed and open versions of the device there are 34392 & 40104 S4 quadrilateral shell elements as illustrated in Figures 3a & 3b respectively.

Fig. 3a 3D view of FEM mesh for closed form of device.

Fig. 3b 3D view of FEM mesh for open form of device.

3.1. Finite element analysis approach

The fundamental assumptions of the structural analysis are:

- All contributing loads in a particular analysis are harmonic with the same circular frequency ω , but of different phasing relative to the incident wave.
- All nonlinearities are neglected. Thus the material is modelled as homogenous and isotropic and subject to linear elastic behaviour.
- Transient effects are not considered.

The analysis is repeated for each incident wave frequency set out in Table 1 of [1].

The first assumption means that incident, diffraction and radiation pressures and the known different rigid body motions are determined at the same incident wave frequency.

The second assumption is met in so far as the motion responses are linearly modelled; additional viscous damping introduced into the rotational degrees-of-freedom is essentially linear. The structural material behaviour is linear elastic.

Transient influences will not naturally vanish if structural damping is neglected. This is not detrimental to the investigation, since within the FEA approach one can readily identify the required steady state solution. Hence the assumption of zero structural damping is not an issue. However, the response of an undamped structure can become unbounded if the forcing function frequency corresponds to an eigen-frequency. Since incident wave frequencies do not exceed 0.3 Hz, the forcing frequencies are below the structural resonance frequencies. A modal analysis has been undertaken for the ‘closed’ structure to demonstrate that this is the case. The first mode frequency is of the order of 1.5Hz.

The harmonic nature of the incident waves and the consequential motion responses lead to structural loads presented as a complex dynamic pressure distribution over the structure. The real and imaginary pressure components represent the in-phase and out-of phase loadings with respect to the incident wave frequency.

The SEA-OWC-Clam structure is modelled as a linear system, with $\mathbf{M}$ denoting the mass matrix, $\mathbf{C}$ the structural damping matrix, $\mathbf{K}$ the stiffness matrix and $\bar{\mathbf{F}}$ the complex external forces acting on the structure. Since the external forces are the consequence of modelling the dynamic response of the structure to waves with hydrostatic restoration (buoyancy) included [1], the stiffness parameter $\mathbf{K}$ used here, represents the structural stiffness. Similarly the device dynamic motion response analysis used to generate the structural loads included radiation wave generated fluid damping effects.

The structural dynamic response can be generally expressed as:

$$\mathbf{M}\ddot{\mathbf{U}} + \mathbf{C}\dot{\mathbf{U}} + \mathbf{K}\mathbf{U} = \bar{\mathbf{F}}(t). \quad (1)$$

Clearly $\ddot{\bar{\mathbf{U}}}$, $\dot{\bar{\mathbf{U}}}$, $\bar{\mathbf{U}}$ are the complex acceleration, velocity and displacement vectors respectively. Knowing the external force is harmonic and expressible as $\bar{\mathbf{F}}(t) = F_0 \exp(-i\omega t) \equiv \bar{\mathbf{F}}(\omega)$ the undamped form of Equation (1) is:

$$\mathbf{M}\ddot{\bar{\mathbf{U}}} + \mathbf{K}\bar{\mathbf{U}} = \bar{\mathbf{F}}(\omega). \quad (2)$$

The harmonic nature of excitation and motion response [10] means Equation (2) can be recast as:

$$[-\omega^2 \mathbf{M} + \mathbf{K}]\bar{\mathbf{U}} = \bar{\mathbf{F}}, \quad (3)$$

for a circular incident wave frequency, ω . Solution of Equation (3) via the steady-state direct method is preferred to an eigen-frequency extraction analysis followed by a modal superposition method.

3.2. The complex variable nature of the structural responses

Equation (3) provides a complex displacement from which a complex strain (derivative of displacement) and complex stress (related to the strain by constitutive law) can be retrieved. Consequently the interpretation of the generated results is not as straight-forward as that of a static analysis or a transient dynamic analysis.

To interpret the significance of the stress levels, arising from the dynamic loads applied as a consequence of the observed rigid body modes, it is physically more meaningful to think in terms of the ‘effective stress’ σ_v , as defined in the von Mises criterion [11]. This effective stress can be compared against the material yielding stress. For a multi-axial loading condition the general state of stress is three-dimensional and therefore σ_v according to von Mises criterion is determined using:

$$\sigma_v = \sqrt{\frac{(\sigma_{11} - \sigma_{22})^2 + (\sigma_{22} - \sigma_{33})^2 + (\sigma_{11} - \sigma_{33})^2 + 6(\sigma_{12}^2 + \sigma_{23}^2 + \sigma_{31}^2)}{2}}. \quad (4)$$

The ‘effective’ stress is essentially a scalar representation of the recognised 3D state. The von Mises stress is a derived quantity which does not recognise the phase characteristics of the different stress components. Hence it may be considered to provide a conservative engineering estimate of the stress. This applies to both the real and imaginary components.

Selection of the S4 thin shell element implies that the Kirchhoff assumption [9] is applicable, and hence:

$$\sigma_v = \sqrt{\frac{(\sigma_{11} - \sigma_{22})^2 + (\sigma_{22})^2 + (\sigma_{11})^2 + 6(\sigma_{12}^2 + \sigma_{23}^2 + \sigma_{31}^2)}{2}}. \quad (5)$$

It is important to note that the effective stress is essentially a modulus of the stress level at any integration or Gaussian point inside each element. A subsequent extrapolation of the known stress at the element integration points permits stress evaluation at each element node.

The overall magnitude of von Mises stress is calculated according to Equation (6), where the real $|\sigma_v^{\text{Re}}|$ and imaginary $|\sigma_v^{\text{Im}}|$ stress components at a specific integration point are combined, viz:

$$|\sigma_v| = \sqrt{|\sigma_v^{\text{Re}}|^2 + |\sigma_v^{\text{Im}}|^2}. \quad (6)$$

In Section 4.1 the illustrated level of absolute real and imaginary von Mises stress will indicate that these can occur in the same physical location, such as the inner faces of the device (see Figures 7a & b) and not occur on the outer faces of the device facing the incident wave.

4. Presentation and discussion of predicted stress levels

For the closed form of the device variation of thickness at those locations corresponding to the OWC openings is a simple attempt to investigate how stress in the remaining structure may be influenced by the existence of such openings (thickness tends to zero). Having investigated this aspect the open structure is investigated before discussing differences in open and closed device stress levels. The earlier idea that the resulting stress is a function of reinforcement and cancellation of different influences, due to the relative phasing with respect to the incident wave, is demonstrated to provide some insights.

4.1. Stress characteristics of closed form of device

In later detailed design of the device the openings leading to the J and N oscillating water columns would properly be stiffened to reduce flexure of side and bottom structure. From a structural point of view the opening might be constructed as suggested in Figure 4.

Fig. 4 Schematic representation of tubular reinforcement of OWC side entrance.

However, including such detail in the hydrodynamics would lead to an unnecessarily large matrix formulation to solve; because the boundary elements modelling the thickness of the stiffeners would necessitate excessively refined discretization of the remaining structure to avoid ill-conditioning. This phenomenon is often induced by having boundary elements of excessive aspect ratio and, or, excessive area ratios [12].

Here we assume that plates of differing thickness would be sufficient to provide an indication of likely stress concentrations around the OWC entrances. This approach provides some justifications for addressing the ‘closed’ form of the device.

The maximum stress levels $|\sigma_v^{\text{Re}}|_{\text{max}}$ & $|\sigma_v^{\text{Im}}|_{\text{max}}$ are plotted separately as a function of wave frequency for the closed version of the device in Figure 5. The observed peak pitch influence of Figure 24 of [1] is reflected in the real (in-phase) stress component of Figure 5.

Many more frequencies have been investigated here than for ship forms or other wave energy devices. This is to try and ensure no unusual response was omitted. The omitted frequencies in the stress plots, with respect to the hydrodynamic analysis [1], provide no real additional information about the stress levels that might be encountered.

Fig. 5 $|\sigma_v^{\text{Re}}|_{\text{max}}$ and $|\sigma_v^{\text{Im}}|_{\text{max}}$ for closed SEA-OWC-Clam versus wave frequency.

In Figure 6 the quantity $|\sigma_v|$, defined in Equation (6) is plotted as function of frequency.

Fig. 6 $|\sigma_v|$ for closed SEA-OWC-Clam versus wave frequency.

Clearly the dominant real peak in Figure 5 occurs at the same frequency as the dominant peak in Figure 6. The frequency of 1.055 rad/s is higher than the ‘closed’ heave peak motion (1.00 rad/s) of Figure 23 of [1], but equals the second peak pitch motion of Figure 24 of [1].

The value of calculated stress is relatively constant over the low wave frequency range below 0.6 rad/s. This form of wave physically corresponds to swell. Above $\omega = 1.055$ rad/s the stress reduces to a new slightly higher ambient level.

Figures 7a & b for the closed form of structure indicate that the location of the maximum magnitude of the real part and the maximum magnitude of the imaginary part of the equivalent von Mises' stress components are not located at the same part of the structure for the selected incident wave frequency of 1.055rad./s.

Fig. 7a $|\sigma_v^{\text{Re}}|$ for closed structure with $\omega = 1.055 \text{ rad/s}$.

Fig. 7b $|\sigma_v^{\text{Im}}|$ for closed structure with $\omega = 1.055$ rad/s.

4.1.1. Influence of closed plate thickness

In Figure 8 the thickness of the closure plate is systematically decreased. Specific thicknesses considered are: 15, 12, 8 & 4 mm. The case of 1mm was evaluated and then rejected as unhelpful.

Reducing the thickness of the 'closure' plate from 15 mm (nominal thickness of all the other structural plates) to 4 mm indicates that the stress levels increase. The three frequencies addressed in Figure 8 correspond to:

- Peak frequency associated with the March South Uist spectral data ($\omega = 0.465$ rad/s);
- Wave frequency associated with the highest stress level predicted ($\omega = 1.055$ rad/s);
- Incident wave frequency exhibiting the higher ambient stress level of Figure 6 ($\omega = 1.375$ rad/s).

Fig. 8 $|\sigma_v|$ for closed SEA-OWC-Clam versus closure plate thickness.

The peak stress in Figure 5 is associated with a particular wave frequency (1.055 rad/s) and this essentially corresponds to a plate thickness of 7mm for the peak energy related frequency of 0.465 rad/s in Figure 8. In Figure 8 the stress levels are particularly high for the thin artificial closures of the OWC entrances. Hence some structural redesign is required and this is to be addressed in Section 5 having appreciated the ambient device stress levels and their distribution in Section 4.

The localization of the higher stress is in the side and bottom plates nearest to the closure plates; this is due to the fact that the fluid loads are now exerted on a thin plate experiencing large deformations due to a lack of plate stiffeners. Further appreciation of this point is realisable from the concentration of high stress values in the side and bottom of the device respectively in Figures 7a & b.

4.2. Stress characteristics of open form of device

For the open structure the stress experienced is not dependent on the ‘closure’ panel concept. Lower levels of stress are obtained for the open structure confirming that allowing fluid motions within the structure should help in diminishing the transfer of energy from the wave to the structure, ideally justifying the concept of a wave energy converter.

Figures 9 and 10 are the open structure results corresponding to Figures 5 and 6.

Fig. 9 $|\sigma_v^{\text{Re}}|_{\text{max}}$ and $|\sigma_v^{\text{Im}}|_{\text{max}}$ for open SEA-OWC-Clam versus wave frequency.

Fig. 10 $|\sigma_v|$ for open SEA-OWC-Clam versus wave frequency.

From Figures 5 and 9 one can deduce that the phases of the stress for closed and open will be quite distinct.

Figures 11a & b are the open form of Figures 7a & b for the same incident wave frequency of $\omega = 1.055$ rad/s. The maximum magnitudes of real and imaginary parts of the equivalent von Mises stress occurs in the same geometric regions.

Fig. 11a $|\sigma_v^{\text{Re}}|$ for open structure with $\omega = 1.055$ rad/s.

Fig. 11b $|\sigma_v^{\text{Im}}|$ for open structure with $\omega = 1.055$ rad/s.

4.3. Closed versus open structure stress levels

Differences of stress phases were noted in previous section. Here Figures 6 and 7 are superimposed in Figure 12 to demonstrate the level of increased $|\sigma_v|$ in the closed structure.

Fig. 12 Comparison of $|\sigma_v|$ for closed & open SEA-OWC-Clam versus wave frequency.

The striking peak value of Figure 12 is readily identified with the frequency of 1.055 rad/s. The open form of SEA-OWC-Clam generally has stress levels lower than those associated with the closed form of SEA-OWC-Clam. The one exception is the frequency corresponding to the peak pitch response of the closed form SEA-OWC-Clam. In this case the open form experiences a maximum stress level 52% higher than the closed geometry for the same incident wave frequency. This higher stress in the open form of the SEA-OWC-Clam geometry does not correspond to a resonant structural frequency. This is readily demonstrated by applying the same load distribution with a different (incorrect) frequency within the FEA, which yields the same level of response. That is, the high stress is due to a high level of hydrodynamic loading. Such high loads must result from the phasing of the contributions of incident, diffraction and radiation waves; in this case there is more reinforcement than cancellation of these contributing loads.

According to Figures 11 to 13 of [1] the open & closed structure are experiencing the same level of total excitation forces and moments; however the wetted surface for the open form of

the SEA-OWC-Clam is almost double that of the closed form and hence implies a lower average pressure acting on the open structure.

4.4. Appreciating stress cancellation and reinforcement within SEA-OWC-Clam

Having presented and compared different forms of stress amplitude at specific limited wave frequencies the authors are aware of the possibility that for any specific observation made, or conclusion drawn, a counter example might exist. This could be provided by considering either a different frequency, or, the structure other than the current structure being addressed.

In this section we therefore look at the actual contributions to the stress by different influences.

The wave excitation consists of the influence of both the incident waves and the diffracted waves. The radiation and diffraction velocity potentials are solved using the MATTHEW suite. The incident wave has a closed analytic form and through the linearized Bernoulli equation provides the so-called Froude Krylov wave load. For the SEA-OWC-Clam structure diffraction is significant. The Froude–Krylov and diffraction loads are of comparable magnitude.

Being mindful of the possibility of the cancellation and reinforcement of different effects the total von Mises equivalent stress is compared with the situation of having only a response motion in isolation or only the wave excitation loads. Essentially one either sets the wave amplitude ‘a’ to zero or sets the real and imaginary motion responses to be omitted to zero. These indicated calculations are undertaken for both geometries, but for different frequencies. The frequencies selected in each case correspond to: peak stress at $\omega = 1.055$ rad/s, a peak pitch at $\omega = 0.75$ rad/s, and a closed and open peak heave response for $\omega = 1.00$ rad/s & 0.95 rad/s respectively.

Each case is illustrated in Figures 13 to 18.

Fig. 13 Comparison of total stress, single motion contributions and combined incident and diffracted wave influences for closed SEA-OWC-Clam for peak stress at $\omega = 1.055$ rad/s.

Fig. 14 Comparison of total stress, single motion contributions and combined incident and diffracted wave influences for open SEA-OWC-Clam for peak stress at $\omega = 1.055$ rad/s.

Figures 13 & 14 indicate that the incident and diffraction based pressures are the major contribution to the observed von Mises stress. The contributions of the surge, heave and the pitch motions are small compared to the wave contribution.

Fig. 15 Comparison of total stress, single motion contributions and combined incident and diffracted wave influences for closed SEA-OWC-Clam for peak pitch at $\omega = 0.75$ rad/s.

Fig. 16 Comparison of total stress, single motion contributions and combined incident and diffracted wave influences for open SEA-OWC-Clam for peak pitch at $\omega = 0.75$ rad/s.

For both open and closed forms of the device Figures 15 & 16 demonstrate incident and diffracted waves significantly influence the total stress. The pitch motion contribution dominates surge and heave motion influences. The pitch motion might be considered to reduce the influence of the excitation waves for the closed form, but appears to increase the resultant stress levels for the open form of the device.

Fig. 17 Comparison of total stress, single motion contributions and combined incident and diffracted wave influences for closed SEA-OWC-Clam for peak heave at $\omega = 1.00$ rad/s.

Fig. 18 Comparison of total stress, single motion contributions and combined incident and diffracted wave influences for open SEA-OWC-Clam for peak heave at $\omega = 0.95$ rad/s.

Figures 17 & 18 are significantly different in terms of the contributions from the difference sources. For the closed form the wave contribution dominates the total stress level (Figure 17). Contributions from the motion are marginally less than each other as one considers in turn the surge, heave and pitch contributions. For the open device of Figure 18 surge is dominant and it would appear that the different influences contribute in a cancelling manner to produce a smaller total stress. In fact the differences are so extreme that the analysis required to produce these figures was double checked.

Figures 13–18 provide an indication of the different phase characteristics of the contributing fluid borne loads.

The total stress level presented in Figures 13-14 exceeds the yields stress of steel S355 specified in Section 3. Therefore to determine the absolute maximum stress, the stress distributions for all wave frequencies were thoroughly searched as discussed next.

5. Extreme stress identification and structural integrity improvement

The external plate stress distributions illustrated in Figures 7a & b and 11a & b, for a wave frequency of 1.055rad/s, provide the highest stress levels for both open & closed structural forms. However, the structure is of greater complexity that the external annulus forms. In searching for high stress levels throughout the structure we recognised that there are different structural arrangements. Each must be considered to identify a possible extreme stress.

5.1. Localised high-stress identification

Since in general the stress levels are larger for the closed structure, we will consider the range of stress levels with varying wave frequencies for the closed structure. The locations designated A to K in Figure 19 have the associated structural arrangements illustrated in Figure 20. The three dimensional distribution of the different structural arrangements is provided in Figure 21.

Fig. 19 FEA node points with distinct structural arrangements.

Fig. 20 Variation of actual structural arrangements.

Fig. 21 Relevant locations of structural arrangement

Stress levels at all points were determined and evaluated. At the lower wave frequencies the maximum von Mises stress is almost uniformly 500MPa, at a frequency of 1.055rad/s stress levels are 1500-1600MPa (despite this frequency not being either a motion or structural resonant frequency). For higher wave frequencies stress levels are almost uniformly 1000MPa. This is for unit wave amplitude. Assuming maximum wave heights are likely to be twice the March and November significant wave heights of 4.8m and 3.5m, associated with the spectra of Figure 4 of [1], the pessimistic working wave amplitude could be 5m.

Prior to suggesting any design changes it is necessary to identify an acceptable stress threshold.

5.2. Appreciation of required stress level reduction

Whilst processes exist to estimate safety factors [11] the assignment of the rating factor influences is not clear for this conceptual design. Therefore given the novelty of the device, improved design of structural plating is to be based on not exceeding 50% of the yield stress of the selected steel. Hence the maximum magnitude of stress for 1.0m wave amplitude needs to be less than 35MPa. This implies a reduction factor between 15 & 45 of the predicted dynamic stress levels. Since the yield stress is readily exceeded, different structural detailing

is required: changes in plate thickness, introduction of plate stiffening or an alternative form of plating is necessary for the device to survive.

If a higher working stress is required the marine grade steel AH40 [13], with a yield stress of 390MPa, would lead to a working stress of the order of 39MPa per unit wave amplitude.

5.3. Consequences of structural changes

Any structural mass increase necessitates a corresponding change in water ballast for the structure to float at the currently specified 6m draught [1]. In the closed and open forms examined, the internal ‘structural space’ required for water ballast was 87.85% & 80.34% of total displaced volume. Further extensions of the internal flow regime to fulfil the intended J-tube and N-tube openings will reduce both natural buoyancy and required water ballast. Thus hydrostatic stiffness and pitch inertia characteristics will change. Thus the resulting stress distribution will be different.

5.4. Identification of possible advantageous structural modifications

Simply changing the structural details and repeating the structural analysis presented in this paper and the hydrostatic, hydrodynamic and motion analysis in Part A, until stress levels become satisfactory is not a good strategy. Instead we shall consider variations of different local structural detailing and use the external fluid loading associated with the closed form to predict stress levels in order to identify structural arrangements exhibiting the required lower stress levels per unit wave amplitude.

The two exploratory structural forms to be examined are a conventionally stiffened plate and the more radical corrugated plate. A stiffened flat plate is preferred hydrodynamically as it would be possible to maintain the smooth J & N tube walls. Ignoring hydrodynamic convenience, a corrugated structure would more readily provide greater strength. The wave frequency considered initially is 0.465 rad/s (0.074Hz) as this corresponds to the highest peak energy level in the South Uist spectral data of Figure 4 of [1].

5.4.1. Flat stiffened plate

We consider a vertical plate of dimensions equal to the outer flat side of the SEA-OWC-Clam. The plate is 6.0m high in accordance with device draught and 20.71m wide as per Figure 2b. Internally the plate is divided by the two anti-sloshing plates perpendicular to the plate at locations B and C and the vertical and horizontal I-beam stiffeners of Figure 22. The I-beam web and flanges are 0.1m with an associated thickness of 0.01m.

A slight variation of this stiffened plate corresponds to removal of all horizontal stiffeners.

The plate is fully constrained (clamped) at all the edges and simply supported at the location of the two anti-sloshing plates. The structural load corresponds to the hydrodynamic load applied to the plate facing the incident wave on the device outer wall.

The highly magnified deformation and the corresponding stress, von Mises stress (σ_v) and the three in-plane stress components $(\sigma_{11}, \sigma_{22} \text{ \& } \sigma_{12})$, are provided in Figures 23 a–h.

Fig. 22 Sketch of stiffened plate with internal anti-sloshing plates (not to scale).

Fig. 23a Magnified deflection (by 50) for von Mises stress levels $|\sigma_v|$ for fully stiffened plate.

Fig. 23b Magnified deflection (by 50) for stress component σ_{11} for fully stiffened plate.

Fig. 23c Magnified deflection (by 50) for stress component σ_{22} for fully stiffened plate.

Fig. 23d Magnified deflection (by 50) for stress component σ_{12} for fully stiffened plate.

Fig. 23e Magnified deflection (by 50) for von Mises stress levels $|\sigma_v|$ for vertical stiffened plate.

Fig. 23f Magnified deflection (by 50) for stress component σ_{11} for vertical stiffened plate.

Fig. 23g Magnified deflection (by 50) for stress component σ_{22} for vertical stiffened plate.

Fig. 23h Magnified deflection (by 50) for stress component σ_{12} for vertical stiffened plate.

Figures 23a–d indicate, as expected that the peak value of the in-plane stress component σ_{12} is an order of magnitude smaller than the other stress distributions presented. The value of $|\sigma_v|$ is comparable with σ_{22} with σ_{11} being slightly smaller. This observation is confirmed in Figures 23e–h. However, the stress levels for the vertical stiffened plate are higher than the grid reinforced plate.

The actual stress in each case is approximately six times higher than the target stress of 35 MPa. This justifies the need to consider an alternative structural configuration that can accommodate the requested level of stress with ideally a reduced structural mass. At this

stage of structural analysis plate buckling is not an issue since the predicted stress levels are unacceptable.

5.4.2. Trapezoidal corrugated plate

Addressing a corrugated structure means ignoring the hydrodynamic convenience of a flat surface. The corrugated plate may be described as a succession of waves characterized by a sloped section with angle ' θ ', length ' a ' & depth ' d ' and a horizontal portion of length ' b ', as per Figure 24. The non-dimensional ratio k corresponds to b/a .

Fig. 24 A wave with parameter $k=2$.

The different shape parameters require investigation to identify optimal stress reduction. As an intermediate study a plate one third the width of the outer module width is addressed, thus the plate length (L) is 6.903 m, the height (H) is 6 m and thickness remains at 0.015 m.

Since this intermediate study seeks a plausible corrugated form, a uniform load of 5000 N/m² is applied. This is of sufficient magnitude, in this parametric study, to represent the known dynamic loads acting on the device. The mass of the corrugated plate for a thickness t is $M = 2\rho NtH(b + \sqrt{a^2 + d^2})$ with $a = L/(2N(1 + k))$.

The plate is fully clamped on all edges and the number of waves N per selected width, parameter k and depth d are investigated. Structural response with values of ' a ', ' b ' and angle ' θ ' follow from continued use of ABAQUS[®] shell element S4. In total 3 values of N and 8 different values of d were investigated for $k = 2$ to yield the sensitivities of Table 1a and Figure 25.

Table 1a von Mises stress level as a function of depth, d , and number of waves, N .

N, k values		$d=0.01\text{m}$	$d=0.02\text{m}$	$d=0.04\text{m}$	$d=0.05\text{m}$	$d=0.07\text{m}$	$d=0.10\text{m}$	$d=0.50\text{m}$
5, 2	Stress [MPa]	246.5	175.4	92.1	72.2	49.3	33.6	5.0
10, 2	Stress [MPa]	257.6	177.7	89.8	69.7	39.4	30.4	5.1
20, 2	Stress [MPa]	255.6	168.9	80.0	60.5	39.7	24.5	3.3

Fig. 25 Graph of the stress level as a function of the depth for different numbers of waves.

Clearly N is not a decisive parameter upon inspection of Figure 25. However, if we address additionally the details of Table 1b, it appears that there is no need to increase N beyond 10 and d values of 0.10m and 0.07m should be further addressed, as mass significantly increases if further reduction in maximum deflection, u , is sought. Hence a compromise between acceptable response and mass is required. Slope is dictated by depth d and k . Having identified an acceptable range of d , the influence of k is next investigated for $d = 0.07\text{m}$ and reported in Table 2.

Table 1b von Mises stress level, total mass and maximum deflection as a function of d & N .

$N=10, k=2$	$d=0.01\text{m}$	$d=0.02\text{m}$	$d=0.04\text{m}$	$d=0.05\text{m}$	$d=0.07\text{m}$	$d=0.10\text{m}$	$d=0.50\text{m}$
Stress [MPa]	257.6	177.7	89.8	69.7	39.4	30.4	5.1
Mass [kg]	4883	4901	4972	5024	5154	5405	10501
u [mm]	108	53.4	16.7	10.9	5.49	2.69	0.09

Table 2 von Mises stress level, total mass and maximum deflection as a function of ratio k

$N=10, d=0.07$	$k=1$	$k=2$	$k=3$	$k=4$	$k=5$	$k=6$	$k=7$	$k=8$	$k \sim \infty$
Stress [MPa]	50.8	45.0	43.3	41.7	40.7	40.0	39.8	39.4	36.1
Mass [kg]	5070	5154	5228	5291	5344	5390	5429	5463	5866
u [mm]	6.49	5.54	5.16	4.95	4.81	4.72	4.65	4.59	4.17

Table 2 suggests that $k = 4$ is a possible compromise regarding stress reduction and mass increase (relative to equivalent unstiffened plate mass of 4877 kg) and the uniform load selected. Finally for selected N , d and k Table 3 presents influence of plate thickness, t .

Table 3 von Mises stress level and total mass as function of t for $N = 10$, $d = 0.07$ m & $k = 4$

t [m]	0.0050	0.0100	0.0125	0.0150	0.0175	0.0200	0.0250	0.0300
Stress [MPa]	113.7	59.8	48.9	41.7	36.5	32.6	27.1	23.2
Mass [kg]	1764	3527	4409	5291	6173	7054	8818	10581

Table 3 suggests that a thickness of 0.0175 m is a reasonable compromise for the trial uniform load.

5.4.2.1. Conclusion for the corrugated plate

We can conclude that for a uniform pressure load of 5000 N/m^2 , the corrugated plate parameters for $N=10$ waves of depth $d = 0.07\text{m}$ provide stress reduction with lightest structural weight for a $k = b/a$ ratio of 4 with a plate thickness of $t = 0.0175\text{m}$.

The next step is to apply the determined load acting on the SEA-OWC-Clam, when excited by the peak energy frequency $\omega = 0.465 \text{ rad/s}$, to the corrugated representation of an outer wave facing side of the device. In this case both potential values of d identified earlier will be considered. However, only an outer vertical face of a module, now modelled as a corrugated structure, is addressed as illustrated in Figure 26. Each of the 90 boundary element pressures is applied as a uniform pressure over that section of the corrugated plate coincident with the covering boundary element. The 1:4 ratio of boundary elements to finite elements used when analysing the complete device, illustrated in Figure 1, is not applied here. This is because the pressure distribution determined from the analysis of the complete structure is to be used to examine the proposed corrugated outer wall. The number of finite elements must be

sufficiently fine to model the corrugated nature of the structure. The number of finite elements over the draught of 6 m is uniform and influenced by the depth of the corrugation (parameter d of Figure 24). Hence the ratios of 1:96 and 1:108 were used in accordance with the parameter d . In Figure 26 the ratio 1:96 is consistently applied.

Fig. 26 Mesh for corrugated panel form of outer vertical wall.

The von Mises stress is required to be lower than 35 MPa. Therefore, the depth d and plate thickness t are now modified to fulfil this requirement. From the flat plate and corrugated plate parametric studies completed, the maximum stress properties are provided in Table 4.

Table 4 Stress reduction levels for different structural configurations and pressure loadings for peak energy frequency of $\omega = 0.465$ rad/s.

Plate Type	d [m]	t [m]	Stress [MPa]	Mass [kg]	Percentage changes in mass
Original Flat Plate	N/A	0.015	406.1	14632	0.00
Flat Plate with Vertical stiffeners	N/A	0.010	247.6	11450	-21.75%
Flat Plate with Grillage	N/A	0.010	200.0	12734	-12.97%
Corrugated Plate (A1)	0.07	0.015	65.0	15872	+8.47%
Corrugated Plate (A2)	0.07	0.0175	57.1	18517	+26.55%
Corrugated Plate (B1)	0.10	0.015	42.2	16856	+15.20%
Corrugated Plate (B2)	0.10	0.0175	36.9	19665	+34.40%
Corrugated Plate (B3)	0.10	0.020	33.0	22474	+53.60%
Corrugated Plate (C)	0.15	0.015	26.0	18704	+27.83%

From Table 4, we can draw the conclusion that the corrugated panel is a more effective solution than a flat plate with grillage based stiffening. For the SEA-OWC-Clam subsection analysed the required stress level is achievable for the wave frequency addressed, see Figures 27. Given maximum stress increases with wave frequency (Section 5.1) further refinement of corrugated plate C is required to reduce again the maximum stress level without an exorbitant mass increase.

Fig. 27a Magnified deflection (by 50) for von Mises stress levels $|\sigma_v|$ for corrugated plate C.

Fig. 27b Magnified deflection (by 50) for stress component σ_{11} for corrugated plate C.

Fig. 27c Magnified deflection (by 50) for stress component σ_{22} for corrugated plate C.

Fig. 27d Magnified deflection (by 50) for stress component σ_{12} for corrugated plate C.

6. Closure

These companion papers address two distinct aspects of analysing any floating structure. Part A details the historical development of the Clam device and how it is changed as a more viable practical form of the device has been sought. Part A also indicates how to undertake the hydrodynamic analysis to appreciate the hydrodynamic and motion response behaviour of both structure and each oscillating water column. However, before implementing such as analysis there is a need to rethink the structural detailing of the device.

In the current paper (Part B) the stress levels in the open form of SEA-OWC-Clam have been shown to be less than those associated with the closed form of the device. Operation of the intended J- and N- form of OWC would reduce further the stress experienced.

Limited initial structural design details of the device have necessitated the reported structural integrity analysis to appreciate the magnitude of the maximum stress likely to be encountered in a sea state consistent with that of South Uist.

In the reporting of the stress level we have endeavoured to give an indication of how the von Mises stress magnitude is affected by increasing wave frequency and to provide insights regarding the reinforcement and cancellation of stress contributions from the differing hydrodynamic and response related quantities. This approach provides a more informed appreciation of the difference in behaviour of the open and closed form of the device analysed.

The device as presented does not exhibit structural integrity. Therefore we have sought to seek possible structural changes that might lead to survival in a proposed wave energy extraction climate such as South Uist. Simple stiffening of flat plate does not provide a very adequate structural solution. Use of a corrugated outer panelling can significantly reduce stress (by a factor of 20 on outer surface of device) for the South Uist peak frequency of 0.465 rad/s. Hydrodynamicists would prefer a flat plate on the outer surface of the corrugated structure, but this would ultimately involve the use of an advanced construction technique such as laser welding.

Whilst curved plates have been considered they have not been reported here, as this necessitates further detailed investigation of the possibility of snap-through behaviour. In the structural analysis of the module, plate characteristics addressed identification of set target stress levels. Since the alternative structural models only address part of the device, buckling has not been considered.

As a final thought, rather than provide the device strength through its outer structure, a carefully designed space frame with panel based closure might meet both the preferred simplistic geometry for hydrodynamic analysis and the structural integrity sought.

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