Computational Model of Aquaculture
Fin-Fish Net-Pens

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Introduction

This work is sponsored by Sea Grant, the School of Marine Sciences at the University of Maine and Defense Advanced Research Project Agency (DARPA) and was completed at the University of Maine, Orono, Maine from September 1996 to May 1997. This work is an outgrowth of work done by Messier and Thompson [1] for DARPA in modeling Very Large Floating Structures (VLFS). The dynamic structural response of a VLFS and typical floating net-pen designs are similar, though of vastly different scales. With a few notable exceptions, net-pens are typically made up of structural beam elements assembled to form a buoyant frame that supports the net pen.

In the last 10 years aquaculture of salmonids has expanded from 5.7% (1985) to 34.5% (1994) share of worldwide production. Originally developed in the Norwegian fjords, the industry has expanded to protected and exposed locations in Canada, Ireland, Peru, and the United States to name a few major salmon producers. With the collapse of many traditional fisheries worldwide, market forces are pressing development of the aquaculture of market species such as cod, fluke, and halibut. Recent developments in the New England area show promising startup projects of these species with the backing of both public and private capital. Similar projects are occurring elsewhere in the world. Due to environmental, regulatory, and user conflict constraints, future expansion of fish farming will be in more exposed, high energy locations.
This shift requires improved analysis of net-pens to determine their performance and survivability in a high energy locations.

To provide an additional tool to evaluate the performance of net-pens, computational models using Finite Element Analysis (FEA) method were developed and applied to net-pen designs used in salt water aquaculture. A commercial FEA software package, ABAQUS AQUA™, was used in developing these models. The objective of this research is to identify failure modes and predict estimated life cycles of net-pen designs in different ocean environments. Successful application of this analysis will provide aquaculture managers, operators and regulators with increased understanding of the performance and survivability of a particular net-pen design under the applied sea state. This paper describes the development and application of this tool on two different net-pen designs.

Theoretical Considerations

The theoretically important aspects of this research are discussed below. They include: nonlinear dynamic finite element method, Airy wave theory and its application through Morison’s equation, and the mapping technique used to model the containment nets.

Nonlinear Dynamic Finite Element Method

When studying the dynamic response of a structure using the finite element method, a determination of the appropriate solution algorithm, implicit or explicit, is required. Belytschko [2] suggests that the problem be classified as either an inertia or wave propagation problem. Wave propagation problems require an accurate reproduction of the wave front (e.g. the response of an impact) and are best solved using an explicit time integration scheme. Inertia problems, or structural dynamic problems, are low frequency response problems such as large displacement. These are best solved using an implicit time integration scheme. Clearly the structural response of net-pen in an ocean environment is of an inertia type requiring an implicit solution technique. Due to the large deformations of the netting and
relatively large wave heights to be modeled, nonlinear, non-symmetrical analysis was selected. This allows the stiffness matrix to be reconstructed at each time step to account for the geometric changes in the structure. The finite element solution algorithm uses a modified Newmark family of equations as the basis of the implicit nonlinear solution algorithm. The nodal equations are:

\[ M^{NM} \dot{u} |_{t+\Delta t} + (1+\alpha)(I^N |_{t+\Delta t} - P^N |_{t+\Delta t}) - \alpha(I^N |_t - P^N |_t) + L^N |_{t+\Delta t} = 0 \]  
[Eq. 1]

\[ \ddot{u} |_{t+\Delta t} = \dot{u} |_t + \Delta t \left[ \begin{array}{c} \gamma \dot{u} |_t + \lambda \ddot{u} |_{t+\Delta t} \end{array} \right] \]  
[Eq. 2]

\[ u |_{t+\Delta t} = u |_t + \Delta t \ddot{u} |_t + \frac{\Delta t^2}{2} \left[ \begin{array}{c} \beta \dot{u} |_t + \beta \ddot{u} |_{t+\Delta t} \end{array} \right] \]  
[Eq. 3]

where: \(-\frac{1}{3} \leq \alpha \leq 0; \ \beta = \frac{(1-\alpha)^2}{4}; \ \gamma = \frac{1}{2} - \alpha; \)

The consistent mass matrix is defined as:

\[ M^{NM} = \int_{V_o} \rho_o N^N \cdot N^M dV_o \]  
(2.1.4)  
[Eq. 4]

The internal force vector is defined as:

\[ I^N = \int_{V_o} \beta^N : \sigma dV_o \]  
(note \(\beta^N \neq \beta\))  
[Eq. 5]

The external force vector is defined as:

\[ P^N = \int_{S} N^N \cdot t dS + \int_{V} N^N \cdot F dV \]  
[Eq. 6]

The Lagrange multipliers are defined as: \(L^N = \sum \)  

**Lagrange Multiplier Forces.**

The parameter, \(\alpha\), controls the numerical damping of the system. Hibbitt and Karlsson [4] have empirically found the
most effective value for \( \alpha \) to be -0.05. This reduces the “ringing” caused by the automatic time stepping and gives good agreement with analytical solutions.

**Airy Wave Theory**

Airy wave theory is a linearized adaptation of the flow potential, \( \phi \). This method allows multiple wave trains to be superimposed over each other to build a good approximation of typical sea state spectrum found at any particular site. This method was used in the analysis presented here. Airy wave theory makes the incompressible, inviscid, irrotational flow assumption over a flat bottom. It further assumes that the waves are planar and the wave amplitude is “small” compared the water depth. This allows the flow potential, \( \phi \), to be defined as:

\[
\nabla^2 \phi = 0 , \text{ with the fluid particle velocities defined as:}
\]

\[
\mathbf{v} = \frac{\partial \phi}{\partial \mathbf{x}}. \text{ Solving for equilibrium yields:}
\]

\[
\rho \left[ \frac{\partial^2 \phi}{\partial \mathbf{x} \partial t} + \frac{\partial \mathbf{v}}{\partial \mathbf{x}} \cdot \mathbf{v} \right] = -\rho \frac{\partial \mathbf{G}}{\partial \mathbf{x}} - \frac{\partial \mathbf{p}}{\partial \mathbf{x}}, \quad [\text{Eq. 7}]
\]

where

\( \rho \) Fluid density

\( \mathbf{p} \) Pressure

\( g \) Gravity constant

\( \mathbf{G} = g(z - z_o) \) Potential energy per unit mass

Applying the boundary conditions and throwing out the higher order terms to linearize the theory yields the following set of equations that define the fluid particle attributes.

Horizontal fluid displacements;

\[
u_i = \frac{g}{2\pi} \sum_{\text{components}} \frac{a_n \tau_n^2}{\lambda_n} \cosh \left( \frac{2\pi}{\lambda_n} (z - z_b) \right) \sin 2\pi \left( \frac{s_n}{\lambda_n} \frac{t}{\tau_n} + \frac{\theta_n}{360} \right)
\]

\[
\quad [\text{Eq. 8}]
\]
Vertical fluid displacements:

\[ u_z = -\frac{g}{2\pi} \sum_{\text{components}} \frac{a_n v_n^z}{\lambda_n} \frac{\sinh[(2\pi / \lambda_n)(z - z_p)]}{\cosh[(2\pi / \lambda_n)(z - z_p)]} \cos 2\pi \left( \frac{s_N}{\lambda_N} - \frac{t}{\tau_N} + \frac{\theta_N}{360} \right) \]

[Eq. 9]

For these equations to be valid, the following inequalities must be true:

\[ \frac{H}{d} < 0.03, \quad \frac{d}{\lambda} > 20, \quad \text{and the Ursell parameter,} \]

\[ \frac{H}{\lambda} \left( \frac{\gamma}{d} \right)^3 \ll 1, \]

where \( H \) is the wave height, \( \lambda \) is the wavelength, and \( d \) is the water depth. Figure 1 portrays the nomenclature for a single wave train.

\[ \text{Figure 1. Wave Train Nomenclature.} \]

By superimposing multiple Airy wave functions, the model can replicate the wave spectra of a particular net-pen site. The wave history of a typical Sea State 5 is shown in Figure 2. Additionally, the FEA code allows constant velocity currents to be modeled variable with position as well.
Morison's Equation

The FEA code uses Morison's equation (Morison et al. 1950) to apply the wave and current forces to the structure. This is an uncoupled scheme that applies the buoyancy, drag, and inertia forces, due to the fluid, to the immersed beam elements of the structure. Morison's equation for a vertically aligned cylinder of differential length, $dz$, that is displaced a horizontal distance, $\eta$, is:

$$
\frac{dF}{dz} = \frac{\rho C_d D}{2} (u - \eta)(u - \eta) + \frac{\rho C_m \pi D^2 a}{4} \frac{d\xi}{dz} - \frac{\rho \pi D^2}{4} (C_m - 1) \ddot{\eta} dz
$$

[Eq. 10]

where:

- $a$ Horizontal fluid acceleration
- $C_d$ Drag coefficient
- $C_m$ Added mass coefficient
- $dF$ Horizontal force per unit length of the cylinder
- $D$ Effective cylinder diameter
- $dz$ Unit length of the cylinder
- $u$ Horizontal fluid velocity
- $\dot{\eta}$ Horizontal velocity of the cylinder
- $\ddot{\eta}$ Horizontal acceleration of the cylinder
- $\rho$ Water density

Figure 2. Generic Sea State 5 Time History.
The program determines whether a beam element is immersed, and then applies the appropriate buoyancy, drag, and inertia forces as defined below. Buoyancy forces are applied only to vertically aligned cylindrical beam elements. To apply buoyancy forces to a beam element that is not vertically aligned, fictitious vertical beam elements are added to the model as appropriate. The buoyancy force per unit length of a beam element is calculated as:

\[
F_b = -\left( f_1 \rho_w r_n^2 - f_2 \rho_f r_i^2 \right) \mathbf{r}_g \cdot [\mathbf{l} - \mathbf{t}_t] \\
+ \left( f_1 (z_w - z) \rho_w r_n^2 - f_2 (z_f - z) \rho_f r_i^2 \right) \mathbf{r}_g \left[ n_1 \frac{d\mathbf{x}}{dS} \cdot \frac{dn_1}{dS} + n_2 \frac{d\mathbf{x}}{dS} \cdot \frac{dn_2}{dS} \right]
\]

[Eq. 11]

where

\[
f_1 = \begin{cases} 
0 & \text{if the elevation is above } z_w, \\
1 & \text{otherwise}
\end{cases}
\]

\[
f_2 = \begin{cases} 
0 & \text{if the elevation is above } z_f, \\
1 & \text{otherwise}
\end{cases}
\]

and

- \( \mathbf{g} \) gravitational acceleration
- \( \mathbf{n}_1 \) first normal of beam cross-section
- \( \mathbf{n}_2 \) second normal of beam cross-section
- \( r_o \) outside radius of the pipe section
- \( r_i \) inside of pipe section
- \( S \) distance along beam centerline
- \( z_w \) free surface elevation of fluid outside of pipe
- \( z_f \) free surface elevation of fluid inside of pipe
- \( \rho_f \) mass density of fluid inside of pipe
- \( \rho_w \) mass density of fluid outside of pipe

Drag forces on immersed beams is broken into the transverse and tangential drag forces to the beam element. The transverse drag force per unit length is calculated as:

\[
F_D = \frac{1}{2} \rho C_D D \Delta v_n \sqrt{\Delta v_n \cdot \Delta v_n}.
\]

[Eq. 11]
For tangential drag forces, the force per unit length is given by:

\[ \mathbf{F}_T = \frac{1}{2} \rho C_T \pi D \Delta \mathbf{v} \left| \Delta \mathbf{v} \right|^h \quad [\text{Eq. 12}] \]

The inertia force per unit length for a submerged beam element is given by:

\[ \mathbf{F}_I = \frac{1}{4} \rho \pi D^2 \left[ C_M (\mathbf{a}_f - \mathbf{a}_f \cdot \mathbf{t} \mathbf{t}) + C_A (\mathbf{a}_p - \mathbf{a}_p \cdot \mathbf{t} \mathbf{t}) \right] \quad [\text{Eq. 13}] \]

where

- \( \mathbf{a}_p \) Acceleration of a point on a beam
- \( \mathbf{a}_f \) Fluid particle acceleration
- \( C_A \) Transverse added mass coefficient
- \( C_D \) Transverse drag coefficient
- \( C_M \) Transverse inertia coefficient
- \( C_T \) Tangential drag coefficient
- \( h \) Tangential drag exponent
- \( \mathbf{t} \) Unit vector defining the axial direction at a point in a beam

\[ \Delta \mathbf{v}_n = \Delta \mathbf{v} - \Delta \mathbf{v}_t \quad \text{Relative transverse velocity of the fluid} \]

\[ \Delta \mathbf{v}_t = (\Delta \mathbf{v} \cdot \mathbf{t}) \mathbf{t} \quad \text{Relative tangential velocity of the fluid} \]

- \( \mathbf{v}_f \) Fluid particle velocity
- \( \mathbf{v}_p \) Velocity of a point on a beam

\[ \Delta \mathbf{v} = \mathbf{v}_f - \alpha_r \mathbf{v}_p \quad \text{Relative fluid velocity} \]

\( \alpha_r \) Structural velocity factor

The specific coefficients for drag and inertia of a beam element are determined experimentally, or analytically for certain shapes.

**Net Mapping Equations**

Typically a net mesh panel subjected to uniform distributed force acts like a two dimensional catenary, i.e. the net strands have little or no bending stiffness, but have measurable axial stiffness in tension. The FEA code applies drag and inertia forces due to a fluid to be applied to beam elements. To remove the artificial stiffness in beam elements
and to reduce the number of elements required to model the containment net, a net mapping algorithm was devised.

The individual net mesh strands are collapsed into a coarser net mesh. Figure 3 shows the process and the terminology used. Typically for grow out of salmonids, 63.5 mm (2.5 inch) nets are used. This gives an average strand length of 31.7 mm. The mapped strand length is typically in the one meter range.

![Net Mapping Diagram](image)

*Figure 3. Net Mapping Terminology.*

This yields a mapping ratio, $\alpha = \frac{LS_{\text{mapped}}}{LS_{\text{org}}}$ \hspace{1cm} [Eq. 15]

For the structural response, the key parameter is the cross sectional area.

$A_{\text{mapped}} = \alpha A_{\text{org}} = \alpha \pi r_{\text{org}}^2$ \hspace{1cm} [Eq. 16]

For the drag forces, the key parameter is the effective diameter:

$D_{\text{mapped}} = \alpha D_{\text{org}}$ \hspace{1cm} [Eq. 17]
To reduce the structural response due to bending, the moment of inertia for the strands are set near zero.

To validate this method, a test panel of the actual mesh and the mapped mesh was modeled and run under different current scenarios. The reaction forces from these tests were compared to the test results of actual nets is a test tank conducted by Mannuzza [3]. These validation checks showed excellent correlation, less than 5% difference.

Model Development

Development of the FEA net-pen models was divided into three tasks for each model:

- Structural frame development,
- Containment net mapping and development,
- Sea state and current extraction.

Both FEA models use the metric (MKS) system of units. All information that is in non MKS system units was converted to MKS to assure consistency. The structural frame development was performed using ABAQUS Pre™ preprocessor software. This software provides the basic input file that ABAQUS AQUA™ uses to define the initial geometric, structural, and material properties of the FEA model. MathCad 6.0+™ was used to develop the net mapping algorithm and sea state properties. This information was used to revise the input file as required.

Model One Development

Model One is an octagonal floating net-pen design that is 20 m across with a pen depth of 10 m. Figure 4 shows the layout excluding the moorings. The floating support ring is a steel box fabrication, one meter (m) wide by _ m deep. It provides the structural rigidity and attachment points for the mooring system and containment net.

The mooring system consists of eight mooring blocks connected to the mooring buoys by chain. Poly-steel cable (0.038 m. diameter) connect the mooring buoys to the corners
of the frame. The net-pen is moored in 30 m of water. To simplify the model, the mooring chain is modeled as a non-linear spring that mimics the response of an actual mooring chain acting as a catenary connector.

The containment net is made up of eight, 8 m by 10 m panels hung square from the frame. The net mesh modeled is 63.5 mm (2.5 in) square, knotless nylon mesh. A steel octagonal ring is fastened to the bottom edge of the containment net to stabilize the containment net in high currents. The bottom of the containment net was not modeled to reduce the effects of numerical buckling. The mapped containment net has a strand length of one m yielding an a ratio of 31.

**Model Two Development**

Model Two, designated the Pull Up Pen (PUP), is a prototype submersible design being developed by the Ocean Engineering Center of the University of New Hampshire. It is designed for deployment in an open ocean environment with water depths in the 60 to 120 m range. It consists of a 20 m
long by \_ m diameter spar buoy moored to the bottom in a
tension leg configuration. A 4.5 m long aluminum frame slides
over the spar buoy. The frame can be raised and lowered from
the surface to the ocean bottom. Attached to the frame are four,
3 m diameter by 3 m deep cylindrical net pens arranged
symmetrically around the sleeve. Figure 5 shows the layout.

![Model 2 PUP](image)

**Figure 5. Model Two Layout.**

Each containment net is made up of three parts, a
floatation collar at the top, a steel ballast collar at the bottom,
and the containment net connecting the two. The floatation
collar is made up of two, 76 mm (3 inch) diameter high density
polyethylene (HDPE) tubes formed into 4 m diameter rings
separated vertically by a \_ m. Trawl floats are attached to the
containment net between the HDPE rings to provide positive
buoyancy to the containment net. The upper ring is shackled to
the frame. A steel ring forms the bottom perimeter of the
containment net to stabilize the containment net shape. Steel
cables secure the bottom ring to the lower frame. The net mesh
modeled is 63.5 mm (2.5 inch) square, knotless nylon mesh.
The mapped containment net has a strand length of 0.3 m yielding an a of 9.5.

Results and Conclusions

The FEA models provide nodal displacement and stress and strain information at the element integration points for each time step. The wealth of data available can be overwhelming. What will be discussed here are the significant data for the elements that are key to visualizing its response and predicting its life cycle.

To predict the expected life cycle of a net-pen, the endurance limit or fatigue strength of its materials need to be determined. If a material is subjected to cyclical stresses greater than its endurance limit, the material will eventually fail due to fatigue. To determine the endurance limit of a material, the tensile strength of that material is modified by a number of factors that take into account the method of manufacture, environmental effects, and size to name a few. With a calculated endurance limit, a factor of safety can be calculated for those elements.

Model One Results

Model One was subjected to three different runs or scenarios. During Run One, the model was subjected to a steady 1.5 m/s (3 knot) current. During Run Two, the model was subjected to 0.05 m/s current and Sea State Five (SS5) waves. During Run Three, the model was subjected to a steady 1.5 m current and SS5. All currents and wave trains flowed in the positive 1 (x) direction.

For Model One, the critical member analyzed is the main steel frame. The endurance limit for the steel in the structure was determined to be 67.2 MPa which is 16.8% of its tensile strength (400 MPa) or 26.9% of its yield strength.

Comparison of net-pen deformations

Figures 6, 8 and 10 depict the net-pens deformed shape near the end of their three runs. Figures 7, 9 and 11 graph the
vertical motion of the main frame as a function of time. Nodes 1, 5, 9 and 13 are in the center of the right, bottom, left and top sides of the frame respectively.

Figure 6. Run 1 Deformed Shape.

Figure 7. Vertical Displacements.
Figure 8. Run 2.

Figure 9. Vertical Displacements.
Figure 10. Run 3.

Figure 11. Vertical Displacements.

The loads are ramped up over the first two seconds. The strong current in Runs 1 and 3 cause a large deformation of the containment net even with the weighted net ring. Without it, the model fails due to Euler buckling in the net mesh when run with a 1.5 m/s current.
Net-Pen Stresses

Figures 12, 13 and 14 depict the s11 stresses in all the elements of the net-pen for each run.

Figure 12. Run 1.

Figure 13. Run 2.
Since Run 1 is essentially a static problem, the maximum axial stress ($\sigma_{11}$) in the main frame was found not to exceed 5.5 MPa. This corresponds to safety factor greater than 10 based on the endurance limit. Figures 15, 17, graph the axial stress ($\sigma_{11}$) at the four corners of the main frame as a function of time for Runs 2 and 3. The critical factor is the maximum stresses plotted. Figures 16 and 18 graph the corresponding inverted factor of safety based on the endurance limit of 67.2 MPa for A36 steel.

Figure 15. Run 2
Figure 16.

Figure 17. Run 3

Figure 18.
Model Two

The PUP is designed for operations in both a surfaced and submerged mode. The primary objective of this analysis is to predict the maximum sea states the model can be operated at both the surface and submerged at 10 m. A secondary objective is to predict failure points and modes in the structure. This information will allow the developers to adjust the construction details to improve the survivability of the PUP.

Surface Mode

To find the members that were most likely to fail, the model was subjected to Sea State 5 in its surface mode. As expected, both the containment net support frame and the HDPE rings failed due to Euler buckling and plastic strain due to bending. The other members, piling, netting, lower ring, etc., are not critically loaded. The following results details the displacements and stresses of the support frame and HDPE rings for both operational modes. The model was subjected to decreasing sea states until the model ran well. In its present configuration, the maximum sea state that the PUP should be exposed while at the surface is Sea State 2 (H ≤ 0.5 m, λ ≥ 3.5 m). Exposed to SS2, the critically loaded members are the connectors between the upper frame and the HDPE rings and the lower frame. Figure 19 details the displacements of the PUP. Figure 20 portrays the σ11 stresses for the entire frame.

Figure 19. PUP Surface Mode.
Figure 20. Frame Stresses.

The maximum stresses in both the net connectors and the lower frame exceeded the yield strength (250 MPa) of the material as modeled. Adjusting the cross-sectional shape should reduce the stress levels in these members. Future work will concentrate on improving these elements.

Submerged Mode

The PUP was submerged to a depth of 10 m. A point mass of 2000 kg was added to the bottom of the frame to balance the buoyancy of the trawl floats. A spring was added between the pile and frame to keep the net-pen from sinking or surfacing. The frame is still allowed to slide along the piling. As modeled, the PUP performs well up to SS4 (H ≤ 1.8 m, λ ≥ 15.5 m). It is expected that improvements to the surface mode model will improve its performance when submerged.

Figure 21 details the displacements of the net-pen. Red depicts the model in its initial, undeformed shape. Figure 22 portrays the s11 stresses in the frame.
Figure 21. PUP submerged.

Figure 22. Frame Stresses.
Final Thoughts

An important aspect of this work that is not yet completed is to validate this technique with experimental results.

Individual parts of the models have been validated (e.g. the net mapping, and the structural response of the frame), but the response of the entire model needs to be validated. As funding becomes available, researchers at the University of Maine plan to complete this portion of this project.

With the validation completed, these models will start a library of FEA net-pen models. Additional models will be added to this library as needed. Aquaculture researchers, designers, and operators will have a useful tool to evaluate the performance and response of these designs under the applied conditions.

References


The Effect of Currents and Waves on Several Classes of Offshore Sea Cages

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The primary characteristic of concern in sea cage design is that “water in the ocean moves.” We don’t want to belabor points we all understand, but the major water motions result from waves, tidal cycles, general ocean circulation, wind shear and storm surges. At any oceanic site, water will move due to one or more of these causes. This will always happen and it is not a question of if, but when, how often, and of what nature and magnitude?

Some sea farming takes place in areas of minimal water motion and these sites were initially the most sought after locations. Sea cage usage is based upon models thought to work well at these still water sites. However, now it is generally accepted that water motion is a benefit because it is needed to carry fresh oxygen to the fish and to distribute their waste products over an area broad enough for natural decomposition. When water moves, it is clean and free to the farmer and the environmental costs are very low because waste is reduced within the marine system. Within practical limits more water motion is better for sea farming (Ref. 1). As we go offshore, we have no choice but to accept that sea farming will take place in moving water, so we must accept the next truth, that the sea cages best serve their purpose if the motions and deformations of the cages are minimized or, at least, optimized. Accepting these facts, our primary design philosophy at OST can then be stated as follows:

1) We believe that the good health of the fish requires a stable and fixed growing volume. Consistent, repetitive, natural and predictable fish behavior patterns can only be established within sea cages of stable shape and volume.
2) We believe that the sea farmer is best served in his business and operation when the growing volume is fixed, definable and effective.

3) We believe that the further evolution of the sea farming industry, both inshore and offshore, requires, at least, a taut netting foundation upon which new equipment and techniques can be developed.

4) We believe that the future sea farming industry must have available cage designs which experience minimal motion, distortions and stresses caused by waves.

5) The industry must have these sea cages designed as a healthy and safe system integrated into the larger marine habitat.

6) Finally, these sea cages must be provided at a cost that promises an attractive return on investment.

The sea farming industry is daily exposed to "new cage" designs or "improved" cage designs which promise ocean performance. In order to make sense of this marketing and sales bombardment and to predict the performance of the many cage designs, it is necessary for someone to classify the different designs according to expected and achievable performance. Based upon the fish habitat requirements and engineering needs for a stable and well defined geometry, we have chosen to establish sea cage classifications based upon the structural means used to fix the growing volume. This approach is absolutely essential if we are to accurately assess our risks gauge the potentials and answer the tough questions being asked by the critical public.

We have defined four sea cage classes:

Class 1 gravity cages rely on buoyancy and weight to hold the cage shape and volume against externally applied forces. Figure 1 illustrates the typical configuration.
Both buoyancy and weight are forces resulting from gravity, thus the name. Gravity cages will not hold their shape or volume in the absence of gravitational accelerations, and furthermore, their enclosed volume is dependent upon the ratio of gravity forces to water motion forces. Most of today’s commercial aquaculture uses this type of cage. Some of the more common class 1 gravity cages are the popular and much copied Polar Cirkle cage. Wavemaster, Bridgestone, Farm Ocean, Dunlop and any cages using suspended weight or buoyancy to provide volume are also gravity cages.

The Tension Leg Cage shown in Figure 2 illustrates the ideal shape and as deformed by a current. It is really an inverted version of the more typical gravity cages.

Figure 1. Class 1 Gravity Cage.

Figure 2. Class 1 Gravity Tension Leg Cage.
Class 2. Anchor Tensioned Cages such as Ocean Spar Sea Cage shown in Figure 3, rely on anchor tension to hold their shape. If these cages are placed in a zero gravity situation they will still retain their full shape and volume. The application of outside water forces to the netting enclosure will cause the anchor line tensions to increase which resists cage deformation.

Figure 3. Class 2 Anchor Tensioned Sea Cage – Ocean Spar.

However, they will not retain shape unless anchor tension is provided. This means that anchor tensioned cages need to be fixed at the site and, thus, they are stationary or immobile cages.

Class 3 sea cages are self tensioned and self supporting cages such as Sea Station, shown in Figure 4. These cages will hold their shape in the absence of gravity but will also do so without any anchor line tensions. The self tensioning structure resists net deformations.

Figure 4. Class 3 Semi-Rigid Sea Station.
This class of cage is made entirely of beams and columns that are connected only by ropes and not by complex and rigid structural joints. Sea Station is an example of a class 3 sea cage where, floatation, stability and rigidity are provided by the spar buoy which in turn is connected to the rigid steel rim with ropes and netting.

The class 4 sea cages are characterized by rigid, self supporting structures made up of jointed beams, columns and trusses capable of withstanding compression, tension and bending loads. Figure 5 illustrates one possible configuration.

![Figure 5. Class 4 Rigid Sea Cage.](image)

They will retain shape in zero gravity and net deformations are resisted by internal structural forces. This class is characterized by several of the large barge structures experimented with in the past several years.

By defining these sea cage classes, we can evaluate performance based upon the class of sea cage. This simplifies the process of sea cage selection for particular sites. Identification of problems and solutions becomes easier, but most important of all, those involved in the industry are more capable of advancing the technology and profitability once general performance can be separated from the details.
presented by the differing designs within the same sea cage class. With this as our guide, we can now discuss the general behavior of each sea cage class in moving water.

**General Performance in Currents**

Currents cause the greatest loads on any of our sea cage classes and yet because netting is mostly made of holes, the drag forces need not be extreme for properly sized netting. The gravity cages are too compliant, and like a window curtain or a flag in a strong breeze, deform and flap in the water with the net result being a severe reduction in growing volume and some reduction in hydrodynamic drag. The deformations cause unpredictable, high loads on individual twines, with the resulting higher potential of failure. On the other hand, the sea cages class 2 through class 4 resist current deformations, but can experience greater drag forces as a result. The three higher class sea cages have a similar response to the currents, so these can be compared as a group against the gravity cage.

For a comparison, in moderate currents a typical sea cage net panel, normal to the current experiences a drag on the order of 2500 kg. The shape response of this net panel can be approximated by knowing the material elasticity, the drag force and the forces resisting deformations. Figure 6 shows an idealized cross section of a two dimensional net panel uniformly loaded to 42 kg/M2 and analyzed using a non linear finite element technique.

Here the collapse of a gravity net cross section is shown for different suspended weights and compared with the expected and observed deformation in the higher class cages, which are minimal. The volume efficiency of the gravity cages is considerably less than that of the higher class cages. Without taking into account three dimensional deformations, the total volumetric efficiency of the gravity cage can be estimated by the depth efficiency percentages shown at the bottom of each section. The maximum tensions in the individual twines are also illustrated in this figure and shown at mid sections. In each of the gravity net panels a buoyancy force, shown by the
vertical arrows, is needed to oppose the suspended weights and the net section drag. In order to get an approximation of the weight and buoyancy required for an entire cage, the values given need be multiplied by the length of the cage in meters. There are few gravity cages in existence that have the weight and buoyancy required to give depth efficiencies greater than the 42% shown in the figure. Add to this the fact that most oceanic sites will at some time experience higher currents than 50 cm/sec, and it is easy to see why gravity cages have little future in the ocean.

In general, the higher class cages exhibit predictable and well distributed netting stresses when compared to gravity cages. The floating collar structures used with gravity cages are usually flexible or hinged, so they move with the waves’ surface. The strong currents can deform the waterplane area of these cages and compound the deformations of the netting enclosing the fish. Figure 7 is taken from reference 1 showing deformation in a moderate 50 cm/sec current.
Figure 7. Deformation of Gravity Cage Float Circle.

This causes a very complex, 3-dimensional net deformation with severe netting stress points causing high loads on individual twines. A proof of this fact is seen by the steady increase of twine sizes used for the gravity cages and the increasing complexity of nets which employ double netting, twin nets, shock absorbers, more strengthening ropes, more floats, etc. For gravity cages, this attempt to compensate for deformations by increasing net strength is a spiral, eventually closing to failure. The nets deform in the current and then break. The response is to build heavier and more complex nets which require more weights and more floats to stabilize volume, they then cause higher drag and more deformation, which requires heavier nets, etc.

Our experiences with Ocean Spar and Sea Station are showing that lighter nets with higher safety factors can be used
for the job offshore. As an example, Sea Station uses twines of 1 mm diameter while Gravity cages in the same conditions use twine as heavy as 3.17 mm diameter. Gravity cages off the coast of Ireland have evolved in complexity and material sizes so that a single 12,000 cubic meter sea cage weighs 4.5 to 5.0 tons. Whereas a 15,500 cubic meter cage made of 100% Spectra fiber for Ocean Spar weighs 0.90 tons. Based upon the weight only, the complexity of operation with the Ocean Spar net is greatly reduced over that of the gravity cage.

**Sea Cage Submergence — Risk Reduction**

A risk reducing behavior of sea cages is their automatic submergence as flow rates increase above a given threshold. Both Ocean Spar Sea cages and Sea Station sea cages are rigged to take advantage of this behavior as illustrated in Figure 8.

![Diagram of sea cages submergence](image)

*Figure 8. Automatic Submergence of Sea Station.*

For example, a sea station can be buoyancy adjusted so that it normally floats on the surface for currents up to 50 cm/sec. Any storm driven currents above this value cause it to automatically sink. This can be an excellent strategy for putting the fish out of harms way until a storm has passed and requires no human intervention. Fish cannot escape because in both Ocean Spar and Sea Station sea cages the top netting is the same as the sides and bottom and completely sealed. The sinking behavior is difficult to achieve with gravity cages that are surface oriented. One of their major selling points is their compliance with the water’s surface. This requires significant
reserve buoyancy and low structural rigidity. This is confirmed by the use of hinged platforms in the case of Wave Master cages, flexible Polyethylene pipe with Polar Cirkles, or flexible rubber pipe used in the Bridgestone cages. Flexibility and reserve buoyancy work against automatic submergence. Excess reserve buoyancy requires high drag loads to cause submergence and low structural rigidity means the floating structure will easily bend (in the vertical direction) once it is submerged. Of the gravity cages, only the Tension Leg Cage easily exhibits this automatic sinking property, but in higher currents, anchoring loads and deformations become extremely high.

**Motion in a Seaway — Risk Reduction**

The criteria of minimizing sea cage motion in a sea way also reduces risk of failure. Our approach has been to use structures with low reserve buoyancy compared to the mass of the system. This puts the sea cage mostly underwater where wave induced water motions are quickly attenuated with depth. Both Ocean Spar Cages and Sea Station cages exhibit excellent performance in a sea way while floating on the surface. The low waterplane area of the spar buoys means that motion inducing forces remain minimal. Short period waves pass through without causing sea cage or fish motions, while both sea cages become wave surface followers for large period waves. For long period waves the surface following characteristic means relative motions between fish and cage are minimized and nearly zero. The relatively high drag of the netting enclosure damps any resonant response that might be expected.

The typical gravity cage floats on the surface and shock loading in the netting and ropes of the sea cage transfer continually between surface float and suspended weights. If it happens that waves are superimposed on the cages floating in currents, it is obvious that the gravity cages will experience very high additional loading on the twines. Because the net deformations are extreme, the high loads can and will occur
anywhere in the net. The result is that the entire net must be made heavier to compensate for this. This is part of the explanation for the increasing complexity seen in the offshore gravity cage nets being used in the salmon industry.

The higher class cages will be relatively unaffected by combined waves and currents since the top and bottom of the net panel, being connected to rigid vertical structures will move in phase. Being rigidly connected to the supporting structure, the stresses in the netting vary gradually and are very predictable. Thus the high stress areas of the fish enclosure can be reinforced and supported without increasing the strength and weight of the entire net. This explains why the higher class cages can have nets that are much lighter and yet significantly more reliable than gravity cage designs. And all of this at nearly 100% volumetric efficiency.

Another strategy for reducing the effects of moving water is to allow the cages to drift with the current. Figure 9 illustrates this point. In the case of the class 1, 3 and 4 cage systems, the drag forces are non existent because the velocity between cage and water is reduced to zero. Although there are no current attempts to use this technique the advantages of it are worth discussing.

![Figure 9. Sea Station-Ocean Drifter (25,000 cubic meters volume).](image)

This strategy is not available to the class 2 anchor tensioned systems because they are not self supporting without anchors. For the class 3 Sea Station, the only induced motions
are then from its response to waves, which tend to be very low or negligible. Although this strategy sounds idealistic, there are areas, such as the Straits of Juan de Fuca, where the water motion cycles will allow this tactic and will keep the sea cages within a bounded area. In this case, divers could work the system 24 hours per day because current is no longer a factor, and waves do not affect the cage. In the version shown in the drawing, it is possible to have a simple diver lock out door well beneath the water surface to make diving even easier. Fish waste would be distributed over a large area for natural decomposition and the fish would experience only minimal water motions. The structural design would be simplified because the water motion forces will be greatly reduced when compared to systems that are anchored. The behavior of the model Sea Station has been investigated under this free drifting mode and it is technically achievable with present technology.

Where Do We Go From Here?

We must concentrate on the class 2 and the class 3 sea cage designs because gravity cages do not meet any of our basic design criteria when applied in opened water. Gravity cage deformations in currents are the culprit. If we insist on taking Class 1 gravity cages to oceanic sites, their performance deficiencies will be compounded and the development of the industry will be greatly hindered. These are strong words, but they can be supported by theory, experience and the growing evidence of operational and environmental problems expressed daily in our publications. We believe the present stagnant state of development in the sea farming industry is a direct result of gravity cage application to sea farming. We believe that every problem confronting the gravity cage industry can be solved by using the higher class sea cages (reference 2). For example, some fish health issues and high stress levels, feed dispersion inefficiencies, operational inefficiencies and predation by marine mammals can all be traced directly to gravity cage enclosures. And think about this, what other animal besides farmed fish are raised in gravity cages that continually change
shape, trapping the creatures in folds of netting, as the growing volumes approach near zero? Can we say that we are treating our final product well?

Since water simulates an anti-gravity environment, stability must be provided by structural rigidity or membrane tensions in the netting enclosure. For an example, without a stable and firm foundation, the development of machines, instrumentation and techniques for improving sea farming is greatly hindered. Class 2 through class 4 cages provide this stable base for sea farming evolution. We need a better understanding of the advantages of each cage class and its potential application. To encourage investment and development, risks must be evaluated based on the higher class sea cages and not on gravity cages. After studying the sea cage designs for nearly 10 years, I have come to the conclusion that gravity cages have no place in the ocean, and there is a growing body of evidence that they are even a poor cage class to use at sheltered water sites.

**Summation**

After attending almost every offshore sea farming conference since 1989, we have decided it is necessary to stand up and talk down to the gravity cage mentality, it is the wrong technology for the application. The ocean does not grant wishes, it doesn’t tolerate the unfit. I would like to end with an analogy describing the gravity cage fixation of our industry.

Imagine that in front of us lies a large pile of rocks—perhaps 10 tons. We ask that each one of you go out to find a vehicle that can haul these rocks away. We will bet that all of you will look for a truck and that none of you come back with a passenger car to do the job. The passenger car represents the class 1 vehicle and the trucks, higher class vehicles. Yes, we can haul the rocks with the passenger car, but not very well. And yes, we can all cite situations were gravity cages have withstood the raging wind, wave and current, but statistically they are bound to fail. And yes, we can modify the passenger car to do the job a bit better, but normal performance of the
vehicle and the sea cages must be matched to the job if statistically low failure rates and efficient, profitable operations are our goals. We all know each vehicle class and its capabilities with hardly a second thought. That is where we have to be in our understanding of sea cages if offshore sea farming is to become an industry in our lifetime. And higher class sea cages must be applied in the sheltered water aquaculture industry if its problems are to be solved and the challenge of continuing lower salmon prices are to be met. If this sounds too good to be true, it is not. It is the reality of the situation. Question us, challenge us, the industry needs to address these issues.

References

