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DOE/NOAA/OTEC-10

AT-SEA TEST SYSTEM POINT DESIGN FOR A ONE-THIRD SCALE COLD WATER PIPE

Technical Report 1601

December 1979

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Silver Spring, Maryland

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AT-SEA TEST SYSTEM POINT DESIGN FOR A ONE-THIRD
SCALE COLD WATER PIPE

William H. Sutherland, Editor

December 1979

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Silver Spring, Maryland

At-Sea Test System Point Design
for a One-Third Scale Cold Water Pipe

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ABSTRACT

One step in the development of the technology for Ocean Thermal Energy Conversion (OTEC) Cold Water Pipes (CWP) is the at-sea testing of a fiberglass reinforced plastic nominal 10-foot diameter pipe. This report presents a design procedure and criteria for developing test hardware by scaling down a 30-foot diameter OTEC 10/40 MW Pilot Plant CWP design. This report contains an example point design for the pipe, instrumentation to be used during the at-sea tests, and methods for selecting the support platform and mooring. The design in this report starts with a scale model of a larger prototype, and then is modified to address the problems of fabrication and of survivability and handling during the 1/3rd scale model tests.

1. INTRODUCTION

1.1 ADMINISTRATIVE INFORMATION

The National Oceanic and Atmospheric Administration (NOAA) of the Department of Commerce (DOC) is assisting the Department of Energy (DOE) in the development of ocean engineering technology for OTEC. Development of a method for the design of CWP's is part of NOAA's responsibility. This report was prepared by ORI, Inc. under their system support contract for the OTEC Ocean Engineering Technology Development Program for NOAA's Office of Ocean Engineering. The point design presented herein was developed under ORI direction to provide contract guidance for the CWP at-sea test. The sections on the test article design (Sections 2 and 3) and the instrumentation system (Section 4) were developed by Science Applications, Inc. (SAI) under a modification to their preliminary cold water pipe design contract. Information on fiberglass design and fabrication was developed with the assistance of Structural Composites Industries, Inc., (SCI) and Fiberglass Structural Engineering, Inc.

1.2 OBJECTIVES

In order to develop the OTEC technology necessary to design, fabricate, deploy, and operate a 10/40 MW Pilot Plant in the 1983 time frame, DOE has identified three critical technical problems that must be solved. These problems are to develop:

- A low technical risk CWP
- A reliable electrical transmission cable from an ocean OTEC plant to shore
- A reliable method of maintaining heat exchangers free of biofouling.

The major objective of NOAA's OTEC CWP Program is to reduce to an acceptable level the technical risk for an OTEC 10/40 MW Pilot Plant CWP. The technical risks in going to a 10/40 MW Pilot Plant directly from mathematical models and very small scale laboratory tests must be reduced. Mathematical models need simplifying assumptions in order to be solved, and laboratory tests are subject to the limitation that not all important hydrodynamic phenomena scale simultaneously. Thus it was determined by DOE and NOAA that a means of reducing technical risks for the pilot plant CWP was to conduct a fairly large scale at-sea test of the CWP.

In achieving the major objective of the NOAA OTEC CWP Program and solving one of DOE's three critical technical problems, the following intermediate objectives will be addressed by the At-Sea Test Program:

- Develop practical handling techniques for the substantial sizes and forces involved in an at-sea environment
- Develop an understanding of CWP scaling by testing much nearer prototype size than before
- Improve computer design tools by providing verification data
- Refine cost-estimating information for FRP CWPs
- Examine the safety and reliability of FRP as a viable option for OTEC CWP's.

1.3 TECHNICAL BACKGROUND

Under previous contracts to NOAA, SAI and TRW have prepared alternative preliminary designs for CWP's to be used on the 10/40 MW Pilot Plant (References 1 and 2). From these analyses fiberglass reinforced plastic (FRP) has been chosen as one of the best material to receive major design emphasis. Steel is also considered as being a viable option for 10/40 MW CWP design. Elastomers, polyethylene, and concrete are also under consideration but are lower on the probability list. Various suspension systems and jointing are being studied to determine the least costly reliable approach. There is some information on open sea performance of a steel CWP available from the Deep Oil Technology X-1 Sea Tests (Reference 3). Since FRP is the most viable alternative, DOE has decided to use it in a 1/3rd scale at-sea tests.

Most data on CWP behavior phenomena such as vortex shedding have been obtained on small laboratory sized models. There is a serious difference of opinion among OTEC experts as to the importance of vortex-induced loading on CWP survival. Some experts think that vortex-induced loading at pipe sizes as large as and larger than the CWP test article will be small while others assert that vortex suppressors will surely be necessary on the CWP test article. Information on vortex shedding from the at-sea tests will allow extrapolation of vortex loads to pilot plant and perhaps commercial size CWP's.

Several computer-aided design tools are available to define bending behavior of a coupled platform/CWP/SKSS system. The most current model is the NOAA/DOE Analytic Code described in Reference 4. Other computerized mathematical models presently under development on NOAA contract examine the shell stresses in a CWP. All models contain necessary simplifying assumptions and have been verified with only limited data. One objective of the At-Sea Test Program is to collect sufficient structural response data near prototype scale to verify or allow modifications and updates to be made to the mathematical models so that they may be used with confidence for the OTEC 10/40 MW Pilot Plant design.

Data on fabrication of large diameter FRP pipes comes from outfall pipes in which (a) the primary stresses are hoop stresses and (b) strict quality control has not been necessary. Deployment steps and procedures follow examples from routine marine construction projects. The CWP test article must be designed with strict quality control and to accept significant lengthwise tensile stresses. Accurate fabrication and deployment data collected during the CWP At-Sea Test Program will reduce the engineering unknowns for the 10/40 MW Pilot Plant.

It is known that FRP offers superior corrosion resistance to steel. Nevertheless, there are questions now about the ability of FRP to withstand the cyclic stresses of wave loading and to be manufactured into a CWP with low factors of safety without failure at sea. If the At-Sea Test Program is able to demonstrate fabrication, deployment, operation at sea, and recovery without significant failures, a great deal of added confidence can be placed in the use of FRP for the OTEC 10/40 MW Pilot Plant design.

1.4 DESIGN PHILOSOPHY

This document describes the development of a preliminary 1/3rd scale CWP test article design with supporting hardware to conduct an At-Sea Test of its performance. The prototype for this design was the SAI 30 foot diameter FRP sandwich CWP attached to a moored barge of Johns Hopkins University Applied Physics Laboratory (APL) design. The environmental characteristics chosen for the design studies for both prototype and 1/3rd scale model combined the worst loads at the Puerto Rico and Hawaii OTEC test sites. The prototype CWP was designed to remain attached to the barge through a 100 year recurrence interval storm. The model CWP is designed to be detached from the barge if waves should be forecast to exceed Froude-scaled replicas of the 100-year storm. The model CWP would be suspended on a cable below a buoy. This detach-reattach sequence is expected to reduce surface platform/CWP coupling to the point that bending loads can be kept below design values with the CWP detached, in waves up to the size of those of the 100 year storm. By lowering the model pipe several hundred feet, strong wind-driven surface currents are also avoided.

The model is not an accurate geometric and mechanical scale model of the prototype SAI 30 foot diameter FRP CWP. The approach taken was first to scale down the prototype and then to modify that design as necessary to accommodate all important practical aspects of the At-Sea Test Program. CWP sandwich construction, deployment and recovery procedures developed for the full-scale pipe were preserved as much as possible.

One of the most difficult aspects of the design process was dealing with current loading. For the prototype 1000 meter (3281 ft) long CWP, currents on the lower two-thirds of the pipe are very low. This creates a large damping effect on pipe motions and exposes only the upper third of the pipe to vortex-excited oscillations. The rapid fall-off in current velocity along the pipe length also keeps vortices from coupling, and allows the prototype to be designed without vortex suppressors. For the model to be deployed in the same geographic locations as the prototype, vortex shedding loads become a prime design requirement. Currents do not fall off with depth as rapidly as waves, and thus for models currents have higher velocities than Froude-scaling would dictate. The model will only be exposed to the upper 1000 feet of the water column. In this range currents may be high over the entire length of the CWP test article. Because currents change slowly with depth on the model, vortex lock-on becomes more probable, while the magnitude of vortex loading is nearly a scale-factor larger than the prototype. Vortex suppressors were judged necessary for the CWP test article design presented here.

In modifying the prototype 10/40 MW Pilot Plant CWP, it was always necessary to insure survival of the CWP test article. Without a complete test program, it would not be possible to demonstrate positively the safety and reliability of FRP as an option for CWP construction.

Extensive preliminary design studies for the OTEC 10/40 MW Pilot Plant form the background material on which the At-Sea Test System is designed. This material has a direct bearing on the requirements for the 1/3rd scale model test article, test support hardware, and at-sea test. The present document, except in a few instances, does not repeat the information

in the full-scale studies but rather emphasizes information concerning the 1/3rd scale test. Several preliminary design studies of various aspects of the OTEC 10/40 MW Pilot Plant are listed in the References.

1.5 REPORT DESCRIPTION

The remainder of this report discusses the methods used to reach design decisions for a specific version of the model At-Sea Test System. Section 2 describes the design procedure followed in defining design criteria. Sample calculations are presented for the example final design. Section 3 contains a physical description of the model 10/40 MW Pilot Plant CWP with preliminary design sketches. Section 4 has an example instrumentation system design which is appropriate for monitoring the behavior of the test article during testing and for obtaining the data necessary to validate analytical models and improve the design methodology. The final section of the report, Section 5, contains guidance for selecting a test platform and mooring.

2. DESIGN PROCEDURE

2.1 INTRODUCTION

This section demonstrates one way of achieving the design of the test article, the 10 ft diameter 1000 foot long FRP cold water pipe. The procedures and results for a specific preliminary design are shown as an example. A brief additional paragraph and diagram (Figure 2.3) outline the design decision steps in a more general way not tied to the specific design.

2.2 GENERAL REQUIREMENTS FOR EXAMPLE TEST ARTICLE

The general requirements for the 1/3rd scale CWP design are presented in Table 2.1.

2.3 SCALING

The question of scaling in the 1/3rd scale CWP design encompasses physical scaling and also fabrication, deployment, and operations of an FRP CWP. The requirements of these latter three aspects of the test conflict with strict physical scaling. The point design described here is as representative of the SAI full-scale design as possible, but scaling was compromised somewhat in order, to achieve a structurally viable design. In the material below, direct physical scaling will be discussed, followed by a discussion of the modifications in scaling which may be required to meet other test objectives.

Table 2.1. GENERAL REQUIREMENTS

1. Platform Requirements.

- 1.1 Approximately 1/3rd scale APL barge.
- 1.2 No maneuvering capability, hence CWP to be designed for all headings. (Active tensioning to change platform heading 90 degrees will be required to expedite testing).
- 1.3 Moonpool required for access to the upper end of the CWP for attach and detach for normal installation and recovery and for survival conditions.

2. CWP Test Article Requirements.

2.1 Environmental requirements.

Hawaii and Puerto Rico environmental specification. (see Reference 5)

2.2 Design requirements

2.2.1 The test article shall be appropriately scaled (linear scale factor, 1/3 of the full scale design.)

2.2.2 The test article shall have a minimum factor of safety of 1.5 on extreme cyclic stress and 2 on ultimate load conditions; e.g., buckling.

2.2.3 The test article shall have a detach capability to be used for sea states in excess of $H_{1/3}=12$ ft.

2.2.4 In the detached mode the CWP shall survive a 100 year storm at the test location.

2.2.5 The test article shall be designed for a fatigue life of 10 years.

The theory of models is based upon dimensional analysis, which states that if all of the dimensionless independent variables are equal between two experiments, the corresponding dependent variables are also equal. In the present case, the independent variables are all of the design parameters which describe the system and all of the environmental parameters which could excite the system. The dependent variables describe the system response. According to theory, there is a minimum number of dimensionless variables which sufficiently describe the system, its excitation, and response.

Dimensional analysis provides useful insights into the appropriate design of experiments, but the problem of proper experiment design also depends on an understanding of physical laws and the issue of technical feasibility. Rather than discussing scaling relations derived from dimensional analysis in general, the scaling relations are presented and their adequacy tested on the basis of known physical laws and on technical feasibility. This tentative set of scaling laws is presented in Table 2.2.

Table 2.2 Tentative Scaling Laws
(Scale Factor = λ)

Length	λ
Mass density	1
Acceleration of gravity	1

The tests under consideration are to be performed in water and in a normal gravitational environment, hence the scaling laws for the mass density of water and the acceleration of gravity must be unity. Since g must be constant with scale factor and we preserve length (geometric) scaling, then time scales as $\sqrt{\lambda}$. The weight of a cube of water will scale as λ^3 , and if the weight of a body which is geometrically scaled displaces this cube, the buoyant force must scale as λ^3 . Thus weight or force scales as λ^3 . Using these derived relations, the tentative scaling laws for a number of quantities of known significance can be derived. They are listed in Table 2.3.

Table 2.3. Derived Scaling Relations for Significant System Parameters

Parameter	Dimensions	Scaling Laws
1. Design Related Cold Water Pipe Parameters		
Length	L	λ
Diameter	L	λ
Flexural rigidity, EI	FL^2	λ^5
Distributed weight in air	FL^{-1}	λ^2
Distributed weight in water	FL^{-1}	λ^2
Bottom weight	F	λ^3
Surface roughness	L	λ
Flow rate	LT^{-1}	$\sqrt{\lambda}$
Volume rate	L^3T^{-1}	$\lambda^{5/2}$
Mass flow rate	ρL^3T^{-1}	$\lambda^{5/2}$
Structural damping ratio	1	1
2. Environment		
Wave height or length	L	λ
Wave period	T	$\sqrt{\lambda}$
Current velocity	LT^{-1}	$\sqrt{\lambda}$
Wind velocity	LT^{-1}	$\sqrt{\lambda}$
3. System Response		
Deflections/displacements	L	λ
Velocities	LT^{-1}	$\sqrt{\lambda}$
Accelerations	1	1
Stress	FL^{-2}	λ
Angles	1	1
Angular rates	T^{-1}	$\lambda^{-1/2}$
Angular accelerations	T^{-2}	λ^{-1}
Pressure	FL^{-2}	λ

The quantities in Table 2.3 are of known physical significance. Quantities known to be insignificant are not included, e.g., temperature, magnetic properties, and others. Other quantities which have not been included might be significant. Examples are the modulus of elasticity, E , alone (i.e., not combined with the pipe-section moment of inertia, I , as the product EI), and CWP wall thickness, t . These two parameters, I and t , govern the buckling behavior of the CWP. E scales as λ . Thus, if in the prototype $E = 3 \times 10^6$ psi, and $\lambda = 3$, then in the model E should be 1×10^6 psi. Similarly, if prototype thickness equals 1 inch, model thickness equals .33 inches. However, if the E value for the model varies significantly from E value for the prototype, the result of the fabrication process will not be correctly modeled. Also, it is noted that an important objective of the test is to validate an analytical beam model of the CWP. For beams the product EI is the significant structural parameter, and will be tentatively used as the governing structural parameter for the 1/3rd scale test.

The tentative scaling relations can be tested by applying them to the well known dimensionless parameters which govern dynamics of the CWP. Table 2.4 shows the relationship between the dimensionless parameters in the prototype and the model when scaling relations of Table 2.2 are used.

The tentative scaling laws meet all of the dimensionless parameters specified except Reynolds number. The maximum value of Reynolds number along the CWP is a factor of 5.19 too small for the 1/3 scale test if current velocity is correctly scaled. However, the current velocity is the same as for the prototype CWP design. Thus, the maximum current is approximately double the correctly scaled value. Also the current profile is not correctly modeled -- over a 1000 foot test pipe specimen the current is essentially uniform, whereas in the prototype the current velocity drops off to a much lower value over the lower portion of the pipe. (See Figure 2.1)

Table 2.4

Comparison of Selected Dimensionless
Parameters for the Prototype and Model

Dimensionless Parameter	For the Prototype	For the Model
Froude number,	$\left(\frac{V}{\sqrt{gD}}\right)_P$	$= \left(\frac{V}{\sqrt{gD}}\right)_m$
Reynold's number,	$\frac{(VD)}{v_{\text{water}}}$	$= \lambda^{-3/2} \frac{(VD)_m}{v_{\text{water}}}$
Reduced stiffness,	$\left(\frac{EI}{\rho g D^5}\right)_P$	$= \left(\frac{EI}{\rho g D^5}\right)_m$
Pipe mass ratio,	$\left(\frac{\rho_P A_P}{\rho D^2}\right)_P$	$= \left(\frac{\rho_P A_P}{\rho D^2}\right)_m$
Internal flow mass ratio, ratio	$\left(\frac{\rho_f A_f}{\rho D^2}\right)_P$	$= \left(\frac{\rho_f A_f}{\rho D^2}\right)_m$
Aspect ratios	$\left(\frac{L_1}{L_2}\right)_P$	$= \left(\frac{L_1}{L_2}\right)_m$

Figure 2.1 Actual Current Profile and Theoretical Profile for the 1/3rd Scale Model

The above discussion shows that strict scaling of the 1/3rd scale model cannot be maintained if (a) the fabrication process is to be retained and (b) the design is based on the same environmental specification as the prototype. However, the design process can be initiated by utilizing the scaled design parameters, and as will be shown in the following, the modifications required to achieve a viable design do not result in a dramatically different CWP wall design.

2.4 DESIGN APPROACH

This section outlines the steps taken in the example design. More detailed descriptions, and diagrams showing the way some of the computational results were arrived at, will be found in Section 2.5.

The design flow chart followed in obtaining the example design is shown in Figure 2.2. Some additional decisions which should be addressed in refining the design approach are portrayed in the flow chart of Figure 2.3. This second flow chart emphasizes the thought which must be given to items other than the CWP itself and the effects of changing scenarios on design allowables.

Referring back to the steps for the specific example, as shown in Figure 2.2, the process begins with the establishment of requirements as outlined in the preceding section. The application of the scaling relations results in a CWP with a 1.7 inch neutrally buoyant core at the upper end and with uniform skins .17 inches thick. The results are as shown in Table 2.5, in the column labelled "direct scaled model requirements." In comparing this design with the test requirements, it can be shown that in the full-scale current environment, the direct scaled CWP will not hang within a desired angle of 6 degrees to the vertical, but instead at an angle of approximately 26° . (The results of the computation leading to this result are among those shown in greater detail in the next section.)

There are two options for modifying the design to achieve the more appropriate attitude for the CWP: (1) add a bottom weight; and (2) increase the skin thickness. A preliminary cost assessment shows that adding a bottom weight is the more cost effective choice.

Figure 2.2 Design Approach for the Example CWP

Figure 2.3 Detailed Design Decision Steps for a Test Article Cold Water Pipe

Table 2.5
Scaling of Design Parameters

PARAMETER	SYMBOL	SCALING LAW	PROTOTYPE*	DIRECT SCALED MODEL REQUIREMENTS	MODEL DESIGN
1. LENGTH	L	λ	3,000 FT	1,000 FT	1,000 FT
2. Inside Diameter	V_O	λ	30 FT	10 FT	10 FT
3. SKIN THICKNESS (AT UPPER END)	t_s	λ	.5 IN.	.17 IN.	.3 IN.
4. CORE THICKNESS (AT UPPER END)	t_c	λ	5 IN.	1.7 IN.	1.84 IN.
5. FLEXURAL STIFFNESS	EI	λ^5	$7 \cdot 10 \text{ LB FT}^2$	$2.88 \cdot 10^9 \text{ LB FT}^2$	$1.63 \cdot 10^{10}$
6. MASS/FT	ρA	λ^2	2800 SLUGS/FT	311 SLUGS/FT	327 SLUGS/FT
7. WEIGHT/FT SUBMERGED	W_W	λ^2	520 LB/FT	58 LB/FT	92 LB/FT
8. WEIGHT/FT UNDER TOW	W_W	λ^2	520 LB/FT	58 LB/FT	92 LB/FT
9. DENSITY OF BALLAST	γ_b	1	63.3 LB/FT^3	63.3 LB/FT^3	62.83 LB/FT^3
10. WEIGHT/FT IN AIR	W_S	λ^2	3700 LB/FT	411	388
11. STRUCTURAL MASS/FT	M_S	λ^2	114 SLUGS/FT	12.7	12
12. BOTTOM WEIGHT	W	λ^3	0	0	283,000 IN AIR 180,000 IN WATER

*NOMINAL PROPERTIES, E.G., $L = 3280$ ACTUALLY.

The next step in the design approach is to consider the vortex shedding response of the CWP. The operating current has a high and comparatively uniform value over the model length, as already discussed under scaling. This results in a potentially severe vortex-induced strumming condition on the model pipe. Preliminary investigation shows that the directly scaled CWP will strum in the third mode. If the flexural stiffness is increased, strumming can be made to occur in the second mode. It is noted that this significantly reduces strum stresses in that strum amplitude may be essentially one diameter, independent of mode number, Reference 6. (That is, for roughly equal amplitudes, there would necessarily be more curvature in the third mode than in the second.)

Second mode strumming stresses are still unacceptably high; However (1) to stiffen the pipe to the point that strumming occurs in the first mode would greatly increase cost, and (2) vortex suppression should reduce strumming stresses to acceptable levels. Thus, at this point 1) a range for CWP stiffness (EI greater than 1.8×10^{12} lb in²), and 2) a requirement for strumming suppression, are established.

Various combinations of E and total skin thickness, $2t_s$, will yield the required EI . Core thickness can be considered somewhat independently because overall EI , for bending of the CWP as a beam, is insensitive to the core thickness. In FRP, material allowables and Young's modulus vary in direct relation to one another. This leads to the selection of a high value for E and small t_s . Also generally, it is to be noted that material costs vary directly with t_s . These considerations lead to a value for $E = 5 \times 10^6$ lb in² and $2t_s = .5$ in. -- similar to the nominal requirements set out in the introduction to section 2.

Other load conditions must also be considered in establishing preliminary CWP parameters and prior to evaluation with the NOAA/ DOE code, Reference 4. During bending, the fibers on one side of the CWP are loaded in compression, while on the opposite side they are loaded in tension. The maximum tensile load is the tensile ultimate for the material, approximately 33 ksi for an $E = 5 \times 10^6$ lb in². Ultimate failure on one side of the pipe is governed by the tensile rupture, while on the opposite side by local

buckling or "crippling." The bending strength of the CWP should not be limited by crippling. Crippling should occur at or above the tensile ultimate. This consideration leads to the requirement that the total equivalent thickness of the CWP, t^* , satisfy t^* greater than .6 in. (Again, further details of the computations made in arriving at this result will be found in a later section).

The CWP is also subject to buckling due to the pressure differences across the CWP walls. Static pressure differences arise when the full scale CWP is pumping water to the surface. In the 1/3rd scale design under consideration, the CWP will not be required to transport water, so these static loads will have no effect in the test. (See Footnote). Dynamic pressure differences are induced by surface waves, and for compliant pipes, are significantly influenced by the longitudinal oscillations of the CWP water column coupled with CWP dilational oscillations. The pipe design at this point in its evolution has already been established to be dilationally stiff. Therefore, the dynamic pressure difference is between a logarithmically decaying external pressure and a linearly varying internal pressure. This leads to a design pressure requirement of 21 psi (which corresponds to an rms pressure of 2.8 psi X factor of safety of 2 X a factor of 3.8 to estimate the extreme value). (Note especially that the factor of safety here is applied to an extreme value.) This design pressure determines the core thickness

Footnote: Another important difference between the full-scale operating plant and the at-sea test is the force associated with the inertia of the water within the pipe as the platform and pipe move in response to wave action, particularly in heave. In the test setup the pipe will be open at the top and bottom, so that the pipe will move vertically essentially independently of the column of water inside it. But in the full scale operation there may be greater constrictions. The design of the full scale pipe and the connection between it and the platform may well have to take into appropriate account some part of the substantial mass of water in the pipe. In fact, even during the 1/3rd scale model test, at such times during the deployment as the ends of the pipe are plugged, the mass of water within the pipe must certainly be considered in calculating inertia terms.

requirement. That is, the cost of core per unit volume is significantly less than that of skins, so the function of the core is to separate the skins to achieve the desired cross-sectional bending stiffness. These considerations lead to a skin thickness of approximately $t_s = .3"$ and core thickness, $t_c = 1.84"$ at the upper end of the CWP. Core thickness tapers along the length of the CWP in direct relation with the buckling pressure. Clearly these properties meet both the crippling and vortex shedding requirements outlined above.

The design is now developed to the point that it should be evaluated by the NOAA/DOE code Reference 4, as the next step (as indicated in the decision step process shown in Figure 2.2). The results of the computer code predictions are summarized in Table 2.6. (Again, further details are shown later.) Preliminary fatigue analysis shows that the CWP life is not governed by these stresses but is governed by vortex shedding providing it is assumed that the CWP is detached for $H_{1/3}$ greater than 12 ft,. If the strum response can be limited to approximately 1/6 diameter, a fatigue life of approximately 10 years is achievable. Hence, for the defined environmental conditions, vortex suppression is required.

Some consideration needs to be given to the allowable weather conditions for towing. (A thorough investigation of CWP fatigue associated with towing requires information on the CWP assembly and installation site locations and intervening current and weather conditions. A specific assessment of the fatigue damage associated with towing must be performed jointly with the detailed development of a towing plan). At this point in the design evolution it is sufficient to show that the acceptable weather conditions are not unduly restrictive. The preliminary towing analysis indicates that the CWP can be towed in significant wave heights $H_{1/3}$ equal to or less than 6 ft. This is not a severe limitation on towing.

Table 2.7 further summarizes the baseline 1/3 scale CWP design presented herein and contrasts it with the FRP full scale design presented in the SAI final report (Reference 1).

Table 2.6
CWP Stresses

● NOAA/DOE-CODE-SUPPLIED STRESSES IN PUERTO RICO ENVIRONMENT

$$t_{\text{core}} = 1.6", t_{\text{skin}} = .27" (.54" \text{ total})$$

● STATIC

$$\sigma_{\text{max}} = 3250 \text{ psi}$$

● DYNAMIC

$$\sigma_{1/3} (H_{1/3} = 4) = 760 \text{ psi}$$

$$\sigma_{1/3} (H_{1/3} = 12) = 12800 \text{ psi}$$

$$\sigma_{1/3} (H_{1/3} = 36) = 38300 \text{ psi}$$

● RELATIVE ANGLE

$$\text{For } H_{1/3} = 4 \text{ ft, } .38^\circ$$

$$\text{For } H_{1/3} = 12 \text{ ft, } 7.2^\circ$$

$$\text{For } H_{1/3} = 36 \text{ ft, } 19^\circ$$

Table 2.7
Comparison of Prototype and 1/3 Scale Model

CONSIDERATION	PROTOTYPE	MODEL
1. CNP SYSTEM DESIGN		
1.1 PLATFORM INTERFACE		
1.1.1 MOONPOOL	REQUIRED.	REQUIRED.
1.1.2 BEARING	2DOF SPHERICAL BEARING INSPECTION, MAINTENANCE IN AIR.	2DOF SPHERICAL BEARING NOT MAINTAINABLE IN AIR.
1.1.3 CNP HANDLING EQUIPMENT	JACKING SYSTEM AND CRANE.	CRANE REQUIRED.
1.2 CNP STRUCTURE		
1.2.1 PIPE WALL DESIGN	SKINS WITH NEUTRALLY BUOYANT SYNTACTIC FOAM CORE.	SKINS WITH NEUTRALLY BUOYANT SYNTACTIC FOAM CORE.
1.2.2 VORTEX SUPPRESSION	RECOMMENDED (1).	REQUIRED.
1.3 LOWER END		
1.3.1 SCREENS	REQUIRED.	NOT REQUIRED.
1.3.2 BOTTOM WEIGHT	NOT REQUIRED.	REQUIRED (APPROX. 180,000 LB.)
2. OPERATIONS - FLOW RATE	6.5 FT/SEC.	0
3. FABRICATION		
3.1 SKINS	SCI INCREMENTAL USING TFT* (CONTINUOUS PIPE).	APPROX. 100' SECTIONS USING TFT OR FILAMENT WINDING.
3.2 CORE	SCI INCREMENTAL USING PRECAST POLYESTER RESIN AND CERAMIC MICRO-BALLOON TYPE.	TFT OR CAST POLYESTER RESIN AND CERAMIC MICRO-BALLOONS.
4. TRANSPORTATION	HORIZONTAL TOW USING BRACKISH WATER BUOYANCY.	HORIZONTAL TOW USING FRESH WATER BUOYANCY AND AUXILIARY BUOYANCY.
5. DEPLOYMENT	RELEASE BRACKISH WATER AND TAKE ON SEA WATER AT LOWER END.	RELEASE BRACKISH WATER AND TAKE ON SEA WATER AT LOWER END.
6. RECOVERY	PUMP BRACKISH WATER INTO UPPER END OF PIPE.	SUPPORT BOTTOM WEIGHT, PUMP BRACKISH WATER INTO UPPER END OF PIPE.
7. INSTALLATION	COMPLETED IN AIR.	COMPLETED IN WATER.
TFT - TRANSVERSE FILAMENT TAPE, AN SCI PROCESS.		
8. ENVIRONMENT		
8.1 CURRENT (HAWAII/PUERTO RICO)	OPERATING: 2 FT/SEC AT SURFACE. SURVIVAL: 6 FT/SEC AT SURFACE.	OPERATING: 2 FT/SEC AT SURFACE. SURVIVAL: 6 FT/SEC AT SURFACE.
8.2 OPERATING SEA STATE	$H^{1/3} \leq 36$ FT AND DETACHABLE FOR $H^{1/3} \geq 12$ FT.	ASSUME DETACHED $H^{1/3} \geq 12$, NOT LIKELY TO SURVIVE $H^{1/3} = 36$ FT IF NOT DETACHED.

(1) REQUIREMENT NOT CERTAIN FOR PROTOTYPE, BUT IS CERTAIN FOR MODEL.

2.5 SUMMARY OF ANALYSIS

The discussion of the design approach presented in Section 2.4 highlights how the results of particular analyses were used in the design decision process. In this section, these analytical techniques and results for the 1/3 scale CWP will be explained further.

2.5.1 Bottom Weight Sizing

For the current specified it was established that the directly scaled model CWP would hang at a static angle of 26° from the vertical. Two considerations lead to the conclusion that this angle is too large: (1) as the angle increases the heave of the platform contributes to the transverse vibration of the CWP; and (2) when combined with the maximum dynamic angle, the articulation range of the CWP bearing at the platform interface can be exceeded. Based on these considerations, a static angle of 6° and an articulation range of plus or minus 30° was selected.

The CWP static angle is influenced both by the distributed submerged weight of the CWP and the bottom weight, if any. Figure 2.4 shows that to achieve a static angle of 6° a weight corresponding to a total skin thickness of 2.67 inches is required. Clearly this is excessive. Figure 2.5 shows the trade-off between the ballast weight and the uniform FRP weight.

2.5.2 Vortex Shedding Analysis

Stresses associated with vortex shedding are potentially the most damaging to the life of the pipe. It is desirable, in order to minimize shedding stresses, to only excite the lowest modal frequency of the CWP by expected shedding frequencies.

Figure 2.4 Variation of Pipe Angle with Wall Thickness
for a Uniformly Weighted CWP

Figure 2.5 Bottom Weight Requirements

$$\omega^* = \frac{2\pi \bar{U} S_0}{D}$$

where $\omega^* = 0.377$ rad/sec for a 10 ft diameter pipe and a Strouhal number of .3.

$$\sigma_v = \frac{MR}{I} \left(\frac{EI}{\rho_i} \right) \left(\frac{R}{I} \right)$$

$$\sigma_v = E \left(\frac{R}{\rho_i} \right)$$

ρ_i is the radius of curvature associated with the excited modal shape. S. Timoshenko, D.H. Young, and W. Weaver, Jr.'s, Vibration Problems in Engineering, 4th Ed., John Wiley and Sons, New York, 1974, gives the approximation for a hinge/free beam modal curvature as:

$$\rho_i = \frac{K_i^{-2}}{D}$$

$$\text{where } K_i = \frac{2i + 1/2}{2L} \times \rho_i$$

Therefore, L = pipe length (inches)

i = mode number

D = pipe diameter (inches)

K_i = wave number

$$\sigma_v = 2EK_i^2 R_i^2$$

The strumming stress is dependent upon the wave number and the pipe modulus. FRP fatigue data indicates that the fatigue properties of the fiberglass increase proportionally to increase in modulus. Hence, the most significant parameter is the wave number (K_i) where $i = 1, 2, 3 \dots N$.

The natural frequency of a hinged/free beam subjected to axial tension is:

$$\omega^* = \omega_i \sqrt{1 + \frac{Pl^2}{i^2 EI \pi^2}}$$

and

$$\omega_i = 2\pi C_i \sqrt{\frac{EI}{M_l^4}}$$

where C_i is a constant depending on mode number. For a monolithic pipe the equivalent wall thickness (t^*) could possibly range between .5 inches and 1.5 inches, and longitudinal moduli between 2×10^6 and $E = 5.2 \times 10^6$.

Consequently the possible range of bending stiffness EI is between 0.68×10^{12} and 5.37×10^{12} .

If $\omega_1^* = \omega_0$, then to excite the first natural pipe frequency would require that $(EI)_1$ must equal 2.47×10^{13} . Similarly, the required stiffness to excite the 2nd mode is $(EI)_2 = 1.825 \times 10^{12}$. Obviously, it is not possible to excite the first modal shape since the required stiffness exceeds the range under consideration. In fact, it would require at least a 7" thick pipe to achieve such a stiffness.

The most cost-effective approach is to use the maximum modulus available. This results in a minimum monolithic wall thickness of approximately .5".

The second-mode strumming stress is calculated from equation (2.5) to be 13000 psi. This stress is too high, thus strum suppression is required. The amount of strum reduction needed is determined by the fatigue analysis shown later.

2.5.3 Crippling Analysis

The design allowable buckling stress for a monolithic shell subject to bending is:

$$\sigma_{CR} = \frac{\gamma C_c E \frac{t}{R}}{\sqrt{1 - \nu^2}}$$

where C_c is the theoretical buckling coefficient (C_c is approximately 1) and γ is a correction factor to reduce theoretical results to values which can be used for design purposes (Figure 2.6). Figure 2.7 shows the relationship of this correction factor to solid wall thickness for a 10 ft diameter pipe.

Source:
 North American Rockwell Structures Manual,
 Section 9, Stability of Shells, p.9.33.16.

Figure 2.6 Design Correction Coefficient for
 Cylinders Subject to Bending

Figure 2.7 Buckling Correction Factor Versus Wall Thickness

If the allowable buckling strength of the pipe is equal to the ultimate tensile strength of the pipe, the ability of the laminate to develop its full load potential will not be compromised. Hence, good design practice dictates that:

$$\sigma_{CR} = \sigma_{ut}$$

Figure 2.8 shows the allowable beam bending buckling stress versus wall thickness based on the appropriate value of the correction factor. The graph clearly shows that in order to develop the full capability of the material, the wall thickness must be equal to or greater than .62 inches regardless of the material modulus.

For a composite sandwich structure where t_s is the skin thickness, t_c is the core thickness and t^* is the equivalent thickness for a monolithic pipe, we find that:

$$t_s = \frac{1}{2} \left[t^* - \frac{E_c}{E_s} t_c \right]$$

Using a value of t_s equals .17", the minimum core thickness t_c calculated from the above equation is .84" .

2.5.4 Crush Buckling Due to Wave Action

When a wave passes over the top section of the CWP two effects are present: (1) a linear pressure distribution in the undisturbed water within the CWP; and (2) a logarithmically decaying external pressure distribution. An analysis of the resulting pressure difference across the walls of the CWP, (which difference can potentially cause crush buckling of the CWP) is presented in Reference 7. The design pressure distribution corresponding to Hurricane loading, $H_{1/3} = 36$ ft. is shown in Figure 2.9. These results are

DESIGN ALLOWABLE CRITICAL BUCKLING STRESS
FOR BEAM BENDING (psi)

Figure 2.8 Critical Buckling Stress Versus Wall Thickness

Figure 2.9 Design Crush Buckling Pressure Distribution

based on the assumption that the CWP is not sealed to the platform sump, but is open at the top. If the CWP were sealed to the sump, the design pressure at the upper end would be on the order of 15 psi.

2.5.5 Transportation Analysis

An analysis was performed on the stresses associated with the towing of the CWP using the SAI-developed PIPTOW Code, Reference 7. Figure 2.10 shows results obtained for the 1/3 scale CWP test article. The analysis shows that the CWP can be towed at a wide range of headings in a sea state with $H_{1/3} = 6$ ft. The tow path of the pipe should be selected so that the CWP generally heads into the seas.

2.5.6 Stresses During Deployment and Recovery

The cold water pipe will be stressed as it goes through various angles with the vertical during initial deployment or during recovery. This section describes an analysis of the static stress from bending as the pipe changes attitude in a 2 ft per second current. This analysis considers the recovery cycle, beginning with the pipe in the near-vertical equilibrium position. The deployment yields the same result in reverse order.

The analysis is in two phases. The first begins with the pipe 6° from the vertical, where it is in equilibrium between the current forces (F) and the combined weights of the pipe (W) and the bottom weight (B). The constant-tension winch on the recovery vessel begins to take a strain on the line to the bottom weight. Phase I continues until the entire load of the bottom weight is carried by the recovery vessel winch.

The equation for equilibrium of the pipe is formed by taking moments about the pivot, where θ is the angle of the pipe from vertical, which results in:

$$B = \frac{1}{2} \left[\frac{F \cos^2 \theta}{\sin \theta} - W \right]$$

$$E = 5.27 \times 10^6 \text{ psi}$$

Towing Condition: Pipe Awash

Figure 2.10 Towing Bending Stress Analysis

the average axial tension in the pipe is;

$$T_A = \left[B + \frac{W}{2} \right] \cos \theta$$

The uniform transverse load acting on the pipe is;

$$Q = \frac{1}{L} \left[F \cos^2 \theta - W \sin \theta \right]$$

where

$$F = \frac{1}{2} \rho C_D L D \bar{U}^2$$

$$\rho = 2 \text{ slugs}; C_D = 1.2; \bar{U} = 2 \text{ ft. per sec.}$$

The analytical Model for the above loading is shown below:

For a beam subjected to axial tension and uniform load the max moment is given in R.J. Roark's, Formulas for Stress and Strain 4th Edition, page 151, Case 16, as:

$$M = Qj^2 \left(1 - \text{Sech} \frac{U}{2} \right)$$

$$\text{where; } j = \left(\frac{EI}{T_x} \right)^{\frac{1}{2}}$$

$$\text{and } U = \frac{L}{j}$$

(Note: Units are inches, pounds, and radians.

$$EI = 2.77 \times 10^{12} \text{ in}^2 \text{ lbs.})$$

The angle (θ) at which the bottom weight force is zero acting on the pipe is found from equation (2.9). Equations (2.10) and (2.11) are used to calculate T_x and Q respectively.

Table 2.8 summarizes Moment vs Angle for Phase I recovery.

Table 2.8
Phase I of Recovery

Angle of Pipe, θ , degrees	Bottom Weight, B, Kips	Bending Moment, M, in.-lb.
6	101	2.5×10^7
12	64.4	2.52×10^7
18	24.25	7.7×10^7
22	9	8.0×10^6

The second phase of the analysis, Phase II, begins with the pipe at 24.45 degrees from vertical. Fresh water is added to the top of the pipe, displacing the heavier sea water, which flows out through the valve at the bottom of the pipe. The static pipe bending moment starts at zero, increases to a peak, and then drops back to zero when the pipe floats at equilibrium in a horizontal position.

The equation of Equilibrium for the system shown is:

$$W \left(L - \frac{\ell}{2} \right) \sin \theta = \frac{FL}{2} \cos^2 \theta$$

where

$$W = 92.$$

From this equation the length of the sea water column (ℓ) as a function of pipe angle is:

$$\ell = L - \frac{1}{2} \sqrt{\frac{96L^2 - FL \cos^2 \theta \sin^{-1} \theta}{24}}$$

The maximum moment on the pipe as a function of angle is determined as follows.

The loading diagram is:

$$q_d = \text{drag unit loading} \\ = 48 \cos^2 \theta$$

$$q_w = \text{unit pipe weight in water} \\ = 92 \sin \theta$$

When θ is between 24.5° and 90° , the moment diagram has the following general shape:

From moment diagram

$$(q_w - q_d)\ell = q_d(x - \ell)$$

$$\therefore x = \ell \left(\frac{q_w}{q_d} \right)$$

The maximum moment is:

$$M = \frac{T_y L}{2} (L - x) = \frac{1}{2} (q_d L - q_w \ell) \left(L - \frac{q_w}{q_d} \ell \right) = \frac{1}{2} \left(q_d L^2 - 2q_w L \ell + \frac{q_w^2 \ell^2}{q_d} \right)$$

Table 2.9 summarizes Moment vs Angle for Phase II recovery.

Table 2.9

Phase II of Recovery

Angle of Pipe θ , degrees	Drag q_d lbs./ft.	Pipe Weight q_w , lbs./ft.	Length of Sea Water Column ℓ , ft.	Bending Moment M , ft.-lbs
25	39.43	38.9	832	6.33×10^5
27	38.1	41.76	645	1.64×10^6
30	36	46	500	4.7×10^6
45	24	65	196	2.64×10^6
60	12	79.7	75	1.51×10^6
80	1.45	90.6	7.7	1.96×10^5

Figure 2.11 shows the bending stress on the pipe for the entire proposed recovery procedure. Deployment is essentially the reverse of the procedure described above.

Figure 2.11 Bending Stresses During Recovery

2.5.7 NOAA/DOE Code Results

Once the preliminary CWP size and mechanical characteristics were established from vortex shedding, fatigue, and buckling considerations, the NOAA/DOE code, Reference 4, was used to determine the dynamic in-situ response of the CWP to the environment.

Both head and beam sea conditions were evaluated for a bottom weighted pipe. The results indicated very little difference in peak stresses and relative pipe/platform rotation with platform heading. Figure 2.12 shows the significant dynamic stresses for $H_{1/3}$ equals 12 ft. The peak dynamic stress ($1.35 \times$ significant stress) is one-half the ultimate tensile strength of the pipe. Hence the selected geometric and mechanical cross sectional properties appear to be an acceptable combination.

2.5.8 Fatigue Stresses

The materials and processes proposed for the composite CWP have been well characterized on previous NASA/DOE programs. Similar composites were used in the 150 ft composite NASA Wind Turbine Generator Blade developed by Structural Composites Industries, Inc., (SCI), and are presently being used to fabricate blades for the MOD-0A and MOD-1 Wind Turbines.

Table 2.8 shows predicted minimum ultimate tensile stresses for various laminate constructions. These predictions are generated by SCI using the SQ-5 computer program, which is a widely used program to predict laminate properties. Input data is based on experimental results from the previous projects mentioned above. Fatigue allowables are based on NASA tests for 90 percent longitudinal reinforcements. This data has been normalized in order to reflect the properties of different laminates.

Figure 2.13 presents the S-N curve for cyclic stress for factors of safety of 1 and 1.5. These correspond to a value of $E = 5.27 \times 10^6$ psi. A high value of E is desirable when loads are determined by vortex shedding.

Figure 2.12 $1/3$ Scale CWP Dynamic Responses, Bottom Weighted Pipe Design

Table 2.10. Normalized Fatigue Data @ 5×10^8 Cycles in Air

% Glass in Axial Direction	E_x	Steady Stress	Cyclic Stress
90	5.27×10^6	9000 psi	3800
70	4.43×10^6	7565	3194
50	3.59×10^6	6130	2588
30	2.74×10^6	4679	1975
10	1.89×10^6	3227	1362

Table 2.11. Design Ultimate Stresses Under Steady Load Conditions

% Glass in Axial Direction	E_x	Design Ultimate Tensile Stress	Design Allowable Shear Stress
90	5.27×10^6	33000	3280
70	4.43×10^6	27000	3382
50	3.59×10^6	21500	3382
30	2.74×10^6	16500	3382
10	1.89×10^6	10000	3382

Figure 2.13 Regression Analysis of NASA Windmill Fatigue Data

In general, the CWP is subject to cyclic stress superimposed on steady stress. The Goodman diagram, Figure 2.14, is used to estimate the cyclic stress which, when associated with a specified dynamic stress, will result in a life of $2 \cdot 10^8$ cycles, or alternatively, $5 \cdot 10^8$ cycles.

Figure 2.15 shows the number of cycles to failure for various levels of strumming stress. If strumming is not suppressed, the life of the CWP is measured in days. To achieve a life of 10 years, strumming amplitudes must be suppressed to 35 percent of their unsuppressed magnitude. One alternative for vortex suppression is spiral strakes. An appropriate design for these would be 1 ft in height and one rotation per 15 diameters along the pipe. Further design effort is necessary to specify the portions of the pipe on which the suppressors must be installed. In addition, consideration must be given as to how the suppressors will be removed for any parts of the test calling for measurements without suppressors.

Figure 2.14 Goodman Diagram from NASA Windmill Program

Figure 2.15 Model Strumming Stress Versus Fatigue Life

3. COLD WATER PIPE SYSTEM DESCRIPTION

This section contains detailed descriptions of the 1/3 rd scale model pipe, the test article, which was developed using the design procedures and loadings developed in Section 2. Also discussed is the proposed method and procedures for fabricating the test article including stringent quality control monitoring and testing. Proposed procedures for transport of the pipe from its assembly site to and from the test site, and the mechanics of deployment and recovery are also discussed.

3.1 COLD WATER PIPE SYSTEM DESCRIPTION

Details of the method in which the strength requirements of Section 2 will be met are shown in Figure 3.1. The material properties used in this design are summarized in the upper right-hand corner of the figure. The elastic modulus of the fiberglass skins will be achieved by using a transverse filament tape (TFT) containing 90 percent of the fibers in the longitudinal direction of the pipe. The skins will contain 65 percent glass by volume in an isophalic polyester resin. Previous experience has shown such material to maintain the critical axial ultimate tensile strength as measured by the appropriate test method involving rapid loadings. Compression strength will be much higher. It is the axial tensile fatigue strength which is to give the pipe its resistance to continued vortex-excited vibrations, but fatigue strength is also based on the ultimate axial tensile strength as discussed in Section 2.

In order to achieve the required resistance to both column and local buckling, the pipe wall has been designed with two equal-thickness fiberglass skins separated by a neutrally-buoyant syntactic foam core. The core consists of microspheres dispersed in a polyester resin base. Core thickness is greatest at the top, in part since the differential pressure is highest at the top of the pipe. Although the required buckling resistance changes exponentially from the top to the bottom, core thickness has been linearly varied from top to bottom.

At the top and bottom of the CWP test article, a transition ring will be employed between the FRP sandwich pipe and steel ends. Stainless steel pins will hold a tongue of fiberglass in a slot of the transition ring. Detail A of Figure 3.1 shows the design of the pipe termination, transition ring, and connection to the end cap. A detailed sizing of the end cap bolts and design of the end cap has not been completed at this time.

Figure 3.2 shows the arrangement of the pipe during the at-sea test. The sketch at the left side of the figure shows the relationship of the CWP test article, test platform, and retrieval line during data taking. The CWP test article is suspended below the platform on a spherical bearing, one-half of which is rigidly attached near the keel of the platform. The retrieval line is connected to a surface buoy, which in turn is tethered to the platform. The end caps at both ends of the pipe are of similar design, containing lightening holes sufficiently large to preclude throttling of the water in the pipe as the platform moves in a seaway.

The details of the spherical bearing attachment are shown at the right side of Figure 3.2. The spherical bearing will be used as the towing attachment and subsequently attached to the main hoist of the platform during deployment. The main hoist of the platform will keel-haul the bearing into place in the moon pool where it will be locked securely to the platform by locking pins driven by hydraulic cylinders.

Bottom weight details are shown at the center of Figure 3.2. The bottom weight is designed to be a steel shell filled with concrete approximately as shown. Final details of the bottom weight attachment to the bottom of the CWP test article have not been determined.

3.2 FABRICATING THE COLD WATER PIPE

This section contains details of the construction process used to build the test article, assemble it into one piece, and control its quality.

3.2.1 General Description of Process and Problems

The design and fabrication of the cold water pipe test article is of an advanced composite process. The advanced composite design is a low density fiber reinforced plastic made with high strength, high modulus fibers bound together with resin. The fibers most often used in this technique are carbon, graphite, boron, kevlar, glass and hydriized combinations. They are usually bound together with an epoxy, polyester or polyamide thermosetting resin; thermo-plastic resins are also used. Fibers are oriented specifically to provide strength and stiffness in an amount and direction needed depending on the forces and loads imparted by the operating environment. When the fibers are aligned directionally, the composites exhibit considerable anisotropy in strength, modulus of elasticity and thermal expansion characteristics. By properly stacking the laminate in unidirectional layers in which the fiber directions change from ply to ply, a variety of specific strengths, stiffness, and thermal expansion properties can be provided.

Because most structures are subjected to a combined loading, laminates are constructed by using interdispersed layers of plus or minus 40 degree and 90 degree plies. However, for the 1/3rd scale CWP test article the primary stresses occur from the cyclic vibration of the pipe in response to vortex shedding, resulting in high axial loads. Because of the predominant axial loading, most of the fibers for the CWP test article must be parallel to the major axis. Details on how this orientation is achieved will be given in the following subsections.

The available allowable stress of fiberglass is very much dependent on the relative direction of the fibers and the applied load. As an example, the charts in Figure 3.3 show the variations in tensile strength and elastic modulus for an angular ply layup for typical high strength carbon/epoxy composite as a function of the percentage composition of zero plus or minus 40 and 90 degree ply. These charts are based on a carbon fiber composite which will allow unidirectional composite tensile strength of 160,000 psi. Tensile modulus is 20 million psi with a density of 0.057 lbs per cubic inch. These charts are presented here only as a general representation of the variations in tensile strength that are available relative to the various angles of ply applied to the structure. It has been shown that the expected allowable stress for the 1/3rd scale CWP test article is approximately 22,000 psi. This indicates that it is unnecessary to use the more expensive carbon filament compounds to meet the required allowable stresses. However, if it is necessary to produce high tensile strength composite CWP structures, the technology is available.

3.2.2 Pipe Fabrication and Assembly

The design of the 1/3rd scale CWP test article point design was based on the assumption that the pipe would be fabricated at standard fiberglass pipe manufacturing facilities. A survey of the state-of-the-art capabilities of these facilities has shown that it is reasonable to assume that the pipe may be fabricated on a mandrel with a ten foot OD with sections approximately 40-60 ft in length.

3.2.2.1 Fabrication Schedule. An estimate of cost, schedule and shipping was based on a pipe ten feet in diameter fabricated in sections 40 ft long. The sequence is shown in Figure 3.4. The pipe sections will be fabricated on a mandrel individually, cured in place, then detached from the mandrel and shipped to the assembly location. The individual pipe sections will be fabricated in an assembly line process; i.e., each section will be fabricated on the mandrel, cured, removed from the mandrel and shipped to the assembly location on a sequential basis.

Modulus of Elasticity of Carbon/Epoxy Composite

Tensile Strength of Carbon/Epoxy Composite

Figure 3.3 Variation of Modulus of Elasticity and Tensile Strength with Fiber Ply Orientation

Figure 3.4 CMP Fabrication Steps and Assembly

As each section arrives at the assembly graving dock it will be joined to the preceding section until a total of 1000 feet of pipe length has been assembled.

3.2.2.2 Pipe Fabrication Description. Each 10 ft ID 40 ft long composite pipe section will consist of filament wound E-glass/resin layers fabricated and assembled around a polyester and ceramic sphere core. The skin section material will be glass/resin composite in which 90% of the structural fibers are oriented nearly parallel to the longitudinal axis of the pipe section. In order to fabricate the pipe sections on a rotating mandrel with primarily 90 degree fiber orientation to the circumference, it is necessary to lay the pipe up with tapes in which most glass fibers are orientated across the tape in the warp. The crosswire glass fibers make up 90% of the glass. The other 10% of the glass fibers are along the length of the tape in the woof.

Although the length of the majority of the fibers which are orientated along the longitudinal axis of the pipe may be only 12 to 18 in., the longitudinal stresses are transferred between tape sections by the resin that binds the composite structure together. Therefore, the completed skin laminate will structurally respond as though there were continuous fiber strands in the longitudinal direction.

The inner layer of the pipe will be layed up with the tapes of filament and resin until the desired thickness of the inner layer of the sandwich is completed. It may be possible to lay up the pipe skins using a system that applies more than one layer of tape at a time. When the inner skin is completed, pre-molded sections of the polyester resin and ceramic spheres will be placed over the inner shell and glued in place with resin. The next step will be to apply the outer shell of the sandwich construction in a manner similar to the inner layer. The pipe section will then be cured in place on the mandrel. When the section is cured, the ends will be cut and trued to have uniform thickness and flush ends in order to butt one section to the following section for assembly. At that time the mandrel will be extracted from the completed pipe section and the section then will be prepared for shipping.

The pipe section will then be lifted with a mobile crane using a spreader sling over the center of gravity of the pipe section. The section will be lifted to a railroad car and secured for shipping to the assembly site. In all, 25 pipe sections of 40 foot length will be necessary to complete the 1000 foot long, 1/3rd scale CWP test article.

Each pipe section will be somewhat different in that the core thickness decreases from 1.84 to 0.84 in. from CWP top to bottom. The technique of varying the wall thickness for all of the 25 sections will be accomplished by varying thickness of the molded core used to lay up the pipe sections. All pipe sections except for the decrease in the thickness are exactly the same. The exception to this is the top and bottom sections which will be fabricated with specific inserts for attachment to the transition rings that will be used to connect the pipe to the gimbal and bottom weight.

3.2.2.3 End Section Termination. Figure 3.2 shows details of the CWP transition ring. The termination sections will be the transition points between the FRP of the pipe and an adapter ring. The transition is accomplished by first placing a high density foam plug at the top of the polyester resin and ceramic sphere core. Unidirectional glass roving is placed around this high density plug. The unidirectional glass roving will be used to build up the terminus area. After curing, holes will be drilled for insertion of bronze bushing bars, which will be held in place by resin. Longitudinal glass roving then is wound around the unidirectional glass fiber to make the termination end of the pipe. Each bronze bushing bar is then drilled to match the transition rings.

3.2.2.4 Pipe-Joining. Each pipe section will be fabricated in 40 foot lengths placed on flat cars and shipped to the assembly area. The assembly area will be constructed to accept the pipe sections, provide a foundation for aligning the pipe sections, and provide facilities for joining them. The pipe sections will arrive individually on flat cars, they will be transferred to the graving yard, and will be moved onto the alignment foundation. As each section arrives it is aligned with the previous section and a lap joint technique is used to bind the two sections together.

The joining process is referred to as a butt-and-strap field joint technique. The butt and strap field joint technique consists of aligning each pipe section so that the surfaces are in contact throughout the circumferential perimeter of the pipe. Gaps in the joint are filled with resin putty and then overlaid outside and inside with fiberglass and resin. The butt-and-strap technique of joining pipes is only one of several ways that the pipe sections may be joined.

Two other examples of field joints that can be used for the CWP test article are external hardness welded joints and internal hardness welded joints. These two techniques require that the pipe section ends be fabricated as a lap instead of a butt joint. The external hardness welded joint provides a perfectly smooth inner surface of the CWP that minimizes the frictional losses of the fluid. The internal hardness welded joint has an inside fiberglass resin overlay causing a perturbation in the smoothness of the inner wall. This may cause increased friction of the fluid flow. The butt and strap joints have an overlaying fiberglass and resin layer both on the outside and the inside. However, the inside surface layer is much smaller and presents only a small perturbation on the inner wall. In the case of the OTEC 10/40 MW Pilot Plant CWP, this would contribute insignificantly to the added drag of the flow within the pipe.

The butt and strap joint was selected because it contributes to ease of fabrication for the individual pipe sections and the joining process. If an internal hardness or external hardness welded joint were employed, the mandrel for the fabrication of the individual pipe sections would be more complicated and the pipe sections would be more costly and difficult to fabricate.

Once all 25 forty-foot pipe sections are assembled in the graving yard, resulting in a 1000 foot long CWP, the center of the pipe will be filled with fresh water to make it neutrally bouyant, the graving yard will be flooded and the pipe will be prepared for towing and deployment.

3.2.2 Assembly Site

The site chosen must permit the assembly of the 1000 foot pipe ready to be towed to the at-sea test site. The choice must consider the design of the pipe, the method of fabrication, the method of obtaining material samples as the fabrication progresses, the installation of the measuring devices, and the way of getting the pipe into the water.

It is anticipated that the assembly area will be constructed on a beach as a graving yard. The graving yard will be constructed so that once the pipe is assembled and the end caps are in place, it may be flooded and the cold water pipe towed out for deployment.

3.2.4 Control of Quality

The objectives of the tasks in this section are twofold; first, to ensure that the properties of the materials are at least equal to those specified in the design and secondly, to establish accurate properties which can be correlated with the measured structural response and thus increase the level of confidence in the design methodology. The factors of safety employed in this design are lower than those that would be routinely used for fiberglass tanks, culverts, and outfalls. This makes it necessary that extreme care be taken during fabrication to ensure that the structure can carry the allowable stress. In addition to being sure that the properties meet or exceed the minimum levels required by the design, particular attention must be paid to elimination of weak spots so that parts containing these weak points can either be rejected from the final product or repaired to meet the minimum specified requirements.

In the manufacture of fiberglass reinforced plastic it is particularly important to maintain accurate records of all components which go into the final part. This tracability allows accurate identification of parts subsequently found to contain defective materials. Each of the finished

spools of pipe will contain a weather and sea resistant identification plate showing the following information:

- date of manufacture
- the resin type, manufacturer, and batch number
- shell laminate sequence and thicknesses
- the total weight of the spool
- the location of the center of gravity

3.2.4.1 Materials Quality Control. The resin used in this design is a polyester thermosetting resin. The resin will be clear and transparent with suspended solids not deviating more than 1 percent from the content specified by the resin manufacturer. The resin will be supplied from the least possible number of individual batches in order to simplify the control of product quality. All records of resin production and quality control will reference batch identification. Resin batch identification, date of resin production, resin type and manufacturer's name will be recorded on each container of the resin used in the manufacture of the pipe. Catalyst and promoter will be of the type and amount recommended by the resin manufacturer for use with their resin. The resin will be used to make up putty for all joint filling operations, particularly those used in the butt and strap joints to join spools of pipe together. The resin will be tested as per ASTM D2471 "Test for Gel Time and Peak Exothermic Temperature of Reacting Thermosetting Resins" within one week prior to the start of use of each batch. The viscosity of the resin solution will be tested per ASTM D1824 "Standard Test Method for Apparent Viscosity of Plasticols and Organosols at low shear rates by Brookfield Viscometer" and will be between 350 and 550 centipoise. Viscosity tests are also required within one week prior to the use of each batch of resin in manufacture of the pipe.

Glass fiber reinforcement used will be commercial grade corrosion resistant or silicate type E glass. All glass fiber reinforcing will have a silane type surface finish or approved equal and binder which is specifically compatible with the resin to be used. This surface finish should allow the maximum chemical bonding between resin and glass. The glass will be supplied from the least possible number of individual batches to simplify control of product quality. All records of glass production and testing will be

referenced to the glass batch number. All packages of glass will be individually identified by manufacturer, batch identification, weight and yield.

Each batch of glass will have a letter of analysis containing the following information:

1. Yield of glass products (ends)
2. Type and amount of sizing
3. Dry tensile strength (roving only)

Records will be maintained which reference glass reinforcement batch identification to each part manufactured. Upon receipt of each batch of glass, reinforcement tests shall be made per ASTM D2343 "Test for Tensile Properties of Glass Fiber Strands, Yarns, and Rovings Used in Reinforced Plastics" and shall be within the specified limits.

3.2.4.2 Production Quality Control. The fabricator of this pipe will show past experience in the field application of large diameter filament wound pipe. In addition, all plant superintendents and shop foremen will have experience in the fabrication and assembly of large industrial fiberglass reinforced plastic equipment. The fabricator will verify his ability by a description of construction projects to which he has successfully applied techniques similar to those proposed for fabricating this pipe.

Positive means will be employed to monitor resin and glass usage while the filament winding is in process. The system must allow adjustments of the glass to resin ratio during manufacturing. The glass to resin ratio by volume will be maintained between 60 and 70 percent. Laminate thickness will be maintained within plus 10 percent of that specified in the design, unless other tolerances are shown on drawings, while total item weight will not exceed the design value used in calculating performance. Tolerance on fabrication of the individual pipe spools will be maintained to plus or minus 1/4 inch on the length.

The assembly of the individual spools into a completed pipe will be conducted in a sheltered area protected from the weather and contaminants. The joint will not be exposed to direct sunlight during layup. All joining materials will be gathered and premeasured prior to the start of the joining procedure. Voids between the joints will be filled with resin putty.

3.2.4.3 Inspection. All parts manufactured for use in the cold water pipe will meet the requirements of ASTM D2563 "Practice for Classifying Visual Defects in Glass Reinforced Plastic Parts", grade 2, for visual defects.

A sample of each separate run of syntactic foam core will be retained and tested for density and compression strength. Values obtained for the mean density and the lower 90 percent confidence level for compression strength on accepted core will equal the values used in the design within 1 percent on density and 5 percent on compression strength.

3.2.4.4 Testing. Approximately eight one-foot square coupons will be cut from each spool as it is manufactured during the trimming operation. Four coupons will be cut from the trimmed ring on each end of the spool located at 90° intervals to each other. From each of these coupons test samples will be cut and subjected to the following tests:

1. Wall thickness will be measured for the inner and outer laminates and the core.
2. Hardness tests will be made on both the inner and outer surfaces using a Barcol impressor model GYZJ 943-1, calibrated at 2 points in accordance with ASTM D2583 "Standard Method of Test for Indentation Hardness of Plastics by Means of a Barcol Impressor". Where low readings are encountered readings will be rechecked by taking 10 or more additional readings and calculating the average value after discarding the two lowest and two highest readings.

3. The completed sandwich sample will be tested per ASTM C297 "Tension Test of Flat Sandwich Constructions in Flatwise Plane".
4. Glass content will be determined by the average of five specimens per ASTM D2584 "Tentative Method of Test for Initial Loss of Cured Reinforced Resins". Both the inner and outer surfaces of each coupon shall be tested using a one-inch square sample.
5. Flexure tests shall be conducted on the completed sandwich construction using ASTM C393 "Flexure Tests of Flat Sandwich Construction".

If the properties required are not met in the average of the test data obtained from the tests listed above, an additional set of tests using more coupons cut from the remainder of the trimmings on the spool may be performed with double the number of tests, to determine if the spool could possibly meet the design requirements. The average of all tests will equal design values or the spool will be rejected.

3.2.5 Schedules, Cost and Instrumentation

Schedule control during the fabrication of the pipe and assembly will be through active use of CPM. The fabricator will have experience in applying CPM to at least two large projects which were controlled throughout their duration by this means including periodic systematic review of the schedule.

Costs will be maintained during the manufacture in accordance with the DOE Work Breakdown Structure, Reference 8. Outer skin strain gage instrumentation will be installed prior to the final layer of the outer laminate. At least two strain gages will be installed for each measurement location to allow for the possibility of damage during shipment and handling. Those gages installed throughout the entire thickness from the inner skin laminate, including the inner and outer layers of the inner skin, the core and inner and outer layers of the outer skin will be installed near one end of the

spool so that cabling can be led out of the laminate and passed through the joint overlay.

3.3 HANDLING THE COLD WATER PIPE

The following sections discuss towing, deployment, and survival procedures.

3.3.1 Towing to the Test Site

Figure 3.5 shows a proposed towing scheme for moving the cold water pipe from its assembly point to the test site. First the CWP will be moved into the water. One tug will do all the towing with a second tug maintaining a slack line to the pipe, and available on standby in case a problem should be encountered. Procedure will start by first flooding the pipe with fresh water employing the bladders at top and bottom ends to seal off the fresh water from sea water outside the pipe. When the pipe is flooded with fresh water it will float with its top slightly above the surface and thus will not be subjected to large stresses associated with a sail area exposed to the wind and waves. The spherical bearing housing for suspension of the CWP will be employed as part of the towing apparatus. This can be seen in detail A between the transition cap and towline at the right hand side of the figure. In order to minimize the shock loadings on the pipe a synthetic line is proposed for use as the towline. The synthetic line will be attached to a traction winch which will also serve to reduce shock loadings through an automatic tensioning feature.

3.3.2 Deployment and Recovery

Figure 3.6 shows the deployment and recovery scheme. After the pipe has reached the test site the towline attached to the spherical bearing will be removed and replaced by the cable going to the deployment winch on the platform. An auxillary winch at the side of the test platform will also be connected to the spherical bearing. The tug then will move around and take a strain on the bottom weight so that the flotation for the bottom weight can be

Figure 3.6 Deployment and Recovery Scheme

removed. The procedure of moving the cold water pipe into place beneath the platform then involves opening the two seawater valves at top and bottom of the pipe. As seawater enters the pipe, displacing the fresh water, the pipe end will swing down from the surface away from the service tug until it is suspended below the platform. At this point the deployment winch pulls the spherical bearing up into the housing and it is locked in place. The bottom weight support line is then bouyed off and the auxillary winch line is slacked and the plugs inside the pipe are deflated. Plugs are removed by pulling their attached inflation hoses and the deflated plugs up the pipe and out through the access holes of the upper end bell. The recovery scheme is the reverse of this and is also outlined in the figure.

3.3.3 Detachment and Reattachment for Survival

The survival procedures are outlined in Figure 3.7. Since the 1/3rd scale model cold water pipe is only designed for waves essentially 1/3rd the maximum expected waves in a 100 year period at the test site, there must be some provision made for the pipe to withstand waves above 12 feet since they are likely to occur. The model test has therefore been designed to permit detaching the CWP test article from the platform during conditions in which the waves have been forecast to exceed 12 foot significant wave height. In this event the pipe will be hung on a cable from a support buoy which has much smaller mass than the platform. The top of the pipe will be at a substantial depth below the water surface to get out of peak currents and wave perturbation. This is intended to reduce the loads on the CWP to the point where even 100 year storms do not cause a failure loading on the pipe.

3.3.4 Assessment of Wear and Tear

Following the return of the cold water pipe to its storage location, samples will be taken at 10 equal intervals along the length of the pipe to determine what has happened to the FRP during exposure to the ocean and stress environment. Four one-foot rectangular coupons will be cut from the pipe at each of the 10 locations. From these coupons tests for tensile strength, ASTM

C297, will be conducted to determine the ageing and effect of cyclic stress on the remaining strength properties of the fiberglass laminate. Samples of the syntactic foam will be tested for the quantity of water which has penetrated the matrix using ASTM D570 "Water Absorption of Plastics; Test For". Finally, fatigue tests at 20, 30 and 40 percent of ultimate tensile strength will be conducted using ASTM D3479 "Tension Fatigue of Oriented Fiber, Resin Matrix Composites; Test For".

4. INSTRUMENTATION SYSTEM DESCRIPTION

4.1 INSTRUMENTATION PLAN

Proper instrumentation and recording systems must be selected to provide measurements necessary to characterize the performance of the 1/3 rd scale cold water pipe. This requires specifying the variables to be measured, estimating their range and the accuracy to which they must be measured, determining the number and locations of the measurements, selecting the transducers, and developing a recording and playback system to store and retrieve the data in a format usable for comparison with engineering calculations and/or computer code simulations.

The instrumentation plan philosophy must be to obtain quality data with a high degree of certainty. An at-sea experiment is a difficult and expensive undertaking, and the opportunity to repeat the experiment is unlikely. Therefore, it is imperative that a sound plan be developed and executed. This must include quality control in the sense of including ways to ensure that data is indeed being recorded, and in the correct range. In general it is not expected that analysis will take place on the platform. However it should be recognized that during the test operation, certain information must be available in near real time to make procedural decisions.

The test is to be conducted to demonstrate all phases of the CWP fabrication, towout, deployment, operation, and retrieval and to validate the CWP design computer codes. Design data must be obtained during all phases of the test program, and some instrumentation will serve multiple phases. Measurement details, data processing requirements and cost estimates are provided in the following sections.

Platform response to ocean waves will be measured in terms of accelerations and angular rates which are translated into motions. Response Amplitude Operators (RAOs) will be determined for six degrees of freedom and will be compared with values predicted by the NOAA/DOE code. These measurements should be performed for the platform alone before the CWP has arrived at the test site, and then for the combined platform/CWP. Loads transmitted from the CWP to the platform must be measured along with the relative motion for proper design of the platform/CWP interface.

The loads on the handling lines during CWP towout, deployment, detachment/reattachment, recovery and towback are required for comparison with analytical models of these phases. The CWP transportation and deployment techniques proposed for the fiberglass CWP are unprecedented at this scale, so detailed validation of the handling loads models must be conducted to gain confidence in the approach.

The majority of the test measurements will be made on the CWP itself. Bending stresses are the result that is of primary interest, and these are determined by measurements of the CWP wall strains (axial and circumferential). All phases of CWP transportation, deployment, operations and recovery must be included for comparison with theoretical predictions; real time monitoring of key locations must be included to guide test implementation. During the CWP towout and towback, pipe heading and qualitative response are to be monitored to insure safety. When the CWP is being deployed, CWP position (angle, depth) must be measured. During CWP operations vortex shedding correlation lengths need to be established with and without vortex strumming suppressors.

While it would be desirable to obtain CWP response measurements when detached from the platform, this is not considered practical. Rather, provision will be made for disconnecting and securing instrumentation cables for reconnection upon CWP reattachment. Oceanographic data will continue to be recorded.

4.2 MEASUREMENT REQUIREMENTS

The major phases involved in the one third scale test program are:

- Site Characterization (i.e., quantifying environmental conditions)
- Platform Response without pipe
- CWP Towout
- CWP Deployment
- Combined Platform/CWP Response
- CWP Detachment/Reattachment
- CWP Recovery
- CWP Towback
- Post-Test Assessment of Pipe Condition.

Measurements must be provided of the oceanographic environment, of the loads on the handling lines, and of the response of the platform and the CWP. Table 4.1 indicates the required measurements during each phase.

Oceanographic parameters required are: directional wave spectra at site, the current profile at the site, the site meteorological conditions (especially winds), and the site surface currents. Between the CWP fabrication area and the test site some of these measurements are necessary at reduced accuracy for transit. Oceanographic measurements are required for the purpose of relating the environmental loads to the calculated and measured platform/CWP response. The oceanographic instrumentation should be in place before the platform is brought to the site to avoid costly delays.

TABLE 4.1 MEASUREMENT REQUIREMENTS

Phase \ Item				
	CWP	Lines	Platform	Oceanographic
Site Characterization	-	-	-	x
Platform Response	-	-	x	x
CWP Towout	x	x	x	x
CWP Deployment	x	x	x	x
Operation	x	-	x	x
Detachment/Reattachment	-	x	x	x
CWP Recovery	x	x	x	x
CWP Towback	x	x	-	x

4.3 DETAILED MEASUREMENT SPECIFICATIONS

Specific measured variables, stations, types of transducers, accuracies and sampling rates are described in this section. Oceanographic measurements are described first, followed by the operating measurements and the towout and deployment measurements (which employ some of the operating equipment). Details of the recording and playback system are described in the next section.

4.3.1 Oceanographic Measurements

Table 4.2 lists the oceanographic measurement requirements. Wave height spectra will be recorded at three or more stations near the platform. The relative positions of these will be controlled, so that data from them can provide directional wave spectra. Conventional wave riders will be employed. Data from them will be collected on the platform and stored along with the response data. A sampling rate of 3 hz will adequately characterize the wave frequencies.

Surface currents will be measured for the trajectory of the CWP during towout and towback. A preliminary measurement will be made during site characterization for planning purposes and repeated prior to CWP towout. Surface drogues can be used with radio receivers at shore based locations.

Profile measurements are required at one station to resolve current velocity and direction as a function of depth. A single current meter will be traversed along the 1000 foot depth from the platform once per day of testing. Ocean current turbulence measurement is required for the vortex shedding environment specifications. At least once per week the current meter will be positioned at 250 feet below the surface for sufficient lengths of time to measure the fluctuations in velocity. Both power spectra and RMS velocity fluctuations are required.

A simple meteorological station will be established on board the platform to record wind speed and direction, dry and wet bulb temperatures, and barometric pressure. The purpose of these measurements is to evaluate

Table 4.2 Oceanographic Measurement List

Parameter	Location	Range	Accuracy	Sampling Rate	Remarks
Wave Height Spectra	3 or More Sensors Near Platform at Test Site			3 samples/sec	Three site stations provide directional spectra.
Current Profile	Adjacent to Platform	0-2 fps $\pm 20^{\circ}$	± 0.02 ft/sec $\pm 10^{\circ}$	1 profile/day	
Water Turbulence	3-100 Upstream 6-20 Downstream	TBD	TBD	3 samples/sec 3 samples/sec	Vortex shedding requires definition of turbulence by instrumentation such as a hot-wire anemometer
Surface Currents	From Construction Area to Site	0-2 fps	0.2 fps	As required	Establish currents during site characterization.
Wind	1 Station Aboard Platform @ 10 m Above Waterline - wind speed - wind direction - wind turbulence	Standard	Standard	Analog 10 minute average and gusts	
Air Temperature	- temp (dry bulb) - temp (wet bulb)	Standard	Standard	Log measurement twice per day	
Barometric Pressure	- barometric pressure	Standard	Standard	Log measurement twice per day	Note conditions when storm conditions arise

wind loads on the platform and to maintain an accurate record of meteorological conditions associated with each test.

4.3.2 Platform/CWP Operating Measurements

Table 4.3 lists the platform/CWP measurement requirements. The large number of strain gages is required to resolve the dynamic bending stresses of the CWP. Figure 4.1 indicates the gage locations.

CWP axial strains will be measured at twelve stations along four generators (i.e., lines at 90 degrees to each other) to resolve the dynamic bending of the CWP. Modal shapes will be reconstructed using Fast Fourier Transform (FFT) routines; up to sixth mode response is expected. (The FFT routine should be operable aboard the platform.) A sampling rate of 1.5 Hz will adequately resolve the sixth modal frequency.

CWP circumferential strains will be measured at two stations along four generators using strain gage rosettes (note: 8 channels provide part of axial requirements). The gages provide a measure of torsion as well as ovaling strains.

In-depth strains will be measured at two locations along the CWP wall to provide details of the fiberglass sandwich beam response. Measurements will be made of the strains inside and outside both skins and in the core material (Figure 4.2). Test data will be compared to analytical models of the composite beam response to provide model verification and to aid in sandwich beam design.

The 1/3 rd scale CWP will be made up of pipe sections using butt and strap joints for cost and schedule reasons even though the final full size design may be of a single piece construction. Axial strain measurements will be made at two joints to provide details of the joint effectiveness in transferring the loads. The strain gages will be located on a single generator just outside the joint strap, near the end of the pipe section under the strap and on the exterior of the strap (Figure 4.3). Sampling rates will be the same as for the CWP bending strains.

Table 4.3 Towing and Deployment Measurement List

Parameter	Location	Number of Channels	Range	Accuracy	Sampling Rate	Remarks
CWP Axial Strain	Six stations along two lines at 90° to each other	12	$< 10^{-3}$ in/in	± 1% F.S.	Analog	Record on FM tape and output on strip chart.
CWP Heading and Modal Shape	16mm movies from helicopter	1	-	-	-	Radio communication with test director.
Tow Line	Load cells on tug attachments	1 each	10 kips	± 5% F.S.	Analog	Backup tug has strip chart only.
Towout Speed	Current meter suspended from tug	1 each	3 fps	± 0.1 fps	Analog	Backup tug has strip chart only.
CWP Depth During Deployment	Static pressure from 3 stations	3	500 psi	± 1 psi	Analog	FM tape and strip chart.

Depth (Ft)	Angle from Bow (deg)			
	0	90	180	270
80	SA	SA, PD	SA	SA
160	SA	SA, PD	SA	SA
240	SR, SI	SR, SI, PD	SR, AC	SR
320	SA	SA, PD	SA	SA
400	SA, SJ	SA, SJ, PD	SA	SA
480	SA	SA, PD	SA	SA
560	SR	SR, PD	SR, AC	SR
640	SA	SA, PD	SA	SA
720	SA	SA, PD	SA	SA
800	SA	SA, PD	SA	SA
880	SA	SA, PD	SA	SA
960	SA	SA, PD	SA, AC	SA
SA = Strain Axial SI = Strain Indepth AC = Accèleration SR = Strain Rosette SJ = Strain in Joint PD = Dynamic Pressure				

FIGURE 4.1 CWP GAGE LOCATIONS

FIGURE 4.2 SANDWICH WALL INDEPTH STRAIN GAGE LOCATIONS

FIGURE 4.3 JOINT STRAIN GAGE LOCATIONS

As a backup measurement of the CWP dynamic bending response, biaxial accelerometers will be used to measure horizontal accelerations at 3 stations along the CWP. Power spectral density analyses will be used to determine the frequency content of the accelerations for comparison with the strain gage output and the computer code calculations.

Dynamic differential pressure measurements will be made at twelve locations along a single generator in order to determine the vortex shedding correlation length. Measurements will be made with and without the vortex strumming suppressors in place. Power spectral density profiles will determine the vortex shedding frequency, and correlation coefficients between the various stations will be integrated to determine the correlation length. These measurements are required to characterize the vortex loads and the performance of vortex strumming suppressors in order to determine their effect on the CWP fatigue lifetime.

Platform response with and without the CWP attached will be determined from 3 axis accelerometers and 3 axis gyros located at the platform center of motion (near the c.g.). Power spectral density data of the response measurements will be divided by the power spectral density data from the oceanographic measurements to provide RAOs in six degrees of freedom. The platform metacentric height (or heights) will be determined for the various load conditions during the test period.

Equally important are the loads and angular deflections of the platform/CWP interface. The design of the instrumentation system is dependent upon final interface design, but provision must be made for resolving the loads in three linear directions and the angles in pitch and roll. If the CWP is allowed to rotate freely in yaw, this angle will be measured.

A master time code generator will be included in the data acquisition system.

4.3.3 Measurements During Towing and Deployment

CWP towing and deployment has been called the most uncertain phase of the test operation, and therefore, real time test data must be obtained to aid decision making and test data must be recorded for post test analyses. Since it would be too costly to provide two complete data acquisition systems, and it is not practical to employ the system on the platform, a separate analog FM tape recording system with a strip chart display will be used on the primary tug during towing and deployment. Table 4.4 lists the towing and deployment measurement requirements.

Both qualitative and quantitative measures will be used to monitor the CWP dynamic modal response during towing. Twelve channels of axial strain gage data selected from the existing transducers will be observed on the strip chart to insure the CWP stresses do not exceed design allowables. An observer following the CWP in a helicopter will describe the CWP motion (heading, mode shapes, amplitude) to the test director and will take 8 mm movies for documentation.

Line tension and towing speed will be monitored on the strip chart to match prescribed test conditions and recorded on the FM tape recorder. To provide for the unlikely event of loss of the primary tug, the secondary tug will also use a strip chart.

During CWP deployment (swing down) axial bending stresses will continue to be monitored and recorded. In addition static pressures at three locations along the CWP will be measured to determine the CWP vertical location.

4.3.4 Cables

A single, multiple conductor, armored underwater cable should connect each station to the data acquisition system. The bottom end of each cable should be terminated in a spider of underwater connectors which mate to the sensor connections. Spares should be provided.

Table 4.4 Platform/CWP Measurements List

PARAMETER	LOCATIONS(S)	# OF CHANNELS	RANGE	ACCURACY	SAMPLING RATE	REMARKS
CWP AXIAL STRAIN	4ea at 12 stations along CWP	40	$\pm 3 \times 10^{-3}$ in/in	$\pm 0.03 \times 10^{-3}$ in/in	1.5 samples/sec	Primary measurement
CWP CIRCUMFERENTIAL STRAIN (ROSETTE)	4ea at 2 stations along CWP	24	$\pm 3 \times 10^{-3}$ in/in	$\pm 0.03 \times 10^{-3}$ in/in	1.5 samples/sec	Resolves ovaling and torsion
INDEPTH WALL STRAIN	5ea at 2 stations	10	$\pm 3 \times 10^{-3}$ in/in	$\pm 0.03 \times 10^{-3}$ in/in	1.5 samples/sec	Provides details of sandwich beam response
JOINT STRAINS	3ea at 2 stations	6	$\pm 3 \times 10^{-3}$ in/in	$\pm 0.03 \times 10^{-3}$ in/in	1.5 samples/sec	Provides details of joint performance
HORIZONTAL ACCELERATION (2 axis)	3 stations along CWP	6	$\pm 2 \text{ft/sec}^2$	---	1.5 samples/sec	Backup measurement primarily for frequency content
DYNAMIC DIFFERENTIAL PRESSURE	12 stations along CWP	12	$\pm 10 \text{ psi}$	---	0.3 samples/sec	Frequency content is important parameter
PLATFORM PITCH, ROLL + YAW	3 axis gyro at CG	3	$\pm 30^\circ$	$\pm 0.5^\circ$	1.5 samples/sec	Required for RAO
HEAVE, SURGE & SWAY	3 axis accelerometer at CG	3	$\pm 2 \text{ft/sec}^2$	$\pm 0.02 \text{ft/sec}^2$	1.5 samples/sec	Required for RAO
CWP/PLATFORM INTERFACE LOADS	3 axis load cell/strain gages	3/10	TBD	$\pm 2\% \text{ F.S.}$	1.5 samples/sec	Accommodation must be made for torsion
CWP/PLATFORM ANGLE	2 axis pendulumeter	2	$\pm 20^\circ$	$\pm 0.5^\circ$	1.5 samples/sec	Maximum angle important for interface design
PLATFORM/SKSS LOADS	1 per line	3-8	TBD	$\pm 2\% \text{ F.S.}$	1.5 samples/sec	---
TIME OF DAY	1 master clock part of DAS	1hr,Min,Sec	---	$\pm 0.01 \text{ sec}$	---	This time required for header of each data block
PLATFORM/SKSS POSITION	2 per line	6-16	TBD	$\pm 2\% \text{ F.S.}$	1.5 samples/sec	Azimuth and Vertical Angle

A disconnect mechanism must be built into the cable termination at the data acquisition system. After the CWP has been lowered under the platform, the connection to the analog recording system will be broken and the connection reattached to the primary ADP system on the platform. Also, during CWP detachment/reattachment the instrument cables will be disconnected and stored for later reconnection.

4.4 DATA PROCESSING REQUIREMENTS

It is simply not acceptable to store the data taken on site for later examination. Some (near) real time information is necessary to make test decisions. In addition, the expense and uniqueness of the test require that the data be verified before the test is completed. Therefore, an automatic data processing (ADP) system will be employed to digitize the data and store it on magnetic tape. A minicomputer and line printer should be in available (not necessarily on board the platform) to process the data on a daily basis and provide hard copy for on site investigation. The data system should incorporate signal conditioning, calibration, circuit checks and threshold electronics.

For maximum industry compatibility, test data should be recorded on IBM standard nine track magnetic tape at 800 BPI. Both the raw transducer data and the calibration factors should be placed on the tape along with a common time code. Redundant tape recorders should be supplied to avoid the loss of data.

An environmentally controlled work area is required for the data acquisition system. This can be accomplished most easily with an air conditioned instrumentation van (say 8' by 20') with a wall mounted outside junction box. The van can be fully equipped and checked out before installing on the test platform. Provision should be made for maintenance, spares, redundancy and safety equipment to provide maximum assurance of returning the necessary data.

To characterize the dynamic response of the CWP, the test data should resolve the sixth bending mode. The present design sixth mode frequency is 0.3 Hz, and a conservative estimate of the number of data points necessary to

resolve a frequency is five. Therefore, the strain gage measurements should be sampled 1.5 times per second. This is the highest sampling rate required since the vortex shedding frequency is 0.06 Hz and the platform will not respond significantly to waves with frequencies greater than 0.3 Hz. Detailed sampling rates are provided in the measurement lists (Tables 4.2-4.4).

Engineering data acquired during the test should be processed and displayed in a format most convenient for comparison with the applicable analyses - for CWP operation the NOAA/DOE codes are to be employed. Platform RAOs will be compared to the predictions of the COTEC code, Reference 4. (One of the inputs to the code is the platform metacentric height, so the value of this must be made available to permit comparisons to be completed.) CWP responses will be compared to the CWPFLX code, Reference 4, calculations in terms of modal shapes and frequencies, the static deflection of the CWP in the current and vortex induced strumming deflections. Response of the coupled platform/CWP to ocean wave inputs will be compared to the XOTEC code, Reference 4, calculations in terms of platform, CWP and interface statistical response.

Some 110 channels of data are anticipated. This includes 40 channels of axial strain, 24 channels from strain gage rosettes, 8 channels of in depth wall strains, 4 channels of additional joint strains, 6 channels of acceleration, 12 channels of dynamic pressure, all on the CWP. In addition there will be 6 channels of data for platform response, 3 channels from the interface loads, 2 channels from interface angles, 3 channels from the wave riders, 1 channel from the current meters and a time code generator. Also needed will be records of wind direction and velocity, and force and direction measurements on the mooring lines.

5. PLATFORM AND STATIONKEEPING SUBSYSTEM REQUIREMENTS

The coupled motions of the platform/CWP/SKSS induce stresses on the CWP. Therefore, the platform to be used in the At-Sea Test Program must be selected with due attention to its influence on test results. This section discusses: criteria for selection, candidate platforms, and platform station-keeping.

5.1 CRITERIA FOR SELECTING A PLATFORM

Criteria for selecting the platform for the test can be grouped into those concerned with:

- Feasibility of accomplishing desired measurements
- Capability to accommodate needed equipment and personnel
- Availability and scheduling, including time for design and accomplishing any needed modification
- Costs.

5.1.1 Beam and Draft of the Platform

The platform size should be chosen so that the platform will respond, in a way appropriate to the test needs, to a seaway reasonably to be expected in the selected test location and season. As a generality, too large a platform will not roll and pitch with amplitudes large enough for a desirable test, and too small a platform would not be sufficiently seaworthy.

The interaction of forces between pipe and platform depends on a number of interrelated factors. These interact in a complex way as the pipe and barge, each with its own resonant characteristics, react to wave and current forces. It is considered that computer-code modelling can provide guidance in the appropriate choice of a platform for the test.

For example, the motion and stresses of the combined platform and pipe depend on the relative masses of the platform and pipe. (In principle this includes entrained masses of water but these are not considered here in order to simplify the discussion.) To put the matter in simplified terms, the pipe may tend to "drive" the platform motions if the platform has a relatively small mass, and conversely, if the platform is relatively large, it may "drive" the motion of the pipe. At this stage of the platform selection process it seems desirable for the 1/3rd scale model tests to come somewhere near matching the currently planned 10/40MW plant with respect to the ratio of platform mass to pipe mass. This ratio can be thought of as the beginning of a size criterion. Even though computer codes presumably could be validated for any mass-ratio that turns out to be convenient in the test, the feeling persists that if the masses are so matched, the results will be more dependable. (Note that this is not so much a matter of scaling up the results as it is of testing a configuration in model size that corresponds to the full scale prototype.)

The results of a short series of systematic computer runs are shown as a start toward developing bounds on the relative masses as a criterion. These runs explore both the pipe and barge motions and the forces as the mass is changed.

The At-Sea Test setup currently envisioned includes pivoting the top of the cold water pipe at the platform. Without going into the construction details, the pipe would act as if it were suspended from a gimbal structure. Such an arrangement would mean that no heeling or pitching moments would be transmitted between the pipe and the barge. Forces and a (presumably minor) twist moment about a vertical axis can occur. Vertically, the weight and inertia of the pipe would act on the hinge structure. The amount of weight can be altered to some degree by changing the buoyancy of the pipe. The vertical inertia of the pipe, that is, its resistance to vertical motion as the connector of the pipe to the platform tries to move vertically in heave, may result in substantial forces. Corresponding inertia forces in the fore and aft directions and across the platform will also act, but are predicted to be considerably smaller.

Figure 5.1 shows predicted effects of changes in platform size upon the maximum platform motion and pipe maximum stress. For the analyses shown, the pipe characteristics remained unchanged while the platform characteristics were varied. Table 5.1 summarizes the computer code inputs for the results shown in the figure. The base case platform around which the changes were made is a rectangular barge 250 ft long, 50 ft wide and 4.2 ft draft. (The dimensions were selected to result in a displacement of 1500 tons. This corresponds to the use of a platform approximately like the SEACON, a vessel owned by the U.S. Navy and considered generally suitable as a platform for the test.) Hinged to all 5 variations of the base case barge is the same pipe: 10.17 feet in outside diameter, 1000 feet vertically, and weighing 340 kips in air and 176 kips in water. (Such a pipe is similar to the design shown in Sections 2 and 3 of this report, except that there is no allowance for a bottom weight on the pipe.) Since the displacement of the barge alone is 3360 kips, the mass of the barge is about 10 times that of the pipe (i.e. $3360 \div 340 = 9.9$). The corresponding ratios for actual designs vary considerably. For the APL/JHU design using a concrete pipe and barge, the ratio is 37.

Figure 5.1 Predictions of Effects of Changes Of Beam and Draft of Platform in six-foot waves.

TABLE 5.1
Inputs to Computer Code for
Exploring Effects of Platform Size

Assumed Platform is rectangular barge. Runs made with a four point mooring*, and with 10' diameter FRP pipe, 1000 ft long, weight 176 kips. Platform length 250 ft. Platform Radius of gyration taken to be same as for a solid.

Inputs for Base Case

Beam = 50 feet	c.g. location 15' above keel
Draft = 4.2 feet	(Displacement = 1500 tons)

Computations were made for beam seas for: (1) "design spectrum" of waves: 12-foot significant height, 7.95 second modal period and (2) for a spectrum we might expect in a test: 6-foot significant height, 5.64 second modal period.

Inputs were changed for sensitivity analysis exploration of effects of size:

- Beam Change to 2/3 and 3/2 of base case, with draft unchanged at base-case value.
- Draft changed to 2/3 and 3/2 of base case, with beam unchanged at base-case value.

* The initial results indicated that the mooring selected had no effect, and subsequent computer runs were made with no mooring line tension.

The changes in beam and draft of course change the mass, and concurrently change the metacentric height. The vertical location of the center of gravity height was kept constant relative to the waterline.

The two parts of the figures show, along the horizontal axis, the changes in beam and draft as percentages of the base case. The vertical axes are: the maximum heave, in feet above an average still-water datum; the roll angle in degrees from vertical; and the highest stress in the pipe, in thousands of pounds per square inch. (The highest stress was in the lower sections of the pipe.)

As will be seen, for the 6 foot waves of the figure the changes in roll and heave are not excessive. Roll angle was decreased by an increase in beam, as might be expected, and was increased by an increase in draft. Presumably if the mass were changed by changing both beam and draft together, there would be only small changes in roll angle. The heave figures vary between about 1 1/2 and 2 1/2 feet and are thus substantial percentages of the half-height of the six-foot waves. Changes in draft have very small effect on the heave, and increases in beam decrease the heave somewhat.

An important prediction is the maximum stress in the pipe. Increases in draft increase the stress; increases in beam decrease it. Neither change is judged to be a matter for concern in the ranges of beam and draft for which the prediction is made.

The conclusion drawn from these preliminary predictions is that a fair amount of latitude is acceptable in selecting the platform mass relative to the mass of the pipe.

5.1.2 Platform Freeboard

A low-freeboard platform design, which might involve having a wave wash over part of the deck might be acceptable from a seaworthiness point of view. However, from the point of view of a computer code, the effects of such a breaking wave are not now taken into account - - the code assumes as much

freeboard as needed. Thus one criterion to be met by the platform is that the freeboard not be exceeded in sea-states suitable for the usual measurement. Since a 12' high sea for the 1/3rd scale model corresponds to a 36 foot (design hurricane) sea for the full size, a freeboard that can accomodate a 12' wave (trough to crest) is adequate for the test. This freeboard, of course, must be available after the platform is loaded with the pipe and stationkeeping subsystem. A 6 or 7 foot still-water freeboard might be expected to be acceptable. (This does rule out certain deck-cargo barges as candidates unless they are modified to increase freeboard.)

5.1.3 Special Features of the Platform

Various quite radically different shapes have been proposed for the full scale OTEC platform and no specific shape is clearly ready for selection. A platform of the spar type, however, is probably not readily available in a size applicable to this test, and for the first exploration is not being considered. This leaves barge-like candidates and ship shaped candidates. Both can be modeled in the computer code, which is also flexible on proportions.

The platform will clearly need some way of working with and holding the pipe. A moon-pool is appropriate and is common in drill rig ships. To provide a moon-pool in some of the types of candidate platform vessels may not be structurally feasible (as has already been found for one otherwise appropriate candidate platform). The forces between the pipe and the platform can be substantial, as discussed earlier. The platform structural elements must be accessible, so a structure can readily be built which will transmit the wave-induced forces to and from the pipe.

For testing of the computer codes it may be desirable to include changes in the characteristics or configuration of the pipe and platform. The design of the experiment has not been decided upon as this is written. Two kinds of changes in the platform that might well be tried are, (a) changes in the trim or displacement, presumably attainable by taking on ballast and, (b) changes in the vertical center of gravity or in the rotational inertia, which

might be accomplished by shifting loads. The implications of such changes on the choice of platform are clear: ballast tanks and the capability to use movable equipment, such as a truck crane, may both be desirable. For either of these kinds of change in platform parameters it would seem probable that the smaller the platform chosen the better. This is because a relatively small platform is more likely to permit substantial changes in the parameters themselves, but also because for a relatively smaller platform there may well be a greater chance that measurable changes in response would occur.

The notion of separating the functions from the platform was also explored with respect to cold-water pipe deployment and handling equipment. If a crane on a separate work boat can handle the over-the-side loads during the cold water pipe keel-hauling operation, the platform can be smaller than would otherwise be necessary. A preliminary computation of stability was made for a 45' wide barge using a 30 ton capacity truck crane to lift a 20 ton load whose centerline is 15 feet from the side of the barge. Such a load would capsize the barge. For the still larger forces that can be expected with a 10 ft cold water pipe, it seems clear that either a separate work boat is needed, or a platform with a beam now estimated as 50 feet or more, but which would need to be explored in more detail with stability as a selection criterion.

Pipe loads handled through the moon-pool must be supported by a structure on the platform itself and cannot be delegated to a workboat. The required load capacity depends on the pipe buoyancy in the design and the procedures for initially placing the pipe, and for detaching and re-attaching in storm emergencies. The sample design of the test article cold water pipe shown in section 2 of this document uses a bottom weight. The required hoist capacity for the pipe itself plus its bottom weight, allowing for dynamic loads, is estimated to be over 250,000 pounds.

5.1.4 Cargo Capacity

To properly set up the requirements for the platform, even a first order estimate involves a preliminary look at what activities are necessary.

The measurements of strains, positions, accelerations in a multitude of positions in the pipe and its connections will require a substantial volume and weight of equipment (more than one hundred channels). Assuming that this equipment is (as has been customary in previous at-sea strain tests for quite different projects) located in trailers or in 8x8x20 cargo-container type boxes, the minimum needs may be estimated as:

- 1 trailer for a diesel electric power supply for equipment and lights
- 1 trailer for strain-gage records
- 1 trailer for wave and motion records
- 1 shelter trailer for personnel.

For the first estimate it is assumed that the measurement records are largely automatic and that a crew of five or six men per shift will be able to maintain and service the equipment. Note that this does not include personnel who analyze the data or those whose role is to decide among alternatives to be scheduled as the test period progresses. For the sake of simplicity it appears logical to support this equipment on the platform. There need be no "hotel load" on the platform. Alternatively, a platform might be used that could house and feed three shifts plus observers.

5.1.5 Availability

The desired test timetable calls for testing in January 1981. Estimates will be required of time required for the various steps needed before the test can be begun (e.g. how long it would take to install a hinged pivot in a moon-pool, together with suitable fasteners for joining the pipe to the pivots.)

5.2 CANDIDATE PLATFORMS

This section describes the results of a preliminary search for possible candidate platforms. The task of exploring for candidate platforms has gone forward concurrently with work on criteria for selecting from among them. The concurrent nature of the work meant that while the criteria were

being developed, the range of dimensions and characteristics of possible candidates necessarily remained somewhat broad. Certain common-sense guidelines, though, keep the list of candidates from being inconveniently large. For example, cost and time constraints preclude considering a specially-built platform at this time.

5.2.1 Potential Candidate Platforms

Table 5.2 lists potential candidate platforms, with information on their location and characteristics.

Data collected on charter costs is included in the table. For example the Navy's SEACON, a 260 ft x 48 ft x 5 ft draft vessel of approximately 1500 tons displacement, and complete with deep water mooring facilities, a moon-pool, and cranes, would charter for \$8500 per day including messing and berthing for 50 people. A commercial oil rig, 204 ft x 34 ft beam, called the "Black River" is now under conversion on the West Coast. It is available. It is designed to lift 1500 feet of oil drilling pipe and can also anchor in deep water. Its charter costs would be on the order of \$5500 per day. Messing and fuel would be in addition, and support for instrumentation would have to be installed.

5.2.2 Other Platforms

The process of simultaneously studying both the possible candidates and the needs (criteria) for the test has resulted in looking at and rejecting certain candidate platforms. An outline description of these is shown as Table 5.3. The use of so large a vessel as the Glomar Explorer (618 feet long, displacement 63,000 tons) was suggested. However, her large size appears inappropriate for the test because in response to small waves she would not move with enough amplitude for good measurements. The others listed also were found unsuitable.

TABLE 5.2 POTENTIAL CANDIDATE PLATFORMS

Identification	Location	Owner	Length ft.	Beam ft.	Draft: Depth	for Displacement Draft Long tons	Moon Pool?	Deep-water Mooring	Other	Cost	Comments
"SEACON"--a Navy ocean construction "barge"	East Coast; Florida or Norfolk	U.S. Navy, NavFac	260	48	3.8 light 7.9 Full Load	1,013 2,300	yes, 16'x32'	yes, with positive navigation	Small Crane: Instrum., Voith-Schneider propellers. Sonar Navigation.	\$8500 per day incl. berthing & messing for 50	Freeboard with light load, 5'. Schedule as now drawn up would permit use in October 1980. Rolls badly in a beam sea.
"BLACK RIVER" - a small oil drill rig	West Coast	Associated Marine Services, Seattle	204	34			yes	yes, six 3000 pound anchors	has draw works	\$5,500 per day not incl. fuel or food	Vessel now in yard (apparently being converted)
"OCEAN HYRRICANE" or "OCEAN TEMPEST" oil drill rigs	Gulf of Mexico	Ocean Drilling & Exploration Co. New Orleans	378	70	draft 29'		yes 22'6" square	8-30,000 lb anchors for 500 ft. (but additional chain can be put in a workboat)	has 1,000,000 Pound Derrick	Cost not discussed	Owners commented that they find it undesirable to commit vessels that far ahead.
U.S. Army DeLong barges with "spuds". Barges designed as portable piers	Some near Norfolk, VA. others at other Corps elsewhere	U.S. Army (Transp. Corps)	Large "A" 300' Small "B" 150'	80' 60'	depth 13' 10'		consid. feasible to install	no	Large has ten six-foot caissons. Small has six six-foot diameter caissons.		No current peacetime commitment for the barges at Ft. Eustis.
Various Steel Barges	various	Commercial & U.S. Army	about 130'	about 30'	8'	Capacity around 400 to 600 tons	consid. feasible to install	no		\$200 to \$300 per day	An estimate from Foss Company, Seattle for \$300/day.
Composite structure assembled from surplus "LASH" barges	Richmond, California	Levin Metals Corp.	each barge 61'	31'		Empty wt. 81.5 long tons	yes	no	Each barge has a coaming & deck-like cover	Buy barges outright at \$12,500 to \$20,000 each	3 to 5 barges could be assembled together in a way that leaves a moon pool. (Barges could be transported to Hawaii as LASH barges aboard Farrel Line LASH ships)
U.S. Army, Beach Discharge Lighter "JOHN U.D. PAGE"	Hawaii	U.S. Army (Transportation Corps.)	338'	65'	Depth 21' Mean draft 7'	Light 1548 tons full load 4126	no	no	Voith-Schneider (Vertical axis prop.)		Currently in use with Troops in Hawaii but substitute craft are available.
ALCOA "SEA PROBE"	Panama City, Fla. (laid up)	Alcoa	243	50'	9½'	1120	14' X 30'	no	Aluminum Hull 250 ton crane Voith-Schneider propellers	\$13,000 to \$15,000 per day	Dynamic Positioned. Has Television cameras and instrumentation. 8 years old.
Mini-OTEC Barge	Hawaii	Lockheed and State of Hawaii	120' O.A., 105' LWL	33.9 ft.	Depth 12' mean draft 4.3'	268 LT operating	space for moon pool but not at c.g.	no	no propulsion	cost not discussed	Used for mini OTEC. This involved a CWP attached to a buoy, with a connector to the barge.

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U.S. Army DeLong barges with "spuds". Barges designed as portable piers	Some near Norfolk, VA. others elsewhere	U.S. Army (Transp. Corps)	Large "A" 300' Small "B" 150'	80' 60'	depth 13' 10'		consid. feasible to install	no	Large has ten six-foot caissons. Small has six six-foot diameter caissons.		No current peacetime commitment for the barges at Ft. Eustis.
Various Steel Barges	various	Commercial & U.S. Army	about 130'	about 30'	8'	Capacity around 400 to 600 tons	consid. feasible to install	no		\$200 to \$300 per day	An estimate from Foss Company, Seattle for \$300/day.
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Mini-OTEC Barge	Hawaii	Lockheed and State of Hawaii	120' O.A., 105' LWL	33.5 ft.	Depth 12' mean draft 4.3'	268 LT operating	space for moon pool but not at c.g.	no	no propulsion	cost not discussed	Used for mini OTEC. This involved a CWP attached to a buoy, with a connector to the barge.
OTEC-1 Platform (Conversion from T-2 Tanker)	OTEC-1 Test in Hawaii	Department of Energy	523'	58'	24' Draft	16470 L.T. Displ.	has gim-balled connection for pipe	has single point mooring 4500' water	Selfpropelled; has equipment for pumping water from CWP	\$18000 per day including crew, data acquisition equipment & personnel, and shore base support.	Available between end of OTEC-1 one MW heat exchanger test and scheduled dry docking of ship.

TABLE 5.3 OTHER PLATFORMS

Identification	Location	Owner	Length ft.	Beam ft.	Draft Depth	Characteristics Displacement Long tons	Moon Pool?	Cost	Comments
Submersible-Support Research Ship	West Coast	C.R. Ward & Associates Sidney, B.C.	191	45	17	-	no	not available	Owner sees no way that structure could be altered to provide moonpool.
Glomar Explorer	West Coast	Under charter to Lockheed	618	115	46'	63,000		Charter is \$4,000,000/year (i.e. \$10,960 per day)	Judged far too large
Floating Drydock	Oregon	Zidell explorations, Inc. Portland, Ore.	150	42	12	-	no	Charter cost \$5,000/month (i.e. \$167 per day)	Not stressed for ocean towing.
Sea Technology (semi submersible). Platform	Long Beach, Calif.	Deep Oil Technology, Irvine, Calif.	10' diameter 120' be- tween centers of columns		Height 66' normally		yes	\$5,000/day (note: half the total cost of test was in modifications)	Semi-submersible not considered appropriate for present test. Was used for test of 5' pipe; probably not too small for a 10' pipe. Has hardware and software for analysis of measurements.

5.3 STATIONKEEPING SUBSYSTEM REQUIREMENTS

This section of the report contains information necessary to develop the design of a stationkeeping subsystem or mooring for the 1/3 scale CWP at-sea test. No completed design example is available. The first subsection which follows discusses the OTEC SKSS program and its present state. The second section presents design criteria for the 1/3 scale SKSS. This is followed by a section on design loads and a design procedure which can be followed to convert the design criteria into a final design.

Three types of 10-40 MW Pilot Plants are now being considered for the Department of Energy OTEC Program. First is the moored plant located near shore, transmitting electricity over a subsea cable to the shore. Secondly, a grazing plant is considered located in the tropics where optimum thermal gradients exist. The third concept is a land-based plant which obtains water from a deep section of ocean near shore and generates electricity on shore. The moored-near-shore electric power generating plant is the leading candidate among the three concepts of Pilot Plants. For this concept two types of design are being considered. One is a barge type platform while the other is a spar buoy shaped platform.

Two contracts are presently underway to develop mooring designs for each of the two near-shore electric power generating plants. The two contractors are M. Rosenblatt and Son, Inc., and Lockheed Ocean Systems. Each of these contractors has now completed a preliminary design for each of the two types of near shore platforms. These designs are available in draft reports, References 9 and 10. Background information for these designs is in References 11 and 12. The designs for the spar platform are still considered beyond the state-of-the-art and require some additional developments. Mooring for a barge type platform is relatively conventional, the design approach being similar to moorings being used in the offshore oil industry for lay barges and drilling platforms.

The design by Lockheed Ocean Systems calls for an 8 point moor. It has been sized for installation on a steeply sloping seabed off Punta Tuna, Puerto Rico. In this Lockheed design two mooring winches are attached to each corner of the Pilot Plant barge, with lines from each leading to a pair of anchors in series on the line. (With two lines from each of the four corners, and two anchors on each line, there are 16 anchors.) Chain is used on the bottom near the anchor to resist chafing on the bottom and also at the winch to allow use of a wildcat on board the barge for an active tensioning system.

The M. Rosenblatt & Son design uses a 12 point mooring system. The main reason Rosenblatt went to a 12-point moor, as opposed to the 8-point moor used by Lockheed, is that their design estimated greater environmental loadings on the platform. In order to keep the size of chain and wire rope within available maximum sizes the additional mooring lines were required.

Both of the current SKSS designs are still in a state of flux as more knowledge is gained both on the environment in which the designs must be deployed and on methods of translating environmental conditions into loadings on the mooring systems. At present, the greatest variation in design loadings results from different predictions of the wave drift force applied to the platform and mooring during survival conditions. NOAA has funded an evaluation of various methods of evaluating wave drift force in the design stage and this is contained in Reference 13. Studies of the environment at proposed OTEC Pilot Plant deployment sites are continuing.

The objectives of the SKSS design for the At-Sea Test Program is to meet the following requirements:

1. Maintain the selected platform and CWP test article within specified limits of excursion at the selected test site considering the expected environmental conditions.
2. Perform as nearly like the 10/40 MW Pilot Plant SKSS as possible within practical engineering limits and reasonable costs.

3. Ensure that the CWP test article performance is not biased from modeled conditions by a dominant SKSS.

5.3.1 Design Criteria

1. The mooring will be a 4 point moor to permit the measured results to be used to simulate either an 8 point or 12 point moor for the prototype.
2. Active tensioning will be required on the mooring winches. Bridles to one adjoining corner of the platform are required, to allow training the platform through a 90 degree arc.
3. The mooring will be designed to withstand forces up to a sea state 6 (i.e., 15 foot significant wave height) with the cold water pipe attached. This is one sea state higher than the modeled survival condition for the OTEC site being considered. After the cold water pipe has been detached from the platform, the SKSS must then be capable of withstanding the current, wind, and wave loads up to the 100 year recurrence-interval storm.
4. Wind force and moment will tentatively be estimated using the coefficients of wind drag developed by Aage, Reference 14. Before the final sizing of SKSS components, wind tunnel or equivalent tests will be conducted, in order to measure the actual platform forces and moments as functions of angle of attack.
5. Wind, current, and wave loads will be assumed co-planar unless it can be shown that higher loading on the SKSS results from reasonable combinations other than co-planar.
6. In calculating the maximum current conditions the wave-induced current and normal current shall be superimposed.

7. The SKSS will be designed to require no maintenance during the entire At-Sea Test Program. Thus it must survive exposure to the marine atmosphere and immersion for a period at least equal to six months.

5.3.2 Estimation of SKSS Loads

In order to properly design the stationkeeping system, it is necessary to develop a reliable set of design loads. Environmental loadings on the system will arise from three sources: wind, currents, and wave drift forces. This section describes the analyses necessary to properly develop these loads for the 1/3rd scale model system.

5.3.2.1 Wind Loading. The normal method for determining the wind loading on the surface platform consists of application of Morison's equation (drag term only). Thus,

$$F = 1/2 \rho W^2 A C_d$$

where ρ is the density of air, W is the wind speed, A is the projected area of the platform above the waterline and C_d is the drag coefficient. It is recommended that a drag coefficient of 0.8 be used for the wind loading.

According to Giannotti & Associates (Reference 12), wind loads normally constitute only about 10 percent of the mooring loads in extreme seas and quite a bit less in operational sea conditions. For the size barge and relative design sea conditions considered for the at-sea test, wind loadings will probably constitute only 3-5 percent of the total mooring loads.

5.3.2.2 Current Loading. For low sea states, current loading is generally the major source of mooring loads for a barge-shaped platform. As with the wind, the loading for currents can be estimated by use of Morison's equation for both the submerged portion of the platform and for the CWP. In addition to this, however, current loadings on the mooring lines themselves may also be significant, particularly if vortex-induced strumming occurs. Finally, vortex

shedding from the pipe itself can greatly increase the apparent drag on the system. It is recommended that an enhanced drag coefficient of 1.8 be used in Morison's equation to account for this increase. This is the same drag coefficient used in the structural design of the test article.

5.3.2.3 Wave Drift Forces. As indicated above, the wave drift forces on the platform are probably the least well understood of the hydrodynamic forces. Reference 13 evaluates a number of models used for the prediction of wave drift force loadings on the OTEC SKSS.

The mean wave drift force on a barge may be estimated using coefficients derived from potential theory and from empirical studies. Lockheed Missiles and Space Co. (Reference 9) used the following to determine wave drift forces:

$$F = 2 \int S(w)F(w)dw$$

where:

$S(w)$: Wave energy spectrum
 $F(w)$: Drift force coefficient

The analysis of wave drift forces for a full scale pilot plant is presented in the above referenced Lockheed Report.

5.3.3 Design Procedure

Figure 5.2 diagrams the SKSS design procedure. The procedure shown in this figure can be used to integrate the design criteria given in section 5.3.1 and the methods of calculating loadings shown in 5.3.2 into a completed preliminary design. Once the preliminary design has been selected, the details of shackles, swivels, hardware selection, and the mooring deployment and recovery scenarios are developed. These are to be included in the completed SKSS design package.

Figure 5-2 SKSS Design Procedure

6. REFERENCES

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