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**LOCALING CONSIDERATIONS FOR THE OCEAN THERMAL ENERGY
CONVERSION (OTEC) COLD WATER PIPE AT-SEA TEST PROGRAM**

By
Lawrence A. Kahn

August 1980

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SCALING CONSIDERATIONS FOR THE OCEAN THERMAL ENERGY CONVERSION
(OTEC) COLD WATER PIPE AT-SEA TEST PROGRAM

ORIGINALLY PREPARED BY ORI, INC.

December, 1979

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FOR

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PREFACE

This Technical Memorandum was originally published by ORI, Inc. in December 1979 as Government-Furnished Information, for research groups submitting proposals for the OTEC Cold Water Pipe At-Sea Test Program sponsored by NOAA/DOE. The original publication has been reviewed and found to contain a number of textual errors and inconsistencies. Specifically, the equations presented to scale flexural responses were erroneous although the parameters tabulated (e.g., Table 3) were correct.

Based on the aforementioned review, NOAA's Office of Ocean Engineering requested Giannotti & Associates, Inc. to edit the original report without changing the author's presentation and format. Equations have been corrected and clarification statements included¹ in this annotated version of the original publication (see Errata, Page 20).

L.A. Vega, Ph.D.

Giannotti & Associates, Inc.

Editor

¹Editorial annotations are included within brackets [].

ABSTRACT

An at-sea experiment will be conducted on an Ocean Thermal Energy Conversion (OTEC) Cold Water Pipe (CWP) to demonstrate the present technology related to CWP design and to provide data for validation of the NOAA/DOE analytic code. In this report, various scaling considerations and modeling relationships are defined between the prototype OTEC CWPs and a scaled-down model at-sea test CWP. Utilizing these scaling laws, [in addition to other analytical design tools] a designer would be able to develop a test CWP for an at-sea experiment. Examples are provided of a 1/3, 1/6, and 1/10 scale CWP to indicate how the scaling laws may be applied.

SCALING CONSIDERATIONS FOR THE OTEC
COLD WATER PIPE AT-SEA TEST PROGRAM

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NOMENCLATURE

<u>Symbol</u>	<u>Unit</u>		<u>Definition</u>
A	- ft^2	-	cross sectional area
C_{Du}	- none	-	unsteady drag coefficient
C_{Ds}	- none	-	steady drag coefficient
C_L	- none	-	unsteady lift coefficient
C_m	- none	-	cylinder added mass coefficient
C	- none	-	mass coefficient, $\bar{\rho}/\rho_w - 1 + C_m$
D	- ft	-	pipe diameter
E	- lb/ft^2	-	Young's modulus
g	- ft/sec^2	-	gravitational acceleration
G	- lb/ft^2	-	shear modulus
I	- ft^4	-	cross sectional moment of inertia
I_α	- slug-ft	-	polar moment of inertia, $I_p \cdot \bar{\rho}$
I_p	- ft^4	-	polar cross sectional moment of inertia
K_1	- none	-	Constant defining boundary in pipe extension condition
K_2	- none	-	Constant defining boundary condition in pipe torsion
k	- ft	-	average roughness height
L	- ft	-	pipe length
m	- slug/ft	-	mass/unit length (including added mass and internal water)
S	- none	-	Strouhal number
t	- ft	-	pipe wall thickness
T	- lb	-	added tension
U	- ft/sec	-	internal flow velocity
V	- ft/sec	-	external flow velocity
[W/L]	- lb/ft	-	pipe weight in water/unit length
[W]	- lb	-	pipe weight in water
x	- ft	-	displacement at the top of the pipe is an inertial reference frame
γ	- lb/ft^3	-	weight density (ρg)
λ	- ft	-	pipe vibration amplitude

λ	- none	-	linear scale ratio (prototype length/model length)
ν	- ft ² /sec	-	kinematic viscosity
$\bar{\rho}$	- slug/ft ³	-	average pipe mass density
ρ_w	- slug/ft ³	-	mass density of water
ω_e	- rad/sec	-	exciting frequency
ω_n	- rad/sec	-	pipe undamped natural frequency
ω_s	- rad/sec	-	vortex shedding frequency

1. INTRODUCTION

The National Oceanic and Atmospheric Administration's (NOAA's) Office of Ocean Engineering (OOE) is providing technical program management support to the Department of Energy (DOE) in developing ocean engineering technology for the Ocean Thermal Energy Conversion (OTEC) Program. A major objective in the support of the program is the development of the technology to design cold water pipes (CWPs). In meeting this objective, NOAA has undertaken the development of analytic design tools. In order to advance the capabilities of these tools a CWP analytic code validation effort has been initiated. As part of this effort, CWP structural response data will be collected during model testing. NOAA's model test effort is divided into two programs, a laboratory and an at-sea testing program. This paper presents scaling considerations for the CWP At-Sea Test Program.

The OTEC Cold Water Pipe At-Sea Test Program is a project sponsored by NOAA's Office of Ocean Engineering which includes the design, fabrication, instrumentation, deployment, testing and demobilization of a one-third scale model of a CWP and platform for the OTEC 10/40 MW Pilot Plant. The one-third scale model CWP has a diameter of 10 feet and a length of 1000 feet which scales a 30 foot diameter, 3000 foot long prototype CWP anticipated for use in the OTEC 10/40 MW Pilot Plant. The Test Program is designed to address the following objectives:

1. Demonstrate, at a reasonably large scale, the technology of fabrication, transportation, deployment, and operation of a realistic OTEC CWP;

2. Collect data directly applicable to [assessing] the existing NOAA/DOE CWP Analytic Code, Reference 1.

As part of NOAA's cold water pipe at-sea test program, a point design of a one-third scale CWP has been developed. Reference 2 contains a discussion of scaling specifically applied to the one-third scale at-sea CWP test article but does not discuss the general scaling relations derived from dimensional analysis.

The purpose of this report is to discuss the general scaling relations and considerations applicable to OTEC cold water pipe modeling. These are discussed using the premise that the CWP will be modeled using linear elastic beam theory as in Reference 2. The general scaling laws are developed [actually developed in Reference 3 and reproduced here] using dimensional analysis techniques and govern the geometric and physical parameters, environmental loadings, and structural responses of model CWP's to potential OTEC environments. Numerical examples are presented for full, one-third, one-sixth, and one-tenth scale CWP's to demonstrate how the scaling laws can be applied to the design of model cold water pipes.

2. SCALING CONSIDERATIONS

The main point of model testing is to simulate the loadings and responses to be expected on a prototype. Problems of scaling CWP and platform responses arise due to difficulties in scaling physical processes such as boundary layer flow and hydrodynamic damping simultaneously with gravity wave and inertial responses [i.e., Reynolds scaling vs. Froude scaling]. A prime consideration in having an at-sea test as large as one-third scale is that as the size difference between prototype and model becomes smaller, differences between the two sets of hydrodynamic properties decrease and the model test will simulate full scale responses more closely. In this section the scaling considerations for the at-sea test article are presented and equations are provided to enable a designer to develop an initial design for the modeling process.

SCALING LAWS

Scaling laws are presented for the model pipe flexure, extension and torsion. In addition, scaling law considerations are discussed relative to platform and environmental scaling.

Pipe Scaling Laws

Pipe Flexure - In order to develop a model test for CWP flexural response, the distribution of the parameters $\rho_w AU^2/L$, EI/L^3 and W/L along the pipe axis should be similar for model and prototype. The following non-dimensional parameters must be identical for both model and prototype (Reference 3):

- a) Flexure forces/inertia forces $[(\frac{1}{C}) (\frac{E}{\rho_w V^2}) (\frac{I}{L^4}) (\frac{V}{\omega_n D})^2]$
- b) Weight/flexure $[\frac{WL^2}{EI}]$ [see Errata]
- c) Exciting frequency/natural frequency ω_e/ω_n
- d) Internal flow force/flexure $\rho_w AU^2 L^2/EI$
- e) Added tension/flexure $\frac{TL^2}{EI}$
- f) Steady drag force/inertia $[(\frac{C_{Ds}}{C}) (\frac{D}{L}) (\frac{V}{\omega_n D})^2]$
- g) Unsteady drag force/inertia $[(\frac{C_{Du}}{C}) (\frac{X}{D})]$
- h) Vortex shedding frequency/natural frequency $S (\frac{V}{\omega_n D})$
- i) Unsteady lift forces/unsteady drag forces $[(\frac{C_L}{C}) (\frac{D}{X}) (\frac{V}{\omega_n D})^2]$

[N.B. Only equations a, b, c, d, and e are utilized here]

The hydrodynamic coefficients are functions of Reynold's number (VD/ν) , Keulegan-Carpenter number $(2\pi\Lambda/D)$, and roughness coefficient (k/D) . Since the model and prototype pipes lie within the turbulent flow range for nearly all environmental conditions, the hydrodynamic coefficients for model and prototype are assumed to be equal for the considerations contained in this paper [i.e., same drag, lift and added mass coefficients].

Pipe Extension - The primary consideration of pipe longitudinal extension is that of exciting a resonant frequency in the pipe, thus greatly influencing the heave motion of the platform/pipe system. The governing equation for longitudinal extension natural frequency is dominated by the structural properties of the pipe and can be expressed by:

$$\omega_n^2 = \frac{K AE}{m_p L^2} \quad [\text{see Errata}]$$

Pipe Torsion - The governing equation for pipe torsion is similar to that for extension and the natural frequencies can be expressed as:

$$\omega_n^2 = \frac{K_2 I_p G}{I_\alpha L^2}$$

Other Scaling Considerations

Platform Scaling - Numerous experiments have been performed related to the prediction of ship structural response to the environment via scaled model tests. The basic difficulty in performing scale model experiments is the inability to simultaneously scale the gravity dominant (Froude) and viscous dominant (Reynolds) forces in one experiment. For the size of the CWP at-sea model test, the differences in model and prototype viscous forces on the platform become small and the environmental forcing functions can be considered gravity dominant. Platform dimensions will be scaled linearly based on the scale ratio.

Environmental Scaling - In an ideal environment for the scaling of a model test, the current velocity magnitudes would scale as $1/\sqrt{\lambda}$ relative to full scale at all points along the vertical current profile. The wave energy spectrum will have a form identical to the full-scale spectrum, with [periods] scaled as $1/\sqrt{\lambda}$ and corresponding wave heights [scaled] as $1/\lambda$.

Since the actual environment cannot be controlled, environmental scaling reduced to a problem of site selection, in which the site chosen should have a high probability of satisfying the scaling relations detailed above. For example, to simulate full-scale conditions on a 1/3 scale model, a site should be chosen at which the current velocity magnitude is $1/\sqrt{3}$ of full scale current at all points along the vertical profile, and the spectrum is defined by a significant wave height and period equal to 1/3 and $1/\sqrt{3}$ of the full scale height and period respectively.

PIPE SCALING METHODOLOGY

The procedure which follows should be applied to evaluate scale model pipe designs to be sure that they perform similarly to the prototype OTEC 10/40 MW Pilot Plant CWP.

For a constant diameter pipe such as envisioned in the at-sea test article, the flexural scaling problem reduced to making equivalent for model and prototype the nondimensional parameters a, b, c, d, and e listed under pipe scaling laws. Since the model test is scaled according to Froude laws (gravity dominant), frequencies scale as $\sqrt{\lambda}$ and the pipe natural frequencies should scale accordingly. The scaling problem becomes the simultaneous solution of the four equations:

$$\frac{m\omega_n^2 L^4}{EI} \text{ prototype} = \frac{m\omega_n^2 L^4}{EI} \text{ model} \quad 1$$

$$\frac{WL^2}{EI} \text{ prototype} = \frac{WL^2}{EI} \text{ model} \quad 2$$

$$\frac{TL^2}{EI} \text{ prototype} = \frac{TL^2}{EI} \text{ model} \quad 3$$

$$\frac{\rho_w AU^2 L^2}{EI} \text{ prototype} = \frac{\rho_w AU^2 L^2}{EI} \text{ model} \quad 4$$

For thin-walled pipes ($t/D < .05$) such as OTEC CWPs, inside and outside diameters are nearly equal and cross sectional areas and moments of inertia can be closely approximated as:

$$A \approx \pi Dt \quad 5$$

$$I \approx \frac{\pi D^3 t}{8} \quad 6$$

Equation 1 related the inertia forces to flexural forces. Total mass per length is the sum of the pipe mass, the mass of the pipe's internal water and the added mass. The cylinder added mass coefficient is approximately equal to one. Using this with Equations 5 and 6, Equation 1 becomes:

$$\left(\frac{4\omega_n^2 L^4 (\rho_w D + 2\bar{\rho}t)}{ED^2 t} \right)_{\text{prototype}} = \left(\frac{4\omega_n^2 L^4 (\rho_w D + 2\bar{\rho}t)}{ED^2 t} \right)_{\text{model}} \quad 7^*$$

Equation 2 relates weight to flexure forces. The pipe weight in water equals the weight in air minus the buoyant force. Using this and Equations 5 and 6, Equation 2 becomes:

$$\left(\frac{8gL^3 (\bar{\rho} - \rho_w)}{ED^2} \right)_{\text{prototype}} = \left(\frac{8gL^3 (\bar{\rho} - \rho_w)}{ED^2} \right)_{\text{model}} \quad 8$$

Inserting Equations 5 and 6 into Equations 3 and 4 yields:

$$\left(\frac{8\rho_w U^2 L^2}{ED^2} \right)_{\text{prototype}} = \left(\frac{8\rho_w U^2 L^2}{ED^2} \right)_{\text{model}} \quad 9^*$$

$$\left(\frac{8TL^2}{E\pi D^3 t} \right)_{\text{prototype}} = \left(\frac{8TL^2}{E\pi D^3 t} \right)_{\text{model}} \quad 10^*$$

Equations 9 and 10 are easily solved since the tension and internal flow in the model can be any value. The unknown in Equations 7 and 8 are the Young's modulus, thickness and average material density of the model. Since there are only two equations, one unknown must be fixed beforehand. Since model density can be adjusted with weights or buoyancy and a major interest is determining the model thickness, Young's modulus of the model was chosen as the independent variable.

* [Equations 7, 9, and 10 have been corrected; see Errata].

Pipe Extension

For pipe extension, the governing equation is:

$$\left(\frac{AE}{m_p \omega_n^2 L^2} \right)_{\text{prototype}} = \left(\frac{AE}{m_p \omega_n^2 L^2} \right)_{\text{model}} \quad 11^*$$

Since $m_p = \bar{\rho} A$ and frequency is scaled as $\sqrt{\lambda}$, [and pipe length scaled as $1/\lambda$] Equation 11 can be expressed as:

$$\left(\frac{E}{\bar{\rho}} \right)_{\text{prototype}} = \left(\lambda \frac{E}{\bar{\rho}} \right)_{\text{model}} \quad 12$$

Pipe Torsion

For torsion, the governing equation is:

$$\left(\frac{I_p G}{I_\alpha \omega_n^2 L^2} \right)_{\text{prototype}} = \left(\frac{I_p G}{I_\alpha \omega_n^2 L^2} \right)_{\text{model}} \quad 13$$

where: $I_\alpha = \bar{\rho} \cdot I_p$ and frequency scales as $\sqrt{\lambda}$ [and pipe length scaled as $1/\lambda$] reducing Equation 13 to:

$$\left(\frac{G}{\bar{\rho}} \right)_{\text{prototype}} = \lambda \left(\frac{G}{\bar{\rho}} \right)_{\text{model}} \quad 14$$

Upon examining equations 7, 8, 13 and 14 it becomes apparent that it will be impossible to simultaneously scale a model test to correctly scale flexural, torsional, and longitudinal responses. The reason for this difficulty is that for flexural response the total mass is dominated by the mass of the internal and external water, while for extension and torsion, the response is dominated by pipe material structural properties. For the extreme cases of no water or pipe density much greater than water density this problem does not exist. Since pipe flexural response appears to have the greatest effect on pipe life the CWP test article should be designed to ensure proper scaling of this response, at a minimum.

* [Equation 11 has been corrected; see Errata].

3. NUMERICAL EXAMPLES

When planning a model test, one must consider certain problems associated with scaling down the prototype pipe. For example, a strictly scaled model may not have sufficient thickness to withstand handling or deployment loads. Additionally, buckling and ovaling loads during operations have not been considered in the context of theory analysis. As an example, consider the problem of scaling a typical prototype OTEC CWP to a 1/3 or 1/6 scale. The full scale length is 3000 ft. with inside diameter being 30 ft. The following material possibilities have been considered in conceptual prototype CWP designs, References (4) and (5):

TABLE 1

<div> <div>PROPERTIES</div> <div>MATERIALS</div> </div>	<div>WEIGHT DENSITIES, (lb/ft³)</div>	<div>YOUNG'S MODULUS, (E) (psi)</div>	<div>THICKNESS FULL SCALE, (t) (in)</div>
Steel	490	30×10^6	.5
Fiberglass Reinforced Plastic (FRP)	112	3×10^6	2.75
Elastomer (steel reinforced)	503	19×10^6	1.25
Elastomer (nylon reinforced)	73	2×10^6	1.25
Polyethylene	61	$.1 \times 10^6$	5.50

Tables 2 and 3 show the comparison of pipe thickness between prototype and model for various materials. The model wall thickness and average material density for proper scaling of prototype CWP's were obtained by fixing the Young's modulus of the model and solving for density and thickness using equations 7 and 8 of section 2. For most cases, the required average model density is not equal to the actual material density. For example, scaling an FRP prototype with FRP requires a model density of 352 lb/ft³ to simultaneously scale EI and weight while the actual density of FRP is only 112 lb/ft³. Added weight would be necessary to increase the average density to the required value.

Upon examination of the required model pipe thicknesses it is apparent that some thickness increase will be necessary over the calculated scaled value from equations 7 and 8 in order that the model not buckle under anticipated in situ loadings. Dynamic buckling pressure scales approximately as the scale ratio. Cylindrical shell buckling strength of extremely long cylinders ($L/D > 50$) is directly related to the value $E(t/D)^3$. When the pipe thickness scaled according to this relationship as compared with the linear loading thickness (Tables 2 and 3) the relative deviation from a correctly scaled pipe is greater for the 1/6 scale (for most cases). In other words, the larger the scale ratio, the greater the thickness deviation from scaled values needed to prevent buckling.

As indicated in Tables 2 and 3, the required average model densities for the 1/6 scale designs are further from the prototype material density values than are the 1/3 scale model densities. For example, if a fiberglass model is used to scale a fiberglass prototype, the 1/6 scale model CWP requires an average density of 352 lb/ft³ but the 1/3 scale model requires a density of 208 lb/ft³. Thus, a proportionally smaller bottom weight will be necessary for the 1/3 scale models, making them easier to fabricate and deploy.

TABLE 2

COMPARISON OF PROTOTYPE CWP THICKNESS WITH MODEL
CWP THICKNESS FOR VARIOUS MATERIALS

PROTOTYPE INSIDE DIAMETER = 30 FT
SCALE = 1/6

PROTOTYPE	MODEL				
	STEEL E=30x10 ⁶ psi	FRP E= 3x10 ⁶ psi	ELASTOMER STEEL E=19x10 ⁶ psi	ELASTOMER NYLON E= 2x10 ⁶ psi	POLYETHYLENE E=.1x10 ⁶ psi
STEEL E=30x10 ⁶ psi Y=490 lb/ft ³ t=.5 in	Y=2626W lb/ft ³ t=.0138 in	Y=319W lb/ft ³ t=.139 in	Y=1683W lb/ft ³ t=.0219 in	Y=234W lb/ft ³ t=.209 in	Y=71W lb/ft ³ t=5.47 in
FRP E=3x10 ⁶ psi Y=112 lb/ft ³ t=2.75 in (Sandwich, FRP 1.75 in)	Y=2987W lb/ft ³ t=.0076 in	Y=352W lb/ft ³ t=.07 in	Y=1888W lb/ft ³ t=.012 in	Y=256W lb/ft ³ t=.113 in	Y=73W lb/ft ³ t=2.55 in
ELASTOMER STEEL E=19x10 ⁶ psi Y=503 lb/ft ³ t=1.25 in	Y=422 lb/ft ³ t=.0219 in	Y=408W lb/ft ³ t=.220 in	Y=2698W lb/ft ³ t=.034 in	Y=341W lb/ft ³ t=.330 in	Y=78W lb/ft ³ t=7.38 in
ELASTOMER NYLON E=2x10 ⁶ psi Y=73 lb/ft ³ t=1.75 in	Y=874W lb/ft ³ t=.0032 in	Y=145W lb/ft ³ t=.032 in	Y=577W lb/ft ³ t=.005 in	Y=118W lb/ft ³ t=.048 in	Y=67W lb/ft ³ t=.983 in
POLYETHYLENE E=.1x10 ⁶ psi Y=61 lb/ft ³	(1)	(1)	(1)	(1)	Y=46B lb/ft ³ t=.151 in
W = Weight needed B = Buoyancy needed (1) = Impossible, Physically unrealistic parameters					

TABLE 3 *

COMPARISON OF PROTOTYPE CWP THICKNESS WITH MODEL
CWP THICKNESS FOR VARIOUS MATERIALSPROTOTYPE INSIDE DIAMETER = 30 FT
SCALE = 1/3

PROTOTYPE	MODEL				
	STEEL E=30x10 ⁶ psi	FRP E= 3x10 ⁶ psi	ELASTOMER STEEL E=19x10 ⁶ psi	ELASTOMER NYLON E= 2x10 ⁶ psi	POLYETHYLENE E=.1x10 ⁶ psi
STEEL E=30x10 ⁶ psi γ=490 lb/ft ³ t=.5 in	γ=1344W lb/ft ³ t=.055 in	γ=191W lb/ft ³ t=.561 in	γ=873W lb/ft ³ t=.088 in	γ=149 lb/ft ³ t=.837 in	γ=68W lb/ft ³ t=19.3 in
FIBERGLASS E=3x10 ⁶ psi γ=112 lb/ft ³ t=2.75 in (Sandwich, FRP 1.75 in)	γ=1504W lb/ft ³ t=.03 in	γ=208W lb/ft ³ t=.304 in	γ=976W lb/ft ³ t=.048 in	γ=150W lb/ft ³ t=.456 in	γ=69W lb/ft ³ t=12.2 in
ELASTOMER STEEL E=19x10 ⁶ psi γ=503 lb/ft ³ t=1.25 in	γ=2143W lb/ft ³ t=.088 in	γ=272W lb/ft ³ t=.893 in	γ=1381W lb/ft ³ t=.138 in	γ=203W lb/ft ³ t=1.33 in	γ=71W lb/ft ³ t=33.7 in
ELASTOMER NYLON E=2x10 ⁶ psi γ=73 lb/ft ³ t=1.75 in	γ=469B lb/ft ³ t=.013 in	γ=104B lb/ft ³ t=.129 in	γ=321B lb/ft ³ t=.02 in	γ=91W lb/ft ³ t=.194 in	γ=55W lb/ft ³ t= 4 in
POLYETHYLENE E=.1x10 ⁶ psi γ=61 lb/ft ³	(1)	(1)	(1)	(1)	γ=55B lb/ft ³ t=.604 in
W = Weight needed B = Buoyancy needed (1) = Impossible, Physically unrealistic parameters					

* See Page 22.

Table 4 contains information on how a model pipe scaled for one prototype material will scale for other prototype materials. Pipe 1 is an FRP model of a steel CWP. Pipe 2 is a polyethelene model of an FRP pipe. For both cases, physical and geometric parameters defining full-scale cold water pipes of other materials are also presented. In this way, one can see the relationship between a single scale model and how it may represent a number of full-sized (30-ft diameter) CWPs of various materials.

TABLE 4
COMPARISON OF VARIOUS PROTOTYPE PIPES WITH 1/6 SCALE MODELS
OF STEEL AND FIBERGLASS PIPES

PIPE 1 (FRP model of steel CWP)	PIPE 2 (Polyethylene model of FRP CWP)
Fiberglass Reinforced Plastic $\gamma = 319.6 \text{ lb/ft}^3$ (added weight) D = 5 ft t = .139 in L = 500 ft Models a 1/2 in. steel prototype, L = 3000 ft as given in Table 2*. In addition, Pipe 1 is also a 1/6 scale model of the following pipes*:	Polyethylene $\gamma = 72.9 \text{ lb/ft}^3$ (added weight) D = 5 ft t = 2.55 in L = 500 ft Models a 2.75 in. fiberglass prototype L = 3000 ft as given in Table 2*. In addition, Pipe 2 is also a 1/6 scale model of the following pipes*:
Prototype-Fiberglass Reinforced Plastic D = 30 ft t = 5.2 in $\gamma = 106 \text{ lb/ft}^3$ Prototype-Elastomer (Steel Reinforced) D = 30 ft t = .75 in $\gamma = 333 \text{ lb/ft}^3$ Prototype-Elastomer (Nylon Reinforced) D = 30 ft t = 7.98 in $\gamma = 91 \text{ lb/ft}^3$ Prototype-Polyethylene Impossible (thickness unrealistic)	Prototype-Steel D = 30 ft t = .27 in $\gamma = 547 \text{ lb/ft}^3$ Prototype-Elastomer (Steel Reinforced) D = 30 ft t = .425 in $\gamma = 372 \text{ lb/ft}^3$ Prototype-Elastomer (Nylon Reinforced) D = 30 ft t = 4.17 in $\gamma = 95.7 \text{ lb/ft}^3$ Prototype-Polyethylene Impossible (thickness unrealistic)

* Clarification comments added by editor.

4. BENCHMARK* ANALYSIS OF SCALED CWP RESPONSES

In order to assess the validity of the scaling procedure within the NOAA/DOE Analytic Code (Reference 1) a typical CWP/Platform configuration was simulated for full scale and 1/10 scale. The pipe parameters for the model were generated via the methodology in Section 2 and the properties for the two pipes are presented in Table 5. The 1/10 scale model is only a hypothetical example which has not been modified to withstand buckling or ovaling stresses, since the analysis within the NOAA/DOE code is restricted to linear elastic beam theory. The assumptions used for the simulation were:

- o Monolithic CWP with hinged joint at top and free at the bottom
- o Platform used is a 1/10th scale model of the APL barge
- o Environmental conditions for full scale are significant wave height of 20 ft and modal frequency of 0.4 rad/sec
- o Environmental conditions for 1/10th scale are significant wave height of 2 ft and modal frequency of 1.27 rad/sec.

The significant responses for these two cases for head and beam seas are presented in Tables 6a and 6b. In Table 7a and 7b, the shear forces and bending moments for the pipe are tabulated. These two tables show that the responses and loadings predicted by the NOAA/DOE code properly follow Froude

* A validated example that can be used as a reference point.

TABLE 5

GEOMETRIC AND PHYSICAL CHARACTERISTICS OF PROTOTYPE AND
1/10 SCALE CWP_s USED TO ASSESS SCALING RELIABILITY
OF NOAA/DOE ANALYTIC CODE

	Prototype*	*1/10 Model
Length (ft)	3000	300
Outside Diameter (ft)	30	3
Wall Thickness (in)	.5	.015
Weight Density (lb/ft ³)	320	923
Young's Modulus (psi)	3x10 ⁶	1x10 ⁶

*[N.B. Although given Young's Modulus corresponds to FRP, the weight density does not (see Table 1). That is, a non-specified pipe material is being used.]

TABLE 6

PREDICTED SIGNIFICANT CWP/PLATFORM RESPONSES FOR PROTOTYPE
AND 1/10 SCALE PIPES (PIPE PARAMETERS DEFINED IN TABLE 5)

Table 6a

Beam Seas

	Prototype	Model
Significant Wave Height (ft)	20	2
Peak Frequency (rad/sec)	.4	1.2649
Sway (ft)	6.8698	.6870
Heave (ft)	12.5239	1.2511
Roll (deg)	2.4939	2.4937
Pipe Heave (ft)	12.5273	1.2514
Pipe Roll (deg)	1.3991	1.3992
Relative Roll (deg)	2.558	2.556

Table 6b

Head Seas

	Prototype	Model
Significant Wave Height (ft)	20	2
Peak Frequency (rad/sec)	.4	1.2649
Surge (ft)	5.321	.5320
Heave (ft)	8.6196	.8604
Pitch (deg)	3.9778	3.9779
Pipe Pitch (deg)	1.7946	1.7946
Relative Pitch (deg)	5.4143	5.4144

TABLE 7
PREDICTED SIGNIFICANT SHEAR FORCE AND BENDING MOMENT FOR PROTOTYPE
AND 1/10 SCALE PIPES (PIPE PARAMETERS DEFINED IN TABLE 5,
ENVIRONMENTAL INPUTS DEFINED IN TABLES 6a AND 6b)

Table 7a

Beam Seas

Station*	Prototype		Model	
	Shear	Bending	Shear	Bending
1	.2193E6	0	.2192E3	0
2	.7387E5	.4586E7	.7388E2	.4587E3
3	.4938E5	.2024E7	.4938E2	.2024E3
4	.3924E5	.1539E7	.3924E2	.1539E3
5	.3464E5	.1301E7	.3464E2	.1302E3
6	0	0	0	0

Table 7b

Head Seas

Station*	Prototype		Model	
	Shear	Bending	Shear	Bending
1	.2555E6	0	.2555E3	0
2	.7442E5	.7216E7	.7443E2	.7215E3
3	.5547E5	.3422E7	.5546E2	.3421E3
4	.4987E5	.2492E7	.4587E2	.2491E3
5	.4137E5	.2112E7	.4138E2	.2112E3
6	0	0	0	0

* Stations on the prototype are every 600 ft, on the model every 60 ft, station 1 is at pipe/platform interface, station 6 is at bottom of pipe.

scaling relations. Under Froude scaling (gravity dominant), dimensions scale linearly as the scale ratio and forces scale as the scale ratio cubed. Since shear is a force, and bending moment is a force times a length, shear scales as the scale ratio cubed, and bending as the scale ratio to the fourth power. Linear motions (heave, surge, sway) scale as the scale ratio while angles remain the same since they are dimensionless. * The analysis performed by the benchmark comparison demonstrates the validity of the NOAA/DOE CWP Analytic Code in designing scaled CWP's.

In the present analysis, the separate effects of currents and waves are scaled but the effects of current/wave interactions are not included. Presently, the NOAA/DOE Analytic Code treats currents as a simple static case and hence cannot simulate any current wave coupling. Static deflections for the full-scale and 1/10th scale current environment showed good agreement with each other. The responses to current wave combinations may not scale as closely as the separate components scale due to viscous and other interaction effects.

*[Should read validity of the scaling procedure and not the actual Structural Analysis Code.]

5. REFERENCES

1. Pompa, J.A. and D.F. Lunz (eds.), "NOAA/DOE CWP Structural Analysis Package," ORI Tech. Rept. 1605, September 1979.
2. Sutherland, W.H. (ed.), "At-Sea Test System Point Design for a 1/3 Scale Cold Water Pipe," ORI Tech. Rept. 1601, December 1979.
3. Barr, R.A. and V.E. Johnson, "Evaluation of Analytical and Experimental Methods for Determining OTEC Plant Dynamics and CWP Loads," 6th OTEC Conference Proceedings, Washington, D.C., June 1979.
4. TRW Defense & Space Systems Group, "Ocean Thermal Energy Conversion Cold Water Pipe Preliminary Design Project," Report prepared for NOAA Office of Ocean Engineering under Dept. of Commerce Contract No. MO-A01-78-00-4141, November 1979.
5. Science Applications, Inc. "OTEC Modular Experiment Cold Water Pipe System Design Study," Report prepared for NOAA Office of Ocean Engineering under Dept. of Commerce Contract No. MO-A01-78-00-4142, December 1979.

ERRATA

PAGE	LOCATION	DESCRIPTION
ii	Line 7 of Abstract	Add [in addition to other analytical design tools] after "...scaling laws,".
iv	12 lines from bottom of page of original report (5 lines from bottom of page of revised report)	The symbol for the "pipe weight in water (lb)" should be W instead of W_w .
iv	13 lines from bottom of page of original report (6 lines from bottom of page of revised report)	The symbol for the "pipe weight in water per unit length" should be W/L instead of W. (see 4th line from bottom of Page 3).
2	Line 7 from bottom of last paragraph	Add [actually developed in Reference 3 and reproduced here] after "... are developed".
3	Line 5, first paragraph	Add [i.e., Reynolds scaling vs. Froude scaling] after "... inertial responses".
4	Line 2	change $\frac{W_w L^2}{EI}$ to $\frac{WL^2}{EI}$
4	Line 9	Add [N.B. Only equations a, b, c, d, and e are utilized here] after Line 9.
4	Line 6, second paragraph	Add [i.e., same drag, lift and added mass coefficients] at the end of second paragraph.
4	Equation at bottom of page	Should be m_p (the mass of pipe material per unit length) in denominator instead of m , i.e., $W_n^2 = \frac{K_1 AE}{m_p L^2}$ <p>The total mass m is defined as $m = m_p + m_a + m_i$ where</p>

ERRATA

PAGE	LOCATION	DESCRIPTION
4 (cont)		$m_p = \pi D t \bar{\rho}$ $m_a = C_a \rho_w \frac{\pi D^2}{4}$ $m_i = \rho_w \frac{\pi D^2}{4}$ <p>If C_a, the added mass coefficient, is assumed equal to 1, then</p> $m = \frac{\pi D}{2} (\rho_w D + 2 \bar{\rho} t)$
5	Last line in second paragraph from bottom of page	Should read "period scaled as $1/\sqrt{\lambda}$ " instead of frequency. Or "frequency scaled as $\sqrt{\lambda}$ " instead of $1/\sqrt{\lambda}$.
6	Equation 2	Note that W is the pipe weight in water as correctly defined above.
6	Equation 6	<p>The polar moment of inertia (utilized for <u>torsion</u> analysis) is</p> $I_p = \frac{\pi}{32} \{(D + 2t)^4 - D^4\}$ <p>and for $D \gg t$</p> $I_p \doteq \frac{\pi}{4} D^3 t$ <p>while here for pipe <u>bending</u> analysis, the moment of inertia is 1/2 the value of I_p as correctly given by author.</p>
7	Equation 7	<p>If $C_a = 1$ as described above, then</p> $m = m_p + 2 m_i = \frac{\pi D}{2} (\rho_w D + 2 \bar{\rho} t)$ <p>and</p> $I \doteq \frac{1}{8} \pi D^3 t$ <p>therefore, Equation 1 on Page 6 can be rewritten as:</p> $\left\{ \frac{4 \omega_n^2 L^4 (\rho_w D + 2 \bar{\rho} t)}{E D^2 t} \right\} = \text{Const.}$

ERRATA

PAGE	LOCATION	DESCRIPTION
7	Equation 9	Numerator: 8 instead of 2 Denominator: D^2 instead of t
7	Equation 10	Denominator: D^3 instead of D
7	Equation 11	Denominator: m_p instead of m $m_p = \bar{\rho} A \dot{=} \bar{\rho} \pi D t$
7	Line after Equation 11 (Page 8 of the revised report)	Add [and pipe length scaled as $1/\lambda$] between ... $\sqrt{\lambda}$, and Equation 11.
8	Line after Equation 13	Add [and pipe length scaled as $1/\lambda$] between ... $\sqrt{\lambda}$ and reducing Equation 13.

For future reference, equations derived to scale flexural responses are included. These equations were utilized by the Editor to check Table 3.

From corrected Equations 7 and 8:

$$\gamma_m = \frac{E_m}{E_p} \left\{ \lambda \bar{\gamma}_p - \left(\lambda - \frac{E_p}{E_m} \right) \gamma_w \right\}$$

$$t_m \left\{ \lambda \left(\frac{E_m}{E_p} \right) \left\{ 2 \bar{\gamma}_p + \gamma_w \left(\frac{D_p}{t_p} \right) \right\} - 2 \bar{\gamma}_m \right\} = \gamma_w D_m$$

D = pipe i.d.

E = Modulus of Elasticity (treated as parameter here)

t = thickness

γ = weight density (= ρg)

$\bar{\gamma}_p$ = average weight density of pipe material (prototype)

$\bar{\gamma}_m$ = average weight density of pipe material (model)

$\bar{\gamma}_w$ = weight density of average sea water (64 lb/ft³)

λ = linear prototype scale/linear model scale

The subscripts m and p refer to model and prototype, respectively.