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AN OTEC COLD WATER PIPE HYDROELASTIC RESPONSE EXPERIMENT

By
S. Taylor
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July 1979

Work Performed Under Contract No. EX-76-C-01-2331

Science Applications, Inc.
El Segundo, California

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U.S. Department of Energy

Solar Energy

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SAI-063-80R-047-LA

PREPARED FOR:

NATIONAL OCEANIC AND ATMOSPHERIC ADMINISTRATION

UNDER CONTRACT:

EX-76-C01-2331

ALBUQUERQUE • ANN ARBOR • ARLINGTON • BOSTON • CHICAGO • DENVER
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This report documents some laboratory experiments conducted at the California Institute of Technology in support of the Ocean Thermal Energy Conversion Program. The present work represents the final phase (after redirection) of an ongoing SAI study of vortex shedding effects under ERDA Contract EX-76-C-01-2331. Dr. Eugene Silva of NAVFAC was the original technical Monitor, and Dr. Terence McGuinness of NOAA is the present technical monitor.

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1.0 INTRODUCTION

OTEC cold water pipes in an ocean current cross-flow will experience time dependent hydrodynamic loads induced by alternate shedding of vortices. For flexible pipes the problem becomes increasingly complex since the hydrodynamics couples with the dynamics of the pipe producing a phenomenon known as vortex strumming. Approximate analytical models have been presented for special cases (1,2), however no generally accepted mathematical theory of the flow/structure interaction has been proposed. Although a complete fluid/structural model is very difficult and has not been solved for current conditions of interest, an oscillator model which does show promise for many practical applications is one developed by Iwan (3) and further advanced by Griffin.

The general approach taken here is to first identify the important parameters influencing the motion of typical OTEC cold water pipes in still water. The relevant parameters are identified by first formulating and nondimensionalizing the structural equation of motion that is applicable to all OTEC CWP designs. An order-of-magnitude analysis then yields the parameters which most strongly influence the natural response of any given cold water pipe design. The nondimensional equation also specifies how one must scale the parameters in order to conduct a useful experiment. The natural response of the pipe serves as basis for comparison for the overall fluid-structural response.

Although most of the proposed CWP designs are geometrically similar and all convey an incompressible fluid, the many combinations of materials, wall thickness-to-diameter ratios, and end conditions provide a wide variety of possible dynamic responses. For example, internal pumping is important in some designs but not in others, as is shown

in Section 2.0. However, through the order-of-magnitude analysis it is realized that most CWP designs fall into one of two distinct classes of problems: one, the pipe offers essentially no resistance to bending (the Bell Aerospace design is a good example), and two the bending restoring force is important. Since attention has been focused to date on the latter class of cold water pipes, the present experimental investigation was directed toward such designs although a compliant rubber pipe was also tested.

1.1 CWP Dynamic Response - Relevant Parameters

The typical OTEC cold water pipe consists of a tubular beam of length L , mass per unit length m , and flexural rigidity EI , conveying an incompressible fluid of mass per unit length M at a speed V , as shown in Figure 1. Earlier investigators [4] interested in the stability of such pipes derived the lateral equation of motion for pipes undergoing small deflections. The equation is applicable to OTEC CWP designs if the effects of buoyancy and apparent mass are included. The modified equation may be written

$$\begin{aligned}
 E^* I \frac{\partial^5 y}{\partial^4 x \partial t} + EI \frac{\partial^4 y}{\partial x^4} + \left[MV^2 - \bar{T} - g(M-M'+m)(L-x) \right] \frac{\partial^2 y}{\partial x^2} \\
 + 2MV \frac{\partial^2 y}{\partial x \partial t} + g(M-M'+m) \frac{\partial y}{\partial x} + C \frac{\partial y}{\partial t} + (M+M'+m) \frac{\partial^2 y}{\partial t^2} = 0.
 \end{aligned} \tag{1}$$

where E^* is a structural damping coefficient, assumed to be viscoelastic and of the Kelvin-Voigt type, c is a coefficient

FIGURE 1. VERTICAL FLEXIBLE PIPE WITH INTERNAL FLOW

of viscous fluid damping, M' is the mass of the displaced fluid, and $\bar{T}$ is a weight attached to the pipe at $x=L$. The effects of internal flow on the motion of the pipe are apparent in the terms involving V^2 and V which represent the centrifugal and Coriolis forces, respectively.

The important dynamic parameters of typical OTEC CWP designs are identified by nondimensionalizing equation (1) and estimating the order-of-magnitude of the terms. Defining the following dimensionless quantities

$$\begin{aligned}\xi &= \frac{X}{L}, \quad \eta = \frac{Y}{L}, \quad \tau = \left(\frac{EI}{M+M'+m} \right)^{\frac{1}{2}} \frac{t}{L^2}, \\ \alpha &= \left[\frac{I}{E(M+M'+m)} \right]^{\frac{1}{2}} \frac{E^*}{L^2}, \quad \beta = \left(\frac{M}{M+M'+m} \right), \quad \gamma = \frac{(M-M'+m)gL^3}{EI}, \quad (2) \\ \gamma &= \frac{(M-M'+m)}{EI} g L^3, \quad u = \left(\frac{M}{EI} \right)^{\frac{1}{2}} LV, \\ v &= \frac{cL^2}{[EI(M+M'+m)]^{\frac{1}{2}}}, \quad \Gamma = \frac{\bar{T}L^2}{EI}\end{aligned}$$

equation (1) may be rewritten

$$\begin{aligned}\alpha \frac{\partial^5 \eta}{\partial \xi^4 \partial \tau} + \frac{\partial^4 \eta}{\partial \xi^4} + \left[u^2 - \Gamma - \gamma(1-\xi) \right] \frac{\partial^2 \eta}{\partial \xi^2} \\ + 2\beta u \frac{\partial^2 \eta}{\partial \xi \partial \tau} + \gamma \frac{\partial \eta}{\partial \xi} + v \frac{\partial \eta}{\partial \tau} + \frac{\partial^2 \eta}{\partial \tau^2} = 0.\end{aligned}\quad (3)$$

Taking ξ and τ of order 1, and using the coefficient of the bending restoring force (of order 1) as a reference, the relative magnitude of each of the coefficients may be assessed for typical OTEC CWP designs.

Terms involving γ represent the action of both gravity and buoyant forces which may be grouped as one and termed the

body force. For $t/D \ll 1$, the body force may be written

$$\gamma = \frac{8g(\rho_p - \rho_f)L}{E} \left(\frac{L}{D} \right)^2$$

Using full scale OTEC dimensions, it may be shown that the body force is at least as important as the bending restoring force unless the pipe is very nearly neutrally buoyant ($\rho_p = \rho_f$). For materials whose density is less than the sea water, the body force tends to enhance the motion of the pipe. That is, the upward buoyant force is greater than the downward gravity force.

The relevance of internal flow is apparent through the dimensionless velocity, u . By first rearranging the expression for u ,

$$u = \frac{L}{D} \sqrt{\frac{2\rho_f}{E} \frac{D}{t}} V,$$

it becomes clear that terms in equation (3) involving this quantity are of higher order for the more rigid materials. For compliant materials the effects of internal flow may become important if the pipe is very slender and thin-walled. It has been shown [5], for materials such as rubber, that instabilities are possible within the expected range of internal flow velocities (it is assumed that $V = 6$ ft/sec).

The term in equation (3) invoking the nondimensional tip weight, Γ , was included for CWP designs that require a light, compliant material such as rubber. For such materials the bending restoring force is, of course, negligible and the weight, T , is needed to oppose the upward buoyant force. However, for the sake of simplicity, the present discussion will limit itself to CWP designs that do not require tip weights.

Neglecting the effects of damping for the moment and reviewing the overall cold water pipe problem, it is realized that most proposed designs fall into one of two categories. In the first category the bending restoring forces in the pipe are significant and must be considered when investigating the

response of the pipe. For pipes in this category, however, the effects of internal pumping are small when compared to the bending forces and may be neglected. The second category contains those pipes which offer essentially no resistance to bending. Internal flow, however, may influence the motion of such pipes. Body forces are important, in general, for pipes in both categories.

It would be possible to design a pipe whose response depended upon both the bending restoring forces and the effects of internal flow. For a better understanding of the complex fluid - structural response, however, it is suggested that attention be initially directed to only those pipes that fall into either of the two mentioned categories. Although the very flexible CWP designs show possible promise more emphasis has been placed on the more rigid pipes. The present experimental investigation, therefore, focuses on the latter designs.

Having better defined the pipes which are presently of interest, it is considerably easier to discuss the effects of damping. The terms in equations (1) and (3) which model both the fluid and structural damping are only representative terms. The first term in equation (1), for example, models the structural damping as viscoelastic and of the Kelvin-Voigt type. However, the model for structural damping depends, of course, on the particular material. The term which models the fluid damping is also only a representative term.

Instead of attempting to accurately model the damping effects within the context of equation (1), it is more convenient to describe the influence of damping in terms of the logarithmic decrement. The total logarithmic decrement of damping, δ , is defined as the logarithm of the ratio of successive free vibration amplitudes and may be determined experimentally. The total decrement is conventionally expressed as the sum of the contributions from structural damping, δ_s , and fluid damping, δ_f .

The effect of the logarithmic decrement on resonant amplitudes is most relevant when considering the fluid-structural response and will be discussed in the following section.

Summarizing, the cold water pipe designs may be placed, for simplicity, into one of two categories: (1) those that offer structural bending restoring forces and whose motion is not influenced by internal flow effects and (2) those pipes that offer relatively no structural resistance to bending, but whose motion may be affected by internal flow. In general, body forces should be considered when attempting to predict the true motion of full-scale cold water pipes. The exceptions include CWP's that are almost neutrally buoyant.

1.2 Hydroelastic Response Model

The term hydroelastic response will be used here to describe the interaction of the CWP with the vortex shedding. Experimental investigations have demonstrated that the hydroelastic response of cylinders consists of oscillations in both the lift and drag directions. Oscillations in the drag direction occur when the shedding frequency, f_s , is approximately one half the natural frequency of the cylinder. Response in the lift direction is observed when the shedding frequency is approximately equal to the cylinder natural frequency. However, substantial responses have been observed when the difference between the shedding and natural frequencies is about 20-30%. Such "lock-on" bandwidths are apparent for oscillations in both the lift and drag directions. Although in-line (drag direction) oscillations have been observed, the amplitudes in the lateral direction are more significant.

Few mathematical models have been proposed which describe the hydroelastic response of cylinders. The model originally developed by Iwan and Blevins [6] and later extended by Iwan [3] seems to be the most useful for predicting the resonant amplitude response of elastically flexible cylinders. The model

is unique in that it attempts to characterize the gross physical behavior of the fluid as well as the structure. From an engineering viewpoint, the maximum amplitude of vibration is generally of most interest. Iwan's model suggests that the resonant amplitude may be predicted by the relation

$$\frac{\gamma_n}{D} = \frac{a_4 \sqrt{4/3} \gamma_n}{2\pi^3 S^2 \mu_r \zeta_n^T} \left[\frac{(a_1 - a_4)}{a_2} + \frac{a_4^2}{\pi^2 a_2 S \mu_r \zeta_n^T} \right]^{1/2} \quad (4)$$

where a_1 , a_2 , a_4 are constants determined from experimental data on fixed and spring-mounted rigid cylinders. The product $\mu_r \zeta_n^T$ is related to the conventional reduced damping parameter δ_r by

$$\mu_r \zeta_n = \frac{1}{\pi^2} \delta_r = \frac{2m\delta}{\pi^2 \rho D^2}$$

where m is the mass per unit length of the structure plus added mass and δ is the logarithmic decrement of damping which may be determined experimentally. The variable γ_n is a dimensionless mode shape parameter of the n th structural mode and may be calculated for a variety of structural elements whose mode shape is analytically expressible.

Shown in Figure 2 is a comparison made by Iwan of his model with available experimental data. Also shown are the calculated values of γ_n for various structural elements. Note that the comparisons do not include the simply supported and cantilevered cylinders. Although the method collapses available experimental data reasonably well, further experimentation is necessary to validate the model. Such experiments should include the simply supported and/or cantilevered cylinders. Should the model prove to be valid for general elastically responding elements, the approach would be most useful in predicting the resonant amplitudes of many OTEC CWP designs.

FIGURE 2. CORRELATION OF PEAK RESPONSE AT STRUMMING
(IWAN, 1975)

2.0 EXPERIMENT DEFINITION

Of the two classes of cold water pipes that were outlined in Section 1.1, those that offer structural resistance to bending will be considered here. Recall that, in addition to the structural bending forces, the body forces are generally important. The dimensionless body force was shown to be given by

$$\gamma = \frac{8g (\rho_p - \rho_f) L}{E} \left(\frac{L}{D} \right)^2 \quad (5)$$

In order to properly scale the experiment, the magnitude of γ must be the same for both experimental and full scale pipes. The dimensionless body force depends, of course, on the dimensions and material properties of the pipe. Considering the various CWP designs, the dimensionless body force may assume a range of values. However, for γ to be of importance - relative to the structural bending forces - it must be at least of order 1 as was shown in Section 1.1.

The magnitude of γ achievable in the experiment is constrained first by the dimensions of the test facility. For reasons primarily of availability and low cost, the CalTech Free Surface Water Tunnel (FSWT) was chosen as the test facility. The maximum length of pipe that may be tested in the tunnel is 18 inches. Optimistically allowing an L/D of 100 ($D = 3/16$ inch) and setting $\gamma = 1$ in relation (5), it is found that the ratio $(\rho_p - \rho_f)/E$ must be of the order $10^{-7} \text{ sec}^2/\text{ft}^2$ or greater for body forces to be important. The ratio of the density to the elastic modulus for most solids is known to be approximately constant from material to material. For example, the ratio for polyethelene is $\sim 6 \times 10^{-9} \text{ sec}^2/\text{ft}^2$; the same ratio for steel is $\sim 4 \times 10^{-9} \text{ sec}^2/\text{ft}^2$. Assuming, optimistically, that a material may be found such that

$(\rho_p - \rho_f)/E = 10^{-8} \text{ sec}^2/\text{ft}^2$, it is obvious that the effects of body forces on the motion of scaled CWP's may not be simulated in the CalTech FSWT. Therefore, the experiment will effectively simulate the motion of a neutrally buoyant full scale pipe. From equation (1), the equation of motion for the present experimental pipe may be written

$$EI \frac{\partial^4 y}{\partial x^4} + (M + M' + m) \frac{\partial^2 y}{\partial t^2} = 0$$

where the damping terms have been omitted for convenience. The equation is recognized as the classical Bernoulli-Euler equation.

A variety of end conditions have been suggested for the full scale cold water pipe. Although the overall response of the pipe depends upon its fixity, it is expected that the resonant motion of all configurations will have similar effects, at least locally, on the vortex shedding. In order to investigate such effects one must first adequately describe the natural response of the pipe. One proposed design that is analytically tractable calls for clamped-free end conditions. An experimental study of the vortex induced resonance of the clamped-free pipe would permit a better understanding of the coupled fluid-structural response. In particular, the results would help to clarify the usefulness of Iwan's hydroelastic model in predicting the resonant response of full scale OTEC designs.

A problem which has received little attention in the past is the possibility that higher mode vortex induced resonance of full scale pipes may occur. OTEC CWP designs typically have third to fifth modes in the vortex shedding frequency ranges. The undamped mode shapes and natural frequencies of the clamped-free cantilevered beam (pipe) are well known thus permitting a useful experimental investigation of higher mode response.

The freestream velocity and pipe diameter are assumed to be uniform along the length of the pipe in Iwan's model, and the spanwise vortex shedding is assumed to be fully correlated. Since OTEC cold water pipes will be subjected to shear flows, the model may not be applicable. The experimental work of Stansby (7) indicates that a cellular vortex structure is formed when a cylinder oscillates in a shear flow. Each cell is locked-on to the natural frequency of the cylinder. Stansby proposed that the extent of the lock-on region may be estimated by:

$$\frac{\Delta x}{D} = 4\eta/\beta$$

where $\eta = A/D$, the dimensionless amplitude structural displacement; $\beta = D/U \, dU/dx$, the dimensionless freestream velocity gradient. However, the formula has not been proved to be valid for general mode shapes. Furthermore, models which predict the fluid forces acting on resonating cylindrical structures in shear flows and the experimental measurement of such forces are lacking. Although significant effort was expended in an attempt to investigate shear flow effects, a suitable test arrangement could not be developed within the scope of this effort.

The present experimental study seeks to quantify the vortex induced resonant response of a geometrically scaled OTEC pipe with clamped-free end conditions. Special attention is paid to higher order modal interaction. With respect to the dynamic response, the experimental pipe is effectively neutrally buoyant. That is, the influence of the body forces on the motion of the pipe is small compared to that of the structural bending forces.

3.0 EXPERIMENT DESIGN

The design of the present experiment began with the selection of geometrically scaled pipes. The selection was subject to three restrictions: (1) resonance must occur within the range of flow velocities achievable in the FSWT (1-20 ft/sec), (2) structural failure of the pipe must be avoided - proper geometric scaling requires wall thicknesses on the order of 5 mils, (3) practical instrumentation must be used to measure the quantities of interest; the measurement of minute forces and displacements arising from the smaller diameter tubes may require state-of-the-art instruments. Several iterations were necessary to select the tube which satisfied all criteria. Once the tube and transducer were selected, the remaining instrumentation and hardware were designed around the characteristics of the tube and the output of the transducers.

3.1 Selection of Experimental Pipe

The first step in the design iteration was to establish the conditions required for vortex induced resonance. Inducing first mode resonance is a relatively simple task. But exciting both the first and second modes of the same pipe within the various restrictions complicates the design.

3.1.1 Resonance Criteria

Recall from Section 2.0 that the equation describing the dynamic response of the experimental pipe is the Bernoulli-Euler equation. The natural frequencies the pipe with clamped-free end conditions are well known to be

$$f_n \text{ (CPS)} = \frac{k_n}{2\pi} \sqrt{\frac{EI}{M_T l^4}}, \quad n = 1, 2, \dots \quad (7)$$

where $k_n = 3.52, 22.0, \dots$

$$M_T = M + M' + m$$

The similarity parameter for the bending frequencies may be written

$$S_n = \frac{f_n D}{U}, \quad U = \text{freestream velocity.}$$

Vortex induced resonance of the pipe exists when the bending Strouhal number, S_n , is near the vortex shedding Strouhal number, S . Assuming that $S=0.2$ and that $S_n/S=1$ during resonance,

$$1 = \frac{k_n}{2\pi S} \frac{D}{U} \sqrt{\frac{EI}{M_T l^4}}. \quad (8)$$

For design purposes, one may assume

$$\begin{aligned} M+M'+m &\doteq M+M' \\ &\doteq \rho_f \pi D^2/2, \quad \text{when } t/D \ll 1. \end{aligned}$$

Substituting the approximation into (8), squaring both sides, and rearranging terms, the condition for resonance becomes

$$\frac{\rho_f U^2}{E} = \left(\frac{k_n}{4\pi S} \right)^2 \left(\frac{t}{D} \right) \left(\frac{1}{L/D} \right)^4. \quad (9)$$

Shown in Figure 3 are the first and second mode resonance curves calculated from equation (7). Using the inner scales, $\rho_f U^2/E$ is plotted versus L/D with t/D as a parameter. The outer ordinates give U for selected values of the elastic modulus E (represented are: steel, $E=30 \times 10^6$ psi, and polyethelene, $E=2 \times 10^6$ psi). The required diameter for an 18-inch length of pipe is scaled as the lower abscissa.

The range of aspect ratio assumed for typical OTEC CWP designs was: $60 < L/D < 100$. The thickness-to-diameter ratio, t/D , was assumed to be of the order $1/100$. Based upon such design requirements and considering the resonance criteria in Figure 3, stainless steel was chosen as the pipe material. Compared to most materials, steel's high modulus of elasticity

FIGURE 3. RESONANCE CONDITIONS FOR CANTILEVERED PIPE IN UNIFORM CROSSFLOW

requires larger values of dynamic pressure, thus permitting higher Reynolds numbers, in order to induce resonance. Stainless steel is also one of the few materials which is commercially available in the dimensions required for proper geometric scaling. For example, an L/D of 70 corresponds to a $\frac{1}{4}$ inch diameter and, for $t/D \sim 1/100$, a 2-3 mil wall thickness. Stainless is available in wall thicknesses as small as 5 mil and may be reduced even further by a chemical milling process.

The parameters of the pipe which were selected last included the diameter and the wall thickness. Possible material failure in the second mode placed the greatest restriction on the selection of the pipe.

3.1.2 Structural Failure Criteria

Two modes of structural failure for the stainless steel tube were considered: (1) yielding of the material due to bending, and (2) crippling, or static instability of the pipe.

For a cantilevered beam subjected to a transverse uniform loading, W_0 , the maximum bending stress may be represented by

$$(\sigma_b)_{\max} = \frac{W_0 L^2 D}{4I}$$

The loading for the pipe in a uniform flow is, of course, $C_D \frac{1}{2} \rho U^2 D$. For $t/D \ll 1$, $I \doteq \pi D^3 t/8$. Yielding of the material may therefore be avoided if

$$\frac{\rho U^2}{E} < \frac{\pi}{C_D} \frac{t/D}{(L/D)^2} \left(f \frac{\sigma_Y}{E} \right) \quad (10)$$

where f is a safety factor, σ_Y is the yield stress of the material and $C_D \doteq 1$.

Using Equation (10) and referring to Figure 3, it is observed that no problems associated with material yielding exist in the first mode ($k_n = 3.52$) for likely values of σ_y/E , L/D , and t/D . Second mode resonance, however, presents significant difficulties. The largest value of σ_y for a commercially available stainless steel tube that is roughly OTEC-scaled ($D \doteq .21$ inches, $t \doteq .005$ inches) is 100K - 110K psi. Assuming static bending only of the stainless steel tube and using $L = 18$ inches, the dynamic pressure required to induce second mode vibration yields a maximum bending stress, $(\sigma_b)_{\max}$, of 99K psi. Obviously, an alternative approach must be considered to achieve second mode resonance without exceeding the yield stress.

Reducing the natural frequency of the pipe, while retaining material strength, will certainly lessen the possibility of exceeding the material's yield stress during resonance. Referring to Equation (7), the frequency may be lowered by increasing the effective mass* of the pipe, M_T . Recall that the approximation made in deriving the design resonance condition, Equation (9), was: $M_T \doteq M' + M \doteq \rho_f \pi D^2 / 2$. Including the mass of the pipe in the calculation of M_T will not sufficiently reduce the natural frequency. Increasing the mass of the fluid inside the pipe, however, may significantly reduce the frequency. Since the effects of internal pumping were shown to be of higher order, excepting the very slender, more compliant designs, the experimental pipe may be filled with a liquid whose density is greater than that of water. For example, the frequency of a clamped-free pipe filled with

*It is understood that any reference to mass should be interpreted as mass per unit length.

mercury is about 40% of the frequency of the same pipe filled with water. Using the dimensions of the stainless steel OTEC-scaled tube mentioned previously, it may be shown that the maximum bending stress in second mode resonance is only 15K psi when the tube is mercury filled. However, use of mercury was ruled out by facility operating requirements.

The second structural failure mode considered was crippling, or static instability. It may be shown that crippling of a clamped-free pipe in a uniform flow may be avoided if:

$$\frac{\rho U^2}{E} < \frac{\pi}{C_D \sqrt{3(1 - \nu^2)}} \left(\frac{t/D}{L/D} \right)^2$$

This mode of failure, however, presented no difficulties.

3.1.3 Characteristics of Selected Pipes

The pipe chosen for the major portion of the experiment was stainless steel with the dimensions: L = 18 in., D = .21 in., t = .005 in. The pipe was instrumented with load cells and accelerometers, and most of the test time was devoted to gathering motion data.

To gain experience with a much more flexible pipe, an uninstrumented rubber tube was also tested. The pipe consisted of 16 inches of 0.01 inch wall surgical tube with a half pound weight attached to the bottom, to hold the pipe vertical. Qualitative pipe behavior was observed from photography.

3.2 Instrumentation

3.2.1 Natural Frequencies

Prior to the in-tunnel testing, the natural dynamic properties of the pipe must first be characterized. The first and second mode frequencies were determined by means of a mechanical oscillator or "shaker" as shown in Figure 4. The output signal from the shaker was fed to a strobe light to measure the frequencies. The results were as follows:

Stainless Steel Tube

Calculated (in Water)	Measured (in Water)
$f_1 = 12.1 \text{ Hz}$ $f_2 = 75.5 \text{ Hz}$	11.8 Hz ---

Rubber Tube

Measured (in Air)
$f_1 = 4.4 \text{ Hz}$ $f_2 = 7.5 \text{ Hz}$ $f_3 = 13.5 \text{ Hz}$ $f_4 = 20.5 \text{ Hz}$

Due to added mass effect, the actual natural frequencies in the water will be substantially lower.

3.2.2 Damping and Displacement

Displacements in both the lift and drag directions were measured simultaneously by mounting two Entran EGA series accelerometers inside the steel pipe and double integrating the output signals. Both accelerometers were encapsulated for protection and mounted near the free end of the pipe where the maximum displacements occur (Figure 5).

FIGURE 4. NATURAL FREQUENCY MEASUREMENT

FIGURE 5. ACCELEROMETER MOUNTING

The structural logarithmic decrement of damping in the fundamental mode was measured by displacing the pipe to various amplitudes in air and observing the amplitude decay of the integrated accelerometer output signal. Structural damping in the second mode is more difficult to measure. The first mode structural logarithmic decrement of damping is assumed here to be applicable to the second mode as well.

The fluid logarithmic decrement of damping for the steel pipe was determined by displacing the pipe in water and observing the amplitude decay. Knowing the structural decrement of damping, the still water fluid damping may be inferred. Although the fluid damping has been observed to vary [9] with the reduced flow velocity, $V_r = U/Df_n$, where f_n is the natural frequency of the structure, it is generally assumed that the still water value may be applied over the range of reduced flow velocities.

From the time response in still water, the log decrement was determined to be .01 - .04. The spread is due to the low damping of the system and uncertainties in amplitude measurements. The equivalent mass including the internal water and added mass was calculated to be 1.26×10^{-3} slugs/ft.

3.2.3 Force Measurements

Force measurements were made at the cantilevered end of the pipe using the load cell arrangement shown in Figure 6. Only one component of the force was measured at a time. The purpose of using two load cells is to cancel any effects of the moment on the output signal. Since one cell is in tension (positive signal) when the other is in compression (negative signal), the moment effects may be canceled by taking the difference between the output signals. The true force is then one half the difference.

FIGURE 6. PIPE AND LOAD CELL SUPPORT

From equation (7), the first mode natural frequency of the water-filled pipe was estimated to be approximately 10 Hz, and the second mode frequency was estimated to be about 70 Hz. The 70 Hz value was used in selecting the desired response characteristics of the load cells. Resolution of the load cells was determined by assuming that the forces of interest are of the order of the total static drag at the first mode resonant condition. The drag was calculated by assuming a uniform loading and a C_D of 1.

Considering the design requirements, a Kistler model 912H quartz load cell was selected. The Kistler load cell has a natural frequency of 70 kHz, a rigidity of 20×10^{-8} in/lb, and a resolution of 0.002 lb.

3.2.4 Flow Monitoring

A Laser Doppler Velocimeter (LDV) velocity measuring system available at CalTech was used to measure the channel flow velocity. An LDV is unique in that it allows velocity measurements to be made without disturbing the fluid. A rough sketch of the basic system used is shown in Figure 7. The laser Doppler measures the fluid velocity component perpendicular to a plane normal to the plane of the two laser rays and which bisects the angle between the rays. The beat frequency of the light scattered from minute particles moving with the fluid through the focal volume is picked up by the photomultiplier. The velocity component at the focal volume is calculated from

$$U_1 = \frac{\lambda}{2 \sin \frac{\theta}{2}} v_D$$

where the wave length of the laser λ , the angle θ between the beams, and the measured beat frequency v_D .

FIGURE 7. SCHEMATIC OF LASER DOPPLER VELOCIMETRY SYSTEM.

The Laser Doppler could be most useful in establishing the spanwise correlation lengths of the shedding vortices by splitting each of the beams once more and focusing the two sets of beams in the fluid. The two channel system would be particularly useful when investigating the response of scaled CWP's to shear layers. However, implementation of such a system was beyond the scope of the present study.

3.2.5 Signal Processing and Data Acquisition

A block diagram of the signal processing and data acquisition system for the experiment is shown in Figure 8. One principle component of the overall system is the Hewlett-Packard Data Acquisition System. The HP system was programmed to manipulate raw digitized data recorded on floppy discs for permanent record. For example, the output from the accelerometers was numerically integrated and the resulting displacements plotted. The frequencies of the displacements and forces were measured by feeding the outputs of the accelerometers and load cells to a standard spectrum analyzer.

3.2.6 Photographic Equipment

Motion pictures were taken of the response of the experimental pipes to various flow conditions from the side and the bottom of the glass walled FSWT. The first set of experiments (steel pipe) were filmed by the CalTech photographer using a high speed 16mm Fastex camera. Difficulties with the lighting were encountered when the camera was operated at the low filming speeds (1000 fps) required here; therefore, the rubber pipe experiments were filmed with an 8mm home movie camera.

FIGURE 8. SIGNAL PROCESSING AND DATA ACQUISITION SYSTEM

3.3 Facility Description

The CalTech Free Surface Water Tunnel (FSWT) has a closed circuit circulation system lying in a vertical plane (Figure 9). The working section (Figure 10) is 20 inches wide, 30 inches deep and 96 inches long. Normally, the water depth is kept at 20 inches, and the flow speed ranges from 1 to 20 ft/sec. Centerline axial turbulence levels are less than 0.8%. Plexiglass side windows allow observation and photography of test models. A full complement of instrumentation, data recording, data processing, photography and machine shop equipment and facilities is available for test support. Details of the facility are available in Reference 10.

FIGURE 9. FREE SURFACE WATER TUNNEL SCHEMATIC

FIGURE 10. CALTECH FREE SURFACE WATER TUNNEL TEST SECTION

4.0 RESULTS AND DISCUSSION

The principal results from the experiments are a set of movies demonstrating the in-line and cross-flow response of a relatively stiff tube made of stainless steel and a flexible tube of rubber weighted on the lower end. Measurements from the movies were made of maximum free end deflection, oscillation frequency and mode shape. For the steel pipe, displacements and frequencies were obtained from the accelerometers. A simple harmonic motion of the pipe is assumed such that

$$|x|_{\max} = |a|_{\max} \omega^2$$

where $|x|_{\max}$, $|a|_{\max}$, and ω are, respectively, the maximum displacement, maximum acceleration and the circular frequency of vibration in either the in-line (drag) or cross-flow (lift) direction. Results are presented as maximum amplitude (cylinder diameters) as a function of reduced velocity for both excitations since much of the existing data are so presented.

Substantial evidence has been shown in small scale experiments that the maximum displacement amplitude of flow-induced oscillations of circular cylinders are limited to 1-2 diameters. The principal non-dimensional parameters governing the structural response are the reduced velocity

$$V_r = \frac{V}{f_n D}$$

and the reduced damping

$$k_s = \frac{2m\delta}{\rho D^2}$$

where v = fluid velocity far from the cylinder (assumed uniform)

f_n = structural natural frequency of the relevant mode of response

δ = log decrement of oscillation in the fluid of interest

D = cylinder diameter

m = equivalent cylinder mass per unit length including the internal water mass and the added mass (assumed independent of flow velocity)

ρ = fluid mass density

A mathematical wake-oscillator model (e.g. Reference 11) based on the van der Pol oscillator has been used extensively in an attempt to explain the experimental data, particularly that the amplitude of motion is limited. Although all available detailed experiments have considered fundamental mode response due to great difficulties obtaining higher modes in the laboratory, there is evidence that higher modes flow induced oscillations do occur as evidenced by limited large scale observations. For the flexible rubber cylinder undergoing higher mode response, the maximum amplitude is also related to V_r and k_s by properly scaling the normal mode shapes of the structure. Since the experimental evidence for the higher mode response is rather skimpy and the mathematical model is not based on fundamental principles, the present experiments that were designed to be reasonably certain for OTEC CWP design that higher mode amplitudes are also limited.

4.1 Steel Tube

The results for the stainless steel pipe are summarized in Table 1a as measured from the movies, and in Table 1b as deduced from the accelerometers. The damping ratio from the

TABLE 1A. STAINLESS STEEL PIPE RESPONSE (MOVIES)

Run	U_{∞} (fps)	f_s (Hz)	f_D (Hz)	f_L (Hz)	$2X_{D/D}$ Peak to Peak	$2X_{L/D}$ Peak to Peak	Comments
1	0.29	3.3	12.22 (LC)	---	--	--	150 Frames/S
2	0.40	4.6	12.2 (LC)	---	--	--	Small motion
3	0.55	6.3	9.2	9.2	0.1	0.1	Coupled motion
4	0.77	8.9	---	9.2	--	0.5	Lateral
5	0.96	10.9	---	9.2	--	1.5	
6	1.07	12.3	---	9.2	--	2	
7	1.23	14.0	17.8	8.9	0.5	2	200 Frames/S
8	1.32	15.1	17.8	8.9	0.25	2	
9	1.60	18.3	---	---	--	--	500 Frames/S
10	2.08	23.8					
11	2.52	28.8					
12	3.02	34.5					
13	4.31	39.9					Permanent set in tube
14	4.31	49.3					

TABLE 1B. STAINLESS STEEL PIPE RESPONSE (ACCELEROMETERS)

Run	V_r	$ a_D _{\max}$ in/s ²	$\frac{x_D^*}{D}$	$ a_L _{\max}$ in/s ²	$\frac{x_L^*}{D}$	$ LC _{\max}$ (Drag)	f_D Hz	f_L Hz	f_{LC} Hz	Comments
1	1.37								12.2	
2	1.89	52	.045	21	.018	.038	11.8		11.8	
3	2.60	193	.15	93	.063	.17	13	13.3	13	
4	3.64	Erratic		98	.12	.074		10	10, 15.4	
5	4.53	170	.05	490	.59		20	10		
6	5.05	380	.11	740	.89		20	10		
7	5.81	1400	.42	1400	1.69	0.9	20	10.5		Erratic
8	6.23									
9	7.56f ₁ 1.22f ₂	2500	.054	1700	.91	1.2	71.4	15, 74.7	71.4	Lift at 15 Hz modulated by 74.7 Hz

*Based on $a_{D,L} = |a_{D,L}|_{\max} \cos \omega t$

$$|x_{D,L}|_{\max} = \omega^2 |a_{D,L}|_{\max}$$

free oscillation measurements was determined to be in the range .05 - .17. Maximum tip displacement is shown as a function of the reduced velocity in Figures 11 and 12 for the in-line and cross-flow motions. Two sets of points are shown in each graph, one with V_r based on the measured natural oscillation frequency of the fundamental mode and the other based on the observed oscillation frequency.

The two sets of reduced data are not significantly different for the cross flow response and are in agreement with previous observations showing resonance in the range $3 \leq V_r \leq 7$ and peak amplitudes near 2. The in-line results indicating tube response at almost twice the natural frequency are not understood. Previous observations show resonance at values of V_r in the range 1.5-4 based on the structural natural frequency as might be expected since the drag excitation frequency is twice that in the lift direction. The maximum amplitude of about .25D is somewhat greater than observed previously, possibly due to the lower damping ratio, k_s , in the present experiments.

Second mode responses were observed by the drag and lift accelerometers for flow velocities greater than about 1.6 fps. The corresponding Strouhal frequency at 1.6 fps is 18.3 Hz about 1/4 of the observed response frequency of 71.4 Hz. The inferred amplitude from the accelerometer readings is only about 5% of the diameter in the drag direction. The lift accelerometers also detect the higher frequency response in addition to the fundamental mode. It is not known now the higher mode is excited. However, the amplitude of the second mode is not large. Permanent deformation of the tube limited the maximum velocities tested so that full second mode resonance could not be achieved.

Load cell readings in the drag directions are also included in Table 1b representing the maximum of the AC component. Whether the continual increase with flow velocity is due

FIGURE 11. TIP DISPLACEMENT FOR IN-LINE MOTION

FIGURE 12. TIP DISPLACEMENT FOR CROSSFLOW MOTION

entirely to in-line oscillations or possible misalignment and consequent pick up of lift motion was not investigated. Analytical calculations of beam response would be useful to help interpret these results.

4.2 Rubber Tube

Movie observations of the rubber pipe were used to determine mode shapes and displacements (Table 2). Strouhal frequencies close to the third mode natural frequency were obtained and higher mode responses were observed in the drag and lift direction. Multiple modes changing with time restricted the quantitative data; however, maximum displacements were never more than about 0.5D.

Very flexible cold water pipes require a weight at the bottom to hold the pipe down in the current and to keep the pipe in tension. Although no attempt was made to scale a particular design to the present flexible pipe test, an important observation was made. The weight was observed to undergo significant oscillations (translational and rotational) which could impose unacceptable cyclical loads on a CWP attachment. This area clearly requires more attention.

TABLE 2. RUBBER TUBE RESPONSE (MOVIES)

Run	U_{∞} (fps)	f_s (Hz)	f_D (Hz)	f_L (Hz)	$2X_D/D$	$2A_L/D$	Comments
1	0.09	0.86	---	0.6	--	0.5	1 Pendulum Mode
2	0.32	3.07	6.1	3.2	0.2	0.15	2 Looks second mode plus pendulum with similar amplitude.
3	0.43	4.13	4.0	3.6	0.05	0.3	3 Very erratic in drag direction. Lift is regular first mode.
4	0.59	5.66	4.7	3.6	0.2	0.5	4 Cross between 1st and 2nd mode in drag.
5	0.87	8.35	Erratic	5.4	0.05	0.25/0.5	5 Weight motion dominating
6	1.15	11.04	Erratic+2nd	6.1	0.05	0.5	
7	0.41	3.94	None Some twist	2.9	--	0.4	Strumming Suppression First mode
8	0.88	8.44	Erratic	5.4	0.1	0.3/0.5	
9	1.21	11.62	Mostly lift	6.5	--	0.5	

5.0 CONCLUSIONS AND RECOMMENDATIONS

Principal observations in the present experiments of significance for OTEC CWP design consideration are as follows:

- Fluid-induced oscillations of a flexible cylinder rigidly mounted at one end are limited in the fundamental mode maximum displacement to about 1.5 to 2 diameters.
- Higher mode oscillations appear limited as well as the fundamental mode based on the rubber tube experiments.
- Higher mode excitation occurs at flow velocities well below the calculated resonance velocity although the displacements are not large.

These initial experiments were limited by funding and facility constraints. Laboratory experiments can contribute detailed information under controlled test conditions and should serve as a precursor to any large scale tests contemplated. Two areas of interest come to mind:

- The correlation length for vortex shedding on a flexible (oscillating) cylinder in a shear flow is highly controversial. Experimental measurements using an LDV could be conducted in a flow channel with scaled models of OTEC designs if a suitable shear layer could be produced. Based on experience gained in the present tests, a fruitful program would require about one manyear of support.
- If vortex strumming proves to be a fatigue problem for the OTEC CWP (some designs appear to sufficiently relieve the loads), vortex suppression devices should be tested in a laboratory. Little fundamental information is available for proposed suppression concepts (i.e., shrouds, strakes, fairings) so performance estimates are based on empirical results from limited flow regimes. A four month program at a two man level would provide guidance to larger scale demonstration experiments.

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