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HYDRODYNAMIC DESIGN LOADS FOR THE OTEC COLD WATER PIPE

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September 1978

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Science Applications, Inc.
El Segundo, California

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HYDRODYNAMIC DESIGN LOADS
FOR
THE OTEC COLD WATER PIPE

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SEPTEMBER 1978

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ALBUQUERQUE • ANN ARBOR • ARLINGTON • BOSTON • CHICAGO • DENVER
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FOREWARD

This program was conducted for the National Oceanic and Atmospheric Administration under Contract NDBO-03-78-G03-0505 to provide engineering design loads for the OTEC Cold Water Pipe. Dr. Terence McGuinness was the technical monitor.

The SAI principal investigator was Duane Hove. Dr. Edward Albano and Dr. W. Shih were the key technical contributors. SAI consultants Dr. Donald Coles and Dr. Anatol Roshko contributed valuable advice.

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LIST OF SYMBOLS

a	amplitude of motion
c	speed of sound
f	frequency
g	gravity
k	dimensionless frequency
m	mass
q	dynamic pressure
r	radius
t	time
x	lateral coordinate
z	axial coordinate, depth
A	pipe cross-sectional area
C	Fourier coefficient
ζ	damping matrix
C_D	drag coefficient
C_F	shear stress
C_L	lift coefficient
C_M	mass coefficient
C_N	Fourier coefficient
C_P	pressure coefficient
D	diameter
E	elastic modulus
F	force
G	structural damping coefficient

H wave length
 I moment of inertia
 K roughness height
 $\bar{K}$ stiffness matrix
 K Keulegan-Carpenter number
 L length
 $\bar{M}$ mass matrix
 N mode number
 P pressure
 R correlation coefficient
 Re flow Reynolds number
 Re_I Reynolds number independence
 Re_K roughness Reynolds number
 S Strouhal number
 S_N Fourier coefficient
 T period
 U velocity
 V_c critical flow velocity
 x lateral displacement
 z fluid variable
 β dimensionless velocity gradient
 ζ fraction of critical damping, amplitude of fluid variable
 η dimensionless amplitude
 θ angular coordinate
 λ correlation length

μ mass ratio
 ν kinematic viscosity
 ξ beam modes
 ρ density
 σ stress
 τ dimensionless time
 ω frequency

Superscripts

- differential with respect to time
- second differential with respect to time
- ' fluctuating component

Subscripts

o stationary cylinder, ambient
 ∞ freestream
c center, cosine term
f fluid
i initial, internal
n mode number
s shedding frequency, sine term
B base
CR critical
D diameter
EFF effective
M maximum
N mode number
RMS root mean square

ABSTRACT

Ocean current and/or plant motion crossflows induce time dependent hydrodynamic loads on the OTEC cold water pipe due to vortex shedding. Design criteria were established for mean and fluctuating loads based on a review of the literature, analysis of test data acquired by SAI under a previous experimental program and an analytical extension of test results to higher Reynolds number. Baseline loads were specified for rigid cylinders in uniform flows. Modifications to the loads by current shear, stratification and cylinder motion, were investigated and final design criteria established. Limited structural response calculations were performed to demonstrate the use of the design criteria and to investigate briefly the possible structural response mode. Comparisons were made with alternate hydrodynamic loads, and recommendations were made for experimental verification.

1.0 INTRODUCTION

Ocean Thermal Energy Conversion (OTEC) power plants are being designed to generate power from the ocean vertical temperature gradients. Warm surface water vaporizes ammonia to drive a turbine, and cold subsurface water recondenses the ammonia for return in a closed cycle. The major design proposals (TRW¹, Lockheed², Carnegie Mellon³, Anderson and Anderson⁴, University of Massachusetts⁵, Applied Physics Laboratory⁶) all suggest a floating plant with a long slender pipe to bring the cold water from the ocean depths to the surface. OTEC power plants will operate in either a station keeping mode held in place against the currents by an anchor or a dynamic positioning system or in a slow grazing motion to avoid thermal contamination in quiescent waters (Figure 1). Ocean current and/or plant motion crossflows will induce hydrodynamic loads on the cold water pipe (CWP), and engineering design loads are required for pipe structural and positioning system development.

Circumferential pressure variations will exist on the CWP in crossflows because the fluid alternately decelerates, accelerates and then decelerates as it flows around the pipe. In an ideal inviscid potential flow, no net force would exist on the pipe, but in real fluids the viscous boundary layer causes the flow to separate from the pipe leaving a low pressure base region which produces a net drag (Figure 2). Flow separation is time dependent with alternate shedding vortices creating oscillatory pressure distributions which result in fluctuating drag and lift (normal) forces. The separation process is not amenable to theoretical analysis and requires the development of empirical coefficients based on simulation experiments.

There exists an extensive body of published literature for rigid cylinders in uniform flows from which to draw baseline crossflow design loads (see bibliography for a

STATIONARY MODE

GRAZING MODE

FIGURE 1. REPRESENTATIVE POWER PLANT CONFIGURATIONS

FIGURE 2. CYLINDER PRESSURE DISTRIBUTIONS

partial listing). However, there is a large data scatter in the literature⁷, and peak fluctuating loads are totally lacking. OTEC CWP crossflow Reynolds numbers may reach 10^8 requiring an extension of existing test data⁸. Furthermore, OTEC CWP design loads must consider the influence of pipe motion (referred to here as hydroelastic response), current shear and stratification for which much less information is available.

Other sources of hydrodynamic loads include surface waves, internal waves and internal flows. Surface waves can provide direct loading on the CWP if the waves extend to sufficient depths; internal waves due to density gradients within the ocean can also produce currents although the induced loads are expected to be small compared to current and surface wave loads. Internal flow CWP loads arise from pressure gradients due to viscous losses and density differences as well as from plant induced transients (the water hammer effect).

1.1 Objectives

The overall objective of the present program was to establish hydrodynamic load design criteria for the OTEC CWP. Specifically, the study concentrated on developing mean and fluctuating circumferential pressure distributions, integrated drag and lift forces and load spectra for currents. Special attention was focused on establishing peak instantaneous loads which were not previously available. In addition, the influence of current shear, stratification, and cylinder motion on the current loads was estimated. For comparison purposes the hydrodynamic forces due to surface waves, internal waves and internal flows were estimated.

1.2 Scope

Engineering design criteria for an elastic beam model and a shell model of the CWP were established based on a review

of the cylinder flow literature, analyses of test data obtained on a previous SAI test program and an analytical extension of test results to higher Reynolds number. Limited structural response calculations were performed to demonstrate the use of the design criteria and to investigate briefly the possible structural response modes.

2.0 BACKGROUND

The OTEC CWP must withstand the stresses resulting from all hydrodynamic loadings. An analytical model development program is underway at the NOAA Office of Ocean Engineering to develop computerized structural analysis methods that can be used to design the CWP. The present program was undertaken to define the hydrodynamic loads in a form that can be input to the computer models. Both beam analysis and shell analysis methods are considered.

2.1 Cold Water Pipe Configurations

Proposed cold water pipe design concepts have ranged from stiff to compliant, and attachments techniques have considered both fully rigid and gimbaled mounts. Lockheed has proposed telescoping sections of concrete pipe, while TRW suggests a constant diameter fiber reinforced plastic pipe. Other proposed configurations include rubber pipes, fabric pipes and a stockade concept. To reach a temperature difference of 20°C the pipe may be on the order of 1000 meters long, and pipe diameters of 2 to 50 meters are being considered for 1 to 400 MWe plants.

The recent design trend has been toward flexible pipes to relieve the bending stresses. Thin walled structures are necessary to keep the cost and weight down, and ring stiffeners might be used to increase the CWP cross-sectional strength. Engineering estimates⁹ of the ring designs have indicated a height of 10^{-1} diameters and a spacing of 1 to 3 pipe diameters. Longitudinal stiffeners have been suggested¹⁰; however, their use could contribute substantial increases in drag¹¹.

The OTEC cold water pipe will of necessity have external surface roughness either due to manufacturing or to marine biofouling, and surface roughness is known to influence cylinder

force coefficients. Surface finishes for cast concrete might be on the order of 10^{-4} pipe diameters while metal plate welds might be on the order of 10^{-3} pipe diameters. Although accurate predictions of the roughness height due to marine bio-fouling are not available, 10^{-4} to 10^{-3} pipe diameters has been suggested. If barnacle clumps occur, the roughness height might be 10^{-2} diameters.

2.2 Ocean Current Environments

OTEC plants will operate in currents which are a function of time, depth and location. For the stationary mode plant, the NOAA CWP design package¹² has specified a surface current of 1.8 meters per second and a normalized current profile (Figure 3) equal to the sum of the geostrophic, tidal and wind driven current.

2.3 Hydrodynamic Load Characteristics

The key environment parameter affecting the current induced loads on a CWP is the flow Reynolds number

$$Re = \frac{UD}{\nu} \quad (1)$$

based on diameter. For smooth cylinders Reynolds number effects are imparted through the boundary layer transition from laminar to turbulent flows with attendant effects on flow separation. For subcritical Reynolds numbers ($Re < 2 \times 10^5$) on smooth cylinders, the boundary layer remains laminar until separation from the cylinder at approximately 80 degrees from the stagnation line. In the critical regime ($2 \times 10^5 < Re < 5 \times 10^5$) the flow transitions to a turbulent state in the detached boundary layer causing the flow to reattach; this rearward motion of the separation line (to 140 degrees) produces a sharp decrease in mean drag. As the Reynolds number increases in the supercritical regime ($5 \times 10^5 < Re < 3 \times 10^6$) the boundary layer

FIGURE 3. NORMALIZED CURRENT PROFILE

transitions to turbulent flow successively more forward of the separation line accompanied by gradual forward movement of the separation line to about 120 degrees. Further increases in Reynolds number in the transcritical regime ($Re > 3 \times 10^6$) cause a slow forward motion of the transition line and, therefore, have little effect on the separation line and the drag.

In the subcritical regime and again in the transcritical regime the cylinder flowfield is characterized by well defined alternate shedding vortices¹³. The effect of the vortices is to modify the pressure distribution around the cylinder in an orderly fashion producing a periodic lateral force referred to as lift. For the OTEC maximum design current the flow Reynolds numbers for a 9 meter diameter CWP (suggested for a demonstration plant) are almost entirely in the transcritical regime (Figure 4), thus simplifying the design load specification. Furthermore, the effect of surface roughness¹⁴ is to lower the flow Reynolds number at which the transcritical regime is achieved (Figure 5). Thus, for a nominal surface roughness on the order of 10^{-3} diameters, the nine meter CWP would experience transcritical flow for surface currents as low as 0.3 m/sec (1 ft/sec).

Circumferential pressure distributions are required for shell type analysis. Integrated pressure loads provide drag (in line with the flow) and lift (normal to the flow) forces required for beam analysis. Although viscosity plays a role in the integrated forces through the boundary layer separation influence on the inviscid flowfield, viscous shear stress is a small part of the total force as demonstrated by Achenbach¹⁵ (Figure 6). Therefore only integrated pressure forces are treated here.

2.4 Potential Structural Response Modes

Structural analysis of the OTEC cold water pipe requires both the mean and fluctuating hydrodynamic loads. The pipe

FIGURE 4. FLOW REYNOLDS NUMBER PROFILE FOR A NINE METER DIAMETER CWP

FIGURE 5. FLOW REGIME DEPENDENCE ON ROUGHNESS

FIGURE 6. SHEAR STRESS CONTRIBUTION TO CYLINDER DRAG

designs must be analyzed for bending, buckling and ovaling to insure structural integrity. Beam analysis methods are being computerized by Hydronautics, Inc. in a companion program to perform bending analyses; lift and drag forces are required as inputs. Shell analysis methods are necessary for ovaling and buckling calculations and require the specification of the sectional pressure distribution.

Fluctuating hydrodynamic loads arising from vortex shedding can excite dynamic structural response phenomena including rigid body and flexible motion. A simplified analysis¹⁶ indicates the cold water pipe bending and ovaling natural frequencies are at or above the vortex shedding frequency for selected design alternatives. Pipe natural frequencies near the shedding (exciting) frequency raise the spector of load and response enhancement, and care should be taken to avoid such a design. With a pipe natural frequency greater than the shedding frequency, the pipe will respond to instantaneous loads and, therefore peak values of the pressure distributions are required.

The spectral content of the forcing functions are important to dynamic structural response analysis and estimation of fatigue life. The vortex shedding process is expected to be periodic or quasi-periodic, and a single characteristic loading frequency is defined. The shedding process may be random (broad-band) over a portion of the Reynolds number range if the CWP is smooth; here statistical response analysis is required based on the spectral content. Also required is a measure of the vortex shedding correlation length along the cylinder axis. If the correlation lengths are long compared to the cylinder diameter, sectional analyses are valid, and the loads act in concert along the pipe. If short correlation lengths are present, the structural analysis must be three-dimensional, and the total loads on the pipe are diminished.

3.0 HYDRODYNAMIC LOADS IN CURRENTS

Crossflow induced hydrodynamic loads are described here in terms of circumferential pressure distributions and integrated forces (drag and lift). Mean values are presented first followed by the fluctuating component. Some detail regarding the source of the data is discussed where a foundation has not previously been established. Recommended values are noted for each parameter; conservative values were selected where no rational basis could be found for selecting other values. Only Reynolds numbers greater than 10^6 are considered, and both smooth and rough cylinders are considered even though the CWP is expected to be rough. Uniform flow results are established as a baseline, and the effects of shear, and cylinder motion were investigated to provide final design criteria.

3.1 Uniform Flows Over Rigid Cylinders

3.1.1 Circumferential Pressure Distributions

The pressure distribution around a cylinder is usually expressed in terms of a nondimensional pressure coefficient

$$C_P = \frac{P - P_\infty}{\frac{1}{2} \rho_f U_\infty^2} \quad (2)$$

For smooth cylinders the mean pressure data of Jones et al.¹⁷ reach the highest Reynolds numbers (Figure 7). Fair agreement (Figure 8) is seen with the data of other investigators¹⁸ including that of SAI¹⁹.

For rough cylinders, the mean pressure data of Achenbach and SAI are in good agreement even though the Reynolds numbers are not matched (Figure 9). The most notable characteristics of the pressure distributions for rough cylinders are the decrease (absolute) in the pressure minimums and the base pressures.

FIGURE 7. SMOOTH CYLINDER MEAN PRESSURE DISTRIBUTION (JONES, CINCOTTA, WALKER)

FIGURE 8. SMOOTH CYLINDER PRESSURE DISTRIBUTIONS

FIGURE 9. ROUGH CYLINDER PRESSURE DISTRIBUTIONS (CORRECTED)

Mean pressure distributions appear reasonably well known although a systematic variation with Reynolds number was not evident; the band indicated in Figure 10 covers the variations for Reynolds numbers from 1×10^6 to 17×10^6 and surface roughness to 10^{-3} diameters. The recommended mean pressure distribution (Table 1) was selected based on the consideration that roughness probably can't be specified accurately and that the largest pressure minimum provides the greatest circumferential bending moments. This approach is believed to be conservative.

Instantaneous pressure distributions (Figure 11) deviate from the mean curve due to the unsteady nature of vortex shedding. Pressure fluctuation data at a given station taken by SAI on a previous program were found to be normally distributed about the mean for smooth and rough cylinders at $Re = 1.2 \times 10^6$ (Figure 12). The data distributions (bars) were determined by cataloging 1000 data samples into bins which represent one tenth of the range between the observed maximum and minimum, and the normal curve was constructed from the measured RMS value. Other stations indicated similar distributions.

The apparent normal distribution of the fluctuating pressure data has important implications. By simply specifying an RMS value, the maximum deviations from the mean can be calculated with a desired degree of confidence (e.g., 2σ estimate of loads). Furthermore, only RMS data are reported in the literature. The assumption of a normal distribution therefore enables peak loads to be estimated from the available data.

The highest Reynolds number fluctuation data for smooth cylinders is due to Van Nunen¹⁸ (Figure 13); test data obtained from James and Malcolm²⁰ showed the same magnitudes but indicated the opposite trends with Reynolds number. The only known pressure fluctuation data for rough cylinders is from the SAI experiments (Figure 14). The unusually high leading

FIGURE 10. MEAN PRESSURE DISTRIBUTION

TABLE 1. CIRCUMFERENTIAL PRESSURE COEFFICIENTS

Angle From Forward Stagnation Point (Degrees)	Pressure Coefficient
0	1.00
10	0.91
20	0.63
30	0.13
40	-0.45
50	-1.13
60	-1.75
70	-2.20
80	-2.50
90	-2.55
100	-2.35
110	-1.75
120	-0.95
130	-0.70
140	-0.68
150	-0.68
160	-0.68
170	-0.68
180	-0.68

FIGURE 11. INSTANTANEOUS PRESSURE DISTRIBUTIONS

FIGURE 12. PRESSURE FLUCTUATION DISTRIBUTIONS ($\theta = 84^\circ$)

FIGURE 13. SMOOTH CYLINDER RMS PRESSURE FLUCTUATIONS (VAN NUNEN)

FIGURE 14. ROUGH CYLINDER RMS PRESSURE FLUCTUATIONS (SAI)

edge fluctuations are thought to be due to freestream turbulence caused by the cylinder wake returning in the closed cycle wind tunnel. Unfortunately, no turbulence measurements were taken during the tests. However, when shifted down to a leading edge value, the data are not greatly different from the smooth cylinder data. Because of the sparsity of data, the solid curve bounding the observed values is recommended (Figure 15).

Pressure fluctuations for smooth cylinders are found to have no particularly strong frequency content in the supercritical regime. Jones refers to the spectra as wide band random, and the SAI data (Figure 16) normalized in terms of Strouhal number

$$S = \frac{Df}{U} \quad (3)$$

reflect this characteristic at $Re = 1.2 \times 10^6$. However, the addition of uniform roughness with $K/D = 10^{-3}$ produced a single shedding frequency at $Re = 1.2 \times 10^6$ characteristic of alternate shedding vortices (Figure 17). This behavior is consistent with the transcritical flow regime mapping of Szechenyi (Figure 5). Thus for the OTEC CWP surface roughness, fluctuating pressure loads due to vortex shedding can be represented by a simple periodic function.

To complete the description of the pressure fluctuations, the relationship between fluctuations at different locations is required. The only available information for smooth and rough cylinders at $Re > 10^6$ was obtained from the SAI tests at $Re = 1.2 \times 10^6$. Here a correlation coefficient

$$R = \frac{1}{T} \int_0^T \frac{C'_P(\theta_1, t) C'_P(\theta_2, t)}{C_{P_{RMS}}(\theta_1) C_{P_{RMS}}(\theta_2)} dt \quad (4)$$

was calculated for each of the twelve available data channels. The results are typified by Figure 18 which presents the correlation of fluctuations at various circumferential stations with the 84° station. At this slightly supercritical Reynolds number

FIGURE 15. RMS PRESSURE FLUCTUATIONS

CALTECH RUN 6 UNSTEADY CENTERLINE DATA RE DIAM=,126E07
RMS PRESSURE AT ANG 54 DEG = 0.064

FIGURE 16. POWER SPECTRAL DENSITY - SMOOTH CYLINDER

FIGURE 17. POWER SPECTRAL DENSITY - ROUGH ($K/D = 10^{-5}$) CYLINDER

FIGURE 18. CIRCUMFERENTIAL CORRELATIONS WITH 84° LOCATION

the fluctuations appear uncorrelated for the smooth cylinder. However for the rough cylinder the fluctuations on one side are well correlated, and those on the opposite side are anticorrelated consistent with the idea of alternate shedding vortices characteristic of the transcritical regime.

For most OTEC applications, pressure fluctuations should be assumed in phase on one side of the CWP and 180° out of phase on the opposite side of the cylinder. If the CWP is truly smooth, the pressure fluctuations may be uncorrelated for supercritical Reynolds number regimes.

3.1.2 Integrated Forces

Cylinder mean drag coefficients have been the subject of experimental investigations for many years. Early investigators experienced difficulties in reaching high Reynolds numbers due to a lack of adequate facilities, and there was a tendency to force high Reynolds numbers in wind tunnels by exceeding prudent blockage ratios. Only recently the effects of surface roughness have been sorted out. Thus, the large scatter in test data for smooth cylinders (Figure 19) can be accounted for in a rational manner, and the recommended curve⁷ is based on the most modern experiments. The drag force on a cylinder expressed in terms of a nondimensional drag coefficient

$$C_D = \frac{\text{Drag}}{\frac{1}{2}\rho U^2 A} \quad (5)$$

was determined by integrating the pressure distribution

$$C_D = \frac{1}{2} \int_0^{2\pi} C_P \cos\theta \, d\theta \quad (6)$$

At sufficiently high Reynolds numbers the mean drag coefficients for rough cylinders become independent of Reynolds

FIGURE 19. CYLINDER MEAN DRAG COEFFICIENTS

number⁸ and are a function only of the relative roughness K/D. The recommended approach is to use the rough cylinder curve of Figure 19 for Reynolds numbers greater than the critical Reynolds number (2×10^5).

Little data are available in the literature for fluctuating drag coefficient. However, the magnitudes of the fluctuating components are small compared to the mean drag values (Figure 20) which was confirmed by the SAI data. A recommended value is not prescribed since the peak values can be accounted for by simply increasing the mean drag loads by twenty percent. However, care should be taken that fluctuating drag loads (twice the vortex shedding frequency) do not excite dynamic structural response modes.

Fluctuating lift coefficients have been reported in the literature in terms of

$$C_L = \frac{\text{Lift}}{\frac{1}{2}\rho U^2 A} \quad (7)$$

where the lift coefficients have been determined from

$$C_L = \frac{1}{2} \int_0^{2\pi} C_P \sin\theta \, d\theta \quad (8)$$

Again analysis of the SAI test data indicated that the lift coefficient magnitudes are normally distributed over the range between the maximum (positive) and minimum (negative) values (Figure 21). Thus, the lift coefficient magnitudes are completely specified by an RMS value, and design values have been recommended previously⁷ (Figure 22).

FIGURE 20. RMS DRAG COEFFICIENT

FIGURE 21. DISTRIBUTION OF LIFT COEFFICIENTS

FIGURE 22. CYLINDER RMS LIFT COEFFICIENTS

Spectral content of the integrated loads has received more attention than for the pressure fluctuations. Jones et al.¹⁷ found that for smooth cylinders power spectra were broad band in $10^6 < Re < 3.5 \times 10^6$, narrow band in $3.5 \times 10^6 \leq Re \leq 6 \times 10^6$ and quasi-periodic for $Re > 6 \times 10^6$ (Figure 23). The SAI data agree with Jones for smooth cylinders (Figure 24) at $Re = 1.2 \times 10^6$ and with Szechenyi's assertion that periodic fluctuations reoccur in the transcritical regime (Figure 25). A single Strouhal shedding frequency has been recommended (Figure 26) as a function of Reynolds number for smooth cylinders and following Szechenyi as a function of roughness Reynolds number

$$Re_K = Re \frac{K}{D} \quad (9)$$

for rough cylinders keeping in mind that smooth cylinder loads may be broad band.

The spanwise correlation length is a measure of the two-dimensionality of the vortex shedding process. Long correlation lengths (say 10 diameters) imply an organized flow while short correlation lengths are associated with random vortex shedding.

Szechenyi¹⁴ found correlation coefficients

$$R = \frac{1}{T} \int_0^T \frac{f(z_1, t) f(z_2, t)}{\sqrt{f^2(z_1, t)} \sqrt{f^2(z_2, t)}} dt \quad (10)$$

to be a strong function of the flow regime (Figure 27). His calculated correlation lengths

$$\lambda = \int_0^1 R\left(\frac{z}{D}\right) d\left(\frac{z}{D}\right) \quad (11)$$

FIGURE 23. SMOOTH CYLINDER LOAD SPECTRAL CHARACTERISTICS (JONES ET AL.)

CALTECH RUN 6 UNSTEADY CENTERLINE DATA RE DIAM=.126E07
RMS LIFT COEFFICIENT = 0.133

FIGURE 24. POWER SPECTRAL DENSITY - SMOOTH

FIGURE 25. POWER SPECTRAL DENSITY - ROUGH ($K/D = 10^{-3}$)

FIGURE 26. CYLINDER STROUHAL SHEDDING FREQUENCY

FIGURE 27. VORTEX SHEDDING SPANWISE CORRELATIONS

were about 1 for supercritical flows and 10 for transcritical flows. OTEC cold water pipes are expected to be in the transcritical regime for all but a few cases and the recommended approach is to assume fully correlated loads.

3.2 Effect of Shear and Stratification

Thus far, discussion of the external hydrodynamic loads on the rigid cylinder has assumed uniform crossflow over the length of the CWP. Actual currents vary with depth as shown in Figure 3 for the recommended design current profile. A limited number of studies reported in the literature (Table 2) indicated that drag forces, local pressure distributions and distribution of shedding frequencies on cylinders and D-sections are modified due to the presence of shear in the on-coming stream. Only linear (or nearly linear) profiles have been considered; the important parameters are shown in Figure 28. The nondimensional velocity gradient β for the recommended OTEC current profile is less than about 0.02, and typical Reynolds numbers based on the mean current velocity are in the range 3×10^6 to 1.6×10^7 . The range of parameters tested and type of data available are limited; however, some general observations may be made:

- Spanwise flow induced
 - Reduces C_D at high velocity end
 - Increases C_D at low velocity end
- Discrete cells along span with constant S (based on U_c) in each cell.
- Changes in C_p related to spanwise cell distribution.
- Number of cells dependent on aspect ratio and end configuration.
- Effects of freestream turbulence on the base pressure C_{pB} variation with Re similar to uniform flow.

TABLE 2. SUMMARY OF SHEAR FLOW INVESTIGATIONS

Investigator	$Re \times 10^{-4}$	β	Comments
Maisch and Moore	1.85-2.25	0.058-0.117	Total force; local pressure distribution
Shaw and Starr	1.5	0.02 - 0.18	Pressure distributions
Davies	5.7 -25	0.18	Base pressure, oil flow
Mau11	2	0.022	Flat plate; spanwise correlation in wake
Mau11 and Young	2.85	0.022,0.029	D-sections; full span, free end. Base pressure, PSD's
Mair and Stansby	1.3 -2.6	0.014-0.05	D-sections, end plates. Base pressure, PSD's
Stansby	0.7, 0.92	0,0.025,0.05	Oscillating cylinder; PSD's

DEFINITIONS

TYPICAL OTEC SITE

$$Re = 3 \times 10^6 - 1.6 \times 10^7$$

$$\beta \leq 0.02 \text{ (LINEAR APPROXIMATION)}$$

FIGURE 28. STRUCTURES IN SHEAR FLOW - DEFINITIONS

- No unsteady lift measurements.
- Lock-on of vortex shedding for oscillating cylinders observed in cell region whose frequency is near oscillation frequency. Other cells not locked-on. Cell size estimated by Stansby model.

The local spanwise drag coefficient variation²¹ due to shear decreases with decreasing β (Figure 29). The slope of the C_D vs spanwise location curve (Figure 30) implies a relatively small spanwise variation in C_D (no more than $\pm 10\%$ over the entire span) for $\beta \approx 0.02$. In the regime of constant drag coefficient (independent of Reynolds number) appropriate to the Maisch and Moore²¹ experiments for subcritical flow and the anticipated rough OTEC CWP in transcritical flow, the experimental scatter in C_D measurements exceed the spanwise variation due to shear. It is therefore consistent to assume that C_D is constant along the cylinder evaluated at a Reynolds number based on the mean velocity of the shear flow. The local drag force should be based on the local dynamic pressure given by the local cross-flow velocity. Spanwise variations in pressure coefficient distribution around the cylinder are considered unimportant for OTEC applications, and the single recommended curve introduced earlier is to be used. Again, the local dynamic pressure is used to evaluate the pressure magnitude.

No force measurements have been made for oscillating cylinders in shear flows, although limited investigations have considered lock-on for small amplitude oscillations. Observations detect spanwise cellular structure within which the shedding frequency is constant and only certain discrete cells that have shedding frequencies close to the oscillation frequency are locked-on to it. The extent of the discrete cells is estimated by an empirical relation due to Stansby,

FIGURE 29. SPANWISE DRAG VARIATION IN SHEAR FLOW (MAISCH AND MOORE)

FIGURE 30. SPANWISE DRAG VARIATION IN SHEAR FLOW

$$\frac{\Delta Z}{D} = \frac{4\eta}{\beta} \quad (12)$$

where

ΔZ = cell extent

D = cylinder diameter

η = oscillation amplitude in cylinder diameters

For typical OTEC values, $\eta \sim 1$, $\beta \leq .02$, the cell extent is of the order of the length of the CWP. Thus, a spanwise uniform shedding frequency based on a mean velocity is recommended. A more detailed discussion is given in Section 3.3.1.1.

In summary, the recommended procedure for specifying loads for the CWP in shear flows is as follow,

Rigid Cylinders

- C_D - constant along span at mean shear Re
- C_P - recommended for uniform flows
- Use local dynamic pressure
- Strouhal frequency constant along span evaluated at mean velocity
- Same C_L as uniform flow of mean velocity

Oscillating Cylinders

- Use amplification factors established for uniform flow
- Use constant spanwise shedding frequency either lock-on or as for rigid cylinder.

Stratification effects will inhibit the pressure driven spanwise flow due to buoyancy effects. Since the recommended approach for shear flows is to ignore them, stratification will not affect the recommended loads whether or not stratification is of significant influence.

3.3 Effect Of Cylinder Motion

The fluid forces on a stationary cylinder in a current or wave are themselves time dependent, and the basic fluid-mechanical equations (the unsteady Navier-Stokes equations) which govern the problems already involve time. In principle, then, prediction of the force on a moving cylinder would involve little more effort than would be needed in an analytical solution for the forces on a stationary cylinder. Unfortunately, Navier-Stokes solutions are not available at the Reynolds number of interest in OTEC CWP applications, and predictions of hydrodynamic loading on cylinders must be based on approximating fluid mechanical equations (potential theory, boundary layer theory) and on experiment.

The analysis of loading due to surface waves, which is approached in the framework of potential theory, is discussed in Section 4.1 of this report. In the present section the discussion is restricted to loading due to currents. Understanding of this area has developed principally through the study of experimental results. Flows over stationary cylinders have been reviewed above; for cylinders which are mechanically driven or which respond dynamically, the experimental data base is not very extensive, but from it some relatively simple and practical analytical models have been developed.

From the point of view of characterizing hydroelastic forces, the most fruitful experimental work has involved the measurement of pressure, lift and drag on transversely oscillated cylinders in a wind tunnel. Figures 31 and 32 indicate the scope

of available results from such experiments. For OTEC applications the following ranges are of interest:

$$10^6 \leq Re \leq 10^8$$

$$0.1 \leq \frac{fD}{U} \leq 0.5$$

$$0.2 \leq \frac{a}{D} \leq 2$$

and it is seen that the data does not cover this area. However, analytical and experimental work at SAI⁸ with flows over stationary cylinders indicates that results from subcritical Reynolds number experiments ($<10^5$) can be applied to the transcritical Reynolds number regime where the OTEC CWP will operate.

The hydrodynamic forces on a cylinder vibrating in a current depend on the cylinder motion, which itself depends on the force, and so the prediction of cylinder structural response cannot be obtained accurately without solving the coupled, "hydroelastic", problem of response of structure and fluid. To date, the most successful approach to modeling the coupled problem is that of the so called wake oscillator model. Less accurate but perhaps more practical bounds on structural response may, however, be achieved through the use of models which ignore the dependence of hydrodynamic force on structural response or which treat this dependence in a simple, more approximate way. Such approaches are recommended below; their application must be accompanied by sufficient experimental verification.

FIGURE 31. Laterally Oscillating Cylinder Test Regimes - Frequency

FIGURE 32. LATERALLY OSCILLATING CYLINDER TEST REGIMES - AMPLITUDE

The structural response of the cold water pipe may be described by modelling it as a beam or as a shell. Hydrodynamic loading for a beam analysis involves only the force perpendicular to the pipe axis; this force is resolved into two components; drag, in the direction of the relative current, and lift, perpendicular to drag. Treatment of the pipe as a shell takes into account the deformation of its cross section and requires a knowledge of the circumferential distribution of pressure, or at least the first few terms in a Fourier series representation of the pressure distribution.

A complete shell model of the pipe includes all of the effects covered by a beam model but involves more detailed (and expensive) structural analyses. The greatest uncertainty in the results of a shell analysis would probably be due to the accuracy with which motion-induced local pressures can be specified. Preliminary analytical estimates indicate that stresses associated with shell-type deformations are not great except for designs which involve highly flexible or low-strength materials, so that the necessity for a shell model of the CWP would depend upon the particular design.

The sections which follow discuss separately the hydrodynamic loadings involved in beam response and shell response.

3.3.1 Beam Mode

The lift and drag forces exerted on a stationary cylinder are time dependent. The drag deviates little from its time averaged value, and the lift fluctuates with a principal frequency measured by the Strouhal number S_o and amplitude measured by the stationary cylinder lift coefficient C_{L_o} .

If the cylinder oscillates laterally at a Strouhal frequency close to the stationary shedding frequency, the mean drag may be significantly enhanced. Griffin²² has shown that this amplification is a function of a Strouhal number where the length scale is $D + 2a$ and a is the amplitude of the motion. (Figure 33). The principal effect on unsteady loading is to modify both the amplitude and frequency of the lift coefficient. Most of the experimental work to measure this effect has involved rigid body translation of spring-mounted cylinders, although some testing has been done with cylinders performing bending oscillations. Broadly speaking these tests establish that when the cylinder oscillates at a frequency within 20 or 30% of the static shedding frequency, the loads and responses behave as follows:

- (i) The frequency of the oscillating lift force tends to "lock on" to the frequency of cylinder oscillation;
- (ii) The amplitude of the oscillating lift coefficient is greater than its static value C_{L0} , and depends nonlinearly on the amplitude of cylinder motion;
- (iii) The phase of the lift, measured relative to the cylinder vibration, depends on the amplitude of cylinder motion;
- (iv) Flexibly mounted cylinders driven by shedding in the locked on condition oscillate with an amplitude which generally is less than two diameters, and which is a function of the product $\mu\zeta$ where μ is the mass ratio (i.e., the ratio of the cylinder mass per unit length to the mass per unit length of a cylinder of fluid of the same diameter) and ζ is the fraction of critical damping in the structure.

FIGURE 33. DRAG AMPLIFICATION DUE TO CYLINDER OSCILLATION AT SHEDDING FREQUENCY

As was suggested by Bishop and Hassan, this behavior can be modelled by a simple dynamic oscillator containing nonlinearities. Hartlen and Currie²³ developed a model equation of the Van der Pol type, and this "wake oscillator" model has been elaborated upon by a number of investigation, including Skop and Griffin,²⁴ Griffin, et.al.,^{25,26} and Iwan and Blevins²⁷. The model was originally used to describe the lateral rigid-body translations of a flexibly mounted cylinder, but recently Iwan²⁸ has extended it to account for bending oscillations (see also Griffin and Koopman²⁹) and has shown that its results correlate well with measured responses (Figure 34).

Iwan's theory can be adapted to a structural dynamics approach, such as the finite element method, which employs matrix methods to calculate the bending displacements at stations along the pipe. The basic structural response equation is taken in the usual form:

$$\underline{\ddot{M}X} + \underline{\dot{C}X} + \underline{KX} = \underline{F}(t) \quad (13)$$

where

$$\begin{aligned} \underline{X}(t) &= \text{vector of nodal displacements} \\ \underline{F}(t) &= \text{vector of fluid forces} \\ \underline{M} &= \text{mass matrix} \\ \underline{C} &= \text{damping matrix} \\ \underline{K} &= \text{stiffness matrix} \end{aligned}$$

The natural modes $\underline{\xi}_n$ and frequencies ω_n are solutions to the homogeneous equation (omitting damping) formed by

$$\underline{X}(t) = \underline{\xi}_n e^{i\omega_n t} \quad (14)$$

so that

$$(-\omega_n^2 \underline{M} + \underline{K}) \underline{\xi}_n = 0 \quad (15)$$

FIGURE 34. CORRELATION OF PEAK RESPONSE AT STRUMMING
(IWAN, 1975)

The arbitrarily normalized equations have the property

$$\underline{\xi}_n \cdot \underline{M} \cdot \underline{\xi}_k = \begin{cases} 0 & \text{if } k \neq n \\ m_n & \text{if } k = n \end{cases} \quad (16)$$

$$\underline{\xi}_n \cdot \underline{K} \cdot \underline{\xi}_k = \begin{cases} 0 & \text{if } k \neq n \\ \omega_n^2 m_n & \text{if } k = n \end{cases} \quad (17)$$

where m_n is the "generalized mass in the n^{th} mode". Following the usual mode expansion method, the response to $\underline{F}(t)$ is assumed in the form

$$\underline{X}(t) = \sum \underline{\xi}_n \bar{x}_n(t) \quad (18)$$

and, with the assumption of proportional damping, the equations for the response coefficients x_n of each mode become

$$\ddot{\bar{x}} + 2 \zeta_n \omega_n \dot{\bar{x}} + \omega_n^2 \bar{x} = m_n^{-1} \underline{\xi}_n \cdot \underline{F}(t) \quad (19)$$

This much is standard linear structural response theory. For the submerged (and fluid containing) pipe, the mass matrix $\underline{M}$ should contain the "apparent mass" of the fluid, accounting for its inertial reaction. Iwan's model takes the remaining force F at any station to be proportional to the difference between $\dot{x}$, the lateral structure velocity, and a variable $\dot{z}$ which measures the lateral fluid velocity:

$$F = a_r \rho D U (\dot{z} = \dot{x}) \quad (20)$$

(Note: This is force per unit length, and to be consistent all terms in the structures response equations should be similarly dimensioned; an alternative would be to multiply by a delta-length term to keep units of force.)

The fluid variable is assumed to be governed by a nonlinear equation, the forced Van der Pol equation:

$$\ddot{z} + \omega_s^2 z - \frac{a_1 - a_4}{a_0} \frac{U}{d} \dot{z} \frac{a_2}{a_0} \frac{z^3}{Ud} = \frac{a_4}{a_0} \frac{U}{d} \dot{x} \quad (21)$$

where ω_s is the stationary cylinder shedding frequency and $a_0 - a_4$ are parameters of the model. Some fluid mechanical theory for this equation is presented in the paper by Iwan and Blevins²⁷. Alternative equations have been proposed; all of them represent oscillators with nonlinear stiffness and/or damping terms. Iwan's equation above is probably the simplest, most well known and best correlation of the wake oscillator models. Based on calculations with experiment, Iwan gives the model parameters as $a_0 = 0.48$, $a_1 = 0.44$, $a_2 = 0.20$ and $a_4 = 0.38$. Following Iwan's approach the vector $\underline{z}$ of fluid variables at each station is expanded in the natural modes

$$\underline{z}(t) = \sum \underline{\xi}_j(t) \quad (22)$$

Substituting into the right hand side of the structural response Equation (19) yields a series proportional to

$$\sum_j \underline{\xi}_n \cdot \underline{\xi}_j \left(\dot{\underline{z}}_j(t) - \dot{\underline{x}}_j(t) \right) \quad (23)$$

in general the product $\underline{\xi}_n \cdot \underline{\xi}_j$ is not zero but in order to simplify the analysis it is assumed negligible except when $n = j$. This assumption is not necessary to linear structure analysis but it does have the advantage of "uncoupling" the structure equations. Similarly, an approximation is made in dealing with the nonlinear damping term when the modal expansion of $\underline{z}$ is substituted into Equation (21) for the dynamics of the fluid variable.

This yields (see Iwan 1975) the series

$$\sum_i \sum_r \sum_p \xi_n^p \frac{M^{pr}}{m_n} \xi_i^r \left(\sum_k \xi_k^r \frac{\dot{z}}{k} \right)^2 \frac{\dot{z}}{z_r} \quad (24)$$

which is approximated by $p_n \frac{\dot{z}^3}{z_n^3}$ where

$$p_n = \sum_r \sum_p \xi_n^p \frac{M^{pr}}{m_n} \left(\xi_n^r \right)^3 \quad (25)$$

(The notation here uses superscripts to identify the element of the vector or matrix.) Often the mass matrix M is diagonal, so that the sum becomes

$$p_n = \sum_r \frac{M^{rr}}{m_n} \left(\xi_n^r \right)^3 \quad (26)$$

it is also convenient to work with dimensionless variables by writing

$$\bar{X} = D \cdot \bar{\eta}, \quad \bar{Z} = \cdot \bar{\zeta}, \quad t = \frac{D}{U} \tau \quad (27)$$

$$k_n = \omega_n \frac{D}{U}$$

then with the above approximations, the equations for the response in the n^{th} mode become

$$\bar{\eta}'' + 2 \zeta_T k_n \bar{\eta}' + k_n^2 \bar{\eta} = \frac{a_4}{\mu_n} \bar{\zeta}' \quad (28)$$

$$\bar{\zeta}'' + k_s^2 \bar{\zeta} - \frac{a_1 - a_4}{a_0} \bar{\zeta}' + \frac{a_2}{a_0} p_n \bar{\zeta}'^3 = \frac{a_4}{a_0} \bar{\eta}' \quad (29)$$

where

$$\frac{1}{\mu_n} = \frac{\rho D^2}{m_n} \xi_i^2, \quad \zeta_T = \zeta_n + \zeta_f, \quad \zeta_f = a_4 / 2\mu_n k_n \quad (30)$$

and

$$(\cdot)' = \frac{d}{d\tau} (\cdot) \quad (31)$$

Iwan²⁸ describes a means of obtaining an approximate solution to this system. Writing

$$\begin{aligned} \bar{\zeta}(\tau) &= \zeta \cos k\tau \\ \bar{\eta}(\tau) &= \eta_c \cos k\tau + \eta_s \sin k\tau \end{aligned} \quad (32)$$

where k is the dimensionless response frequency, it follows that

$$\bar{\eta}' = \frac{\eta_c}{\zeta} \bar{\zeta}' + k \frac{\eta_s}{\zeta} \bar{\zeta} \quad (33)$$

using this relation in the system equations gives

$$\eta_c = \frac{\hat{c}k}{\mu_n} a_4 \zeta \quad (34)$$

$$\eta_s = \frac{\hat{s}k}{\mu_n} a_4 \zeta \quad (35)$$

where

$$\hat{c} = 2 \zeta_T k k_n / \left[(k^2 - k_n^2)^2 + (2 \zeta_T k k_n)^2 \right] \quad (36)$$

$$\hat{s} = (k^2 - k_n^2) / \left[(k^2 - k_n^2)^2 + (2 \zeta_T k k_n)^2 \right] \quad (37)$$

and $\bar{\zeta}(\tau)$ satisfies

$$\begin{aligned} \bar{\zeta}'' - a_0^{-1} (a_1 - a_4 + \frac{k\hat{c}}{\mu_n} a_4^2) \bar{\zeta}' + a_0^{-1} a_2 p_n \bar{\zeta}'^3 \\ + (k^2 - k^2 \hat{s} a_0^{-1} a_4^2 / \mu_n) \bar{\zeta} = 0 \end{aligned} \quad (38)$$

The theory of the Van der Pol equation provides an approximate solution. The frequency is the solution of a cubic equation in k^2

$$k^2 = k_s^2 - k^2 \hat{s} a_0^{-1} \frac{a_4^2}{\mu_n} \quad (39)$$

and the amplitude is given by

$$\zeta = \frac{1}{k} \left(\frac{4}{3} \frac{a_1 - a_4 + k\hat{c} \frac{a_4^2}{\mu_n}}{a_2 p_n} \right)^{\frac{1}{2}} \quad (40)$$

The maximum response is attained where the natural frequency k_n is equal to the shedding frequency k_s and the correlation for this situation made by Iwan (Figure 34) provides a high degree of confidence in the model. (Note: Iwan defines $\mu_r = m/\pi r^2$ and shows results where m is the structural plus added mass per unit length, and m is uniform spanwise. For beam modes with uniform m the terms corresponding to p_n have been worked out analytically.)

For analysis of the OTEC cold water pipe, there is of course interest in responses where $k_n \neq k_s$. Experiments have shown substantial responses in the lock-on condition where the difference between the natural and shedding frequencies is about 20 - 30%. Then the response frequency k is close to k_n .

In order to illustrate this effect calculations using the wake oscillator model were performed for a possible CWP design, summarized in Figure 35. The pipe is assumed built in to a stationary platform, and cantilever modes are provided by the well known beam solution. Taking the shedding Strouhal number to be $S_o = 0.2$, flow speeds U_n such that $S_o = S_n \equiv \omega_n d / 2\pi U_n$ are labelled "critical velocities" in the Figure. Structural damping is assumed to be zero, and continuous beam modes $\xi(z)$ are normalized for maximum value unity.

Solution for modal response amplitudes and frequencies in the first three modes are shown in Figures 36 and 37. It is seen that peak displacements are substantial for current velocities U such that $0.75 < U/U_n < 1.4$. For response in the fundamental mode this corresponds to $0.43 < U < 0.60$ ft/sec. Since flow speeds of this order can be expected from ocean currents, this example indicates that current induced oscillatory pipe response can be a significant problem. For stiffer structures, however, it should be possible to keep natural frequencies above the vortex shedding frequency.

3.3.1.1 Limitations

The wake oscillator models were derived under the assumption that the pipe diameter and local current velocity are uniform along the length of the pipe. However, for the OTEC environment the current speed varies with depth, and some proposed CWP concept involve pipes of nonuniform diameter. In

GRP	$E = 2 \times 10^6$ PSI
DIAMETER	50 FT
LENGTH	3000 FT
WALL THICKNESS	0.8 FT
MASS RATIO	2
STRUCTURAL DAMPING	0.

CANTILEVER FREQUENCIES Hz	CRITICAL VELOCITY FT/SEC
.00229	0.57
.01432	3.58
.04009	10.02
.	.
.	.
.	.

FIGURE 35. TYPICAL CASE

FIGURE 36. MODAL RESPONSE - GRP EXAMPLE

FIGURE 37. RESPONSE FREQUENCY - GRP EXAMPLE

Section 3.2 a constant shedding frequency was recommended. The unsteady shedding process under these conditions is not fully understood, but the experimental work of Stansby³⁰ with cylinders oscillating in shear flows indicates that a cellular structure is set up in the wake, with each cell having a distinct shedding frequency. Only one of these cells is locked on, and a formula for estimating its spanwise extent Δz has been proposed

$$\frac{\Delta z}{D} = 4 \frac{\eta}{\beta}$$

where η is the dimensionless amplitude of oscillation and

$$\beta = \frac{D}{U} \frac{dU}{dz}$$

is the dimensionless velocity gradient

Models to predict the fluid forces in locked on conditions in shear flows have not been developed, and experimental measurements of forces under these conditions are lacking. An estimate of the significance of shear in an OTEC environment can be had by calculating the size of the locked on region assuming $\eta = 1$, $S_n/S_o = 1$. Since design current profile have $\beta \approx 0.02$, the above equation provides $\Delta z/D \approx 200$. This indicates that the locked on region should extend the length of the pipe. The effect of pipe taper, if any, should also be taken into account, but its tendency will most likely be to make the ratio D/U more uniform.

In summary, the indications are that for OTEC design environments, the effects of spanwise uniformities will not be as great as to negate the utility of the wake oscillator model as a design analysis tool.

3.3.1.2 Simplified Approach

Results of the wake oscillator model may be applied in an approximate manner which involves retaining only the structural response equation and providing a means for determining the fluid-induced damping and the oscillatory forcing function. This is achieved by noting that the term $a_4 \rho D U Z$ on right hand side of Equation (20) defines an effective lift force amplitude:

$$C_{L_{EFF}} = 2 a_4 k \zeta$$

where ζ is the amplitude of the "fluid variable" and k is the dimensionless response frequency. The fluid-induced damping ζ_f is given by Equation (30). Figure 38 shows results for the above example.

A survey of results for conditions of mass ratios of interest in OTEC conditions and making the conservative assumption of zero structural damping yields a "lift amplification curve" illustrated in Figure 39. The engineer can apply these results to a structural dynamic analysis by following the procedure set forth in Figure 40. Although the method may overestimate responses in certain conditions, the approach avoids the necessity for solving the wake oscillator equations. The rationale behind the simplified approach is that (1) at lock on the response frequency will be very close to the natural frequency and deflections should be less than obtained with zero structural damping and (2) when no lock on is predicted, the stationary cylinder shedding forces will induce some structural response. Appropriate values of the stationary lift coefficient amplitude C_{L_0} and Strouhal Number S_0 , provided in Section 3.1.2 can be used directly in the simplified model.

More "exact" procedures may of course be utilized following the wake oscillator model, but it is expected that use of the approximations recommended here will simplify design analysis efforts without underestimating response magnitudes.

FIGURE 38. EFFECTIVE MODAL FORCE - GRP EXAMPLE

FIGURE 39. LIFT AMPLIFICATION

MODE BY MODE $S_N \equiv \frac{F_N^D}{U} = S_N(z)$

1. IF $S_N(z)$ FALLS WITHIN 30% OF S_0 , THEN

$C_L(z)/C_{L_0}$ FROM LIFT AMPLIFICATION CURVE

$\omega = \omega_N$ = DRIVING FREQUENCY

ζ_{FN} FROM IWAN MODEL

2. IF NO $S_N(z)$ QUALIFIES, THEN

$C_L = C_{L_0}$

$\omega = \omega_0$

ζ_{FN} FROM IWAN MODEL

FIGURE 40. RECOMMENDED ANALYSIS FOR NONUNIFORM D/U

3.3.2 Shell Mode

Most full scale design concepts of the cold water pipe provide for thickness-to-diameter ratios which are small enough to characterize the pipe as a thin shell ($t/d \lesssim 10^{-2}$). Particularly for flexible materials (plastics, elastomers) consideration should be given to responses where the cross-sectional shape of the pipe deforms, whether statically or dynamically, and where failure may occur in buckling, fatigue or exceedance of ultimate stresses.

The deformation of cylindrical shells may be adequately characterized by specifying the radial displacement $U = U(\theta, z, t)$ where θ and z are angular and axial coordinates respectively, and t is time. Where end effects may be neglected, a "ring approximation" may be applied, writing $U = U(\theta, t)$ and assuming no variation with z . The discussion here proceeds in the context of the ring approximation.

The natural modes of deformation of the cross section of a thin shell are the cosine and sine functions of $n\theta$. Figure 41 illustrates the first few modes, where it is seen that the mode corresponding to $n = 1$ involves pure translation, and the ovaling mode corresponds to $n = 2$. It is the ovaling which is the fundamental mode of shell deformation; its mechanism is one of shell wall bending and generally it occurs at a lower frequency than the "breathing" mode ($n = 0$) which involves shell wall extension.

The excitation of shell ovaling modes by vortex sheddings in a current is an area which has not been sufficiently explored. Johns and Sharma³¹ have reported an important experiment involving model testing of thin cylindrical shells in a wind tunnel. They described hydroelastic responses where the response frequency (approximately equal to the natural fre-

$$U = \cos N\theta \quad (\sin N\theta)$$

$N = 0$

$N = 1$

$N = 2$

$N = 3, \dots$

$N = 0$ BREATHING. GENERALLY HIGH FREQUENCY. (WALL EXTENSION)

$N = 1$ BEAM MODE. (LATERAL BENDING)

$N = 2$ OVALLING, AND HIGHER (TO $N = 6$) OBSERVED. (JOHNS & SHARMA).
(IN-PLANE BENDING)

FIGURE 41. SHELLS: IN-PLANE MODES

quency of the responding mode) is essentially an integral multiple of the static shedding frequency (see Figure 42 which illustrates various vortex shedding alternatives corresponding to the mode $n = 2$). In their experiments, Johns and Sharma observed the responses indicated in Figure 43; with the response frequency to shedding frequency ratio R ranging to 6 and n to 4.

From the theory of flexural vibrations of thin rings³² as modified using linear potential theory to give the added mass due to the motion of the internal and external water¹⁶, a formula for the natural frequencies in the various motions when the modulus E is replaced by $E(1 - \nu^2)$. Setting the corresponding Strouhal number equal to the product RS_0 , where S_0 is the stationary cylinder shedding Strouhal number, yields a formula for the resonant responses of the shell in ovaling (Figure 44). This formula can be used to map out resonance areas as illustrated for example in Figure 45, where fixed values were taken for the shell material constants and for the shedding frequency. In this case it is seen that only very small values of thickness to diameter ratio will be of interest in dynamic ovaling. More flexible materials may, however, be encountered and a quick check of their design as to dynamic ovaling may be made by using Figure 46 where a band of low-order resonances has been plotted in terms of thickness to diameter ratio and current velocity rendered dimensionless by a velocity formed of the elastic properties of the shell material and the density of water. (The mass of the shell wall turns out to be negligible in this approximation.) The labeled points correspond to current speeds of 1, 5, and 10 feet per second past a shell with the properties $t/D = 0.016$, $E = 2 \times 10^6$ psi, $\nu = 0.3$ and with $S_0 = 0.2$. The elastic modulus here is that of glass reinforced plastic (GRP).

FIGURE 42. OVALLING EXCITATION MODES (JOHNS AND SHARMA, 1972)

		RATIO R					
		1	2	3	4	5	6
MODE N	1	X	X			X	
	2		X	X	X	X	
	3			X	X	X	
	4						X

FIGURE 43. HYDROELASTIC SHELL RESPONSE OBSERVATIONS
(JOHNS AND SHARMA)

"RING THEORY"

$N \geq 2$:

$$R^2 S_0^2 \approx \frac{2}{3\pi^2} N (1 - N^2)^2 \frac{E/\rho_F (1 - \nu^2)}{U^2} \left(\frac{T}{D}\right)^3$$

TO AVOID RESONANCES

INCREASE WALL BENDING STIFFNESS:

RAISE E , T/D

EXTERNAL STIFFENERS

MATERIAL MASS UNIMPORTANT

FIGURE 44. OVALLING "RESONANCES"

$$\begin{aligned}
 E &= 2 \times 10^6 \text{ PSI} \\
 \nu &= .3 \\
 S_0 &= .2
 \end{aligned}$$

FIGURE 45. POTENTIAL RESONANCES IN DYNAMIC OVALING

FIGURE 46. POTENTIAL RESONANCES - DYNAMIC OVALLING

The discussion thus far has concerned design to avoid hydroelastic "resonance" in shell wall ovalling. This is considered to be the most important aspect of the shell wall design in that the effects can be quite severe and also in that quantitative response predictions cannot be made with a high degree of confidence, given the present state of the art.

Shell design analysis also involves calculation of static response. Buckling failure can be estimated early using the well known thin shell formula. Figure 47 gives the result for the critical value of the difference between outer and inner pressure illustrated for the GRP example described above. Means to calculate the expected values of P_o and P_i are given in Section 4.3 below; it is enough to note here that their difference is not close enough to critical for buckling to be considered a serious problem for this example.

If the external pressure is considered imposed, then the linear ring theory provides a means of calculating both static and dynamic ovalling response, using the modal expansions. Following this technique, the forcing functions for each mode are simply the Fourier coefficients C_N , S_N of a series expansion of the pressure in the circumferential direction (Figure 48). It is noted that C_1 and S_1 are proportional to the drag and lift coefficients, respectively; higher order coefficients are generalized forces in the various modes which involve wall bending. Of primary interest here are the coefficients C_2 and S_2 which correspond to the first ovalling mode.

It appears that these force coefficients have not heretofore been measured experimentally for cylinders in high Reynolds number flow. In a recent wind tunnel test program conducted at the Caltech 10-foot tunnel SAI has recorded the time dependent pressure distributions about the circumference at one spanwise location of a stationary cylinder and has reduced this data to modal force coefficients.

BUCKLING

$$\left(P_0 - P_1 \right)_{CR} = 2 \left(\frac{I}{D} \right)^3 E (1 - \nu^2) = 14.9 \text{ PSI}$$

STATIC OVALLING

$$\sigma_2 = \frac{1}{4} \rho U^2 \left(\frac{b}{I} \right)^2 \bar{C}_2$$

AT 10 FT/SEC: $\sigma_2 = 880 \text{ PSI}$

FIGURE 47. STATIC RESPONSE ESTIMATES - GRP CASE

$$[C_N, S_N] = \frac{1}{\pi} \int_0^{2\pi} P(\theta) [\cos N\theta, \sin N\theta] d\theta$$

$C_1 \sim$ DRAG COEFFICIENT

$S_1 \sim$ LIFT COEFFICIENT

$C_2, S_2 \sim$ FIRST OVALING MODE

STATIC - TIME AVERAGE

"DYNAMIC" - DEVIATION FROM STATIC

FIGURE 48. SHELL RESPONSE: MODAL FORCES

Figures 49 through 51 show the power spectra of S_2 , S_3 , S_4 . It is seen that for these sine-modal forces, essentially all of the power is confined to a narrow band centered at the static shedding frequency. The spectra for the coefficients C_2 , C_3 , C_4 , reproduced in Figures 52 through 54, show a somewhat different character: The primary frequency is the same (S_0), but there is substantial power at integral multiples of S_0 and, apparently even at fractions of S_0 . This is explainable to some extent if it is recalled that θ is measured from an axis aligned with the flow. The sine function, being antisymmetric with respect to the flow, will measure projections of the pressure distribution displaying that antisymmetry and will thus emphasize the vortex-shedding character of the loading (positive pressure above when negative below). The cosine function, being antisymmetric with respect to the line $\theta = \pi/2$, will "pull out" the fore-aft pressure pulsations, and it should be expected that significant power in the cosine mode will be first seen at a frequency of $\frac{1}{2} S_0$, as for example in the spectrum for C_3 .

There is clearly a relation between the multiple peaks in the C_N power spectra (measured on a stationary cylinder) and the hydroelastic response modes of Johns and Sharma, discussed above. Exactly how these peaks are amplified by shell deformation is a question which has not yet been answered. One might expect that a nonlinear model, analogous to the wake oscillator theory, could be developed, but there would be required a substantially larger experimental data base than is presently available. In fact, it appears that no modal force measurements from dynamically deforming shells have been made.

FIGURE 49. CALTECH RUN 19 UNSTEADY CENTERLINE DATA RE DIAM = .121E07
RMS VALUE OF $S_2 = 0.056$.

FIGURE 50. CALTECH RUN 19 UNSTEADY CENTERLINE DATA RE DIAM = .121E07
RMS VALUE OF $S_3 = 0.033$

FIGURE 51. CALTECH RUN 19 UNSTEADY CENTERLINE DATA RE DIAM = .121E07
RMS VALUE OF S_4 = 0.044

FIGURE 52. CALTECH RUN 19 UNSTEADY CENTERLINE DATA RE DIAM = .121E07
RMS VALUE OF C_2 = 0.041

FIGURE 53. CALTECH RUN 19 UNSTEADY CENTERLINE DATA RE DIAM = .121E07
RMS VALUE OF C_3 = 0.025

FIGURE 54. CALTECH RUN 19 UNSTEADY CENTERLINE DATA RE DIAM = .121E07
RMS VALUE OF C_4 = 0.032

Another view of the response problem can be had by looking at the record of modal force as a function of time, as in Figure 55. There it is seen that S_2 may be approximated by a function with sinusoidal time dependence, and with slowly varying amplitude. For illustration, the peak value of the amplitude of S_2 is compared with the time averaged value of C_2 . It is seen that they are of the same order of magnitude. Dynamic stress levels in ovalling will therefore be comparable to static stresses.

To estimate the responses in a shell mode, linearized vibration theory may be applied. Figure 56 gives the results for modal displacement amplitude a'_N where the forcing function has reduced frequency $k = \omega D/U$ and amplitude C'_N (or S'_N). Here μ is the mass ratio and $\epsilon = 2$ for the fluid filled pipe. The reduced frequency of the natural mode is k_N , and a structural damping coefficient G has been assumed. Comparing with the formula for static response, the ratio of oscillating to static stress in mode N is seen to be the product of the ratio of oscillatory to static modal forces and the usual "amplification factor". To illustrate results, calculations were carried out for a GRP shell in mode $N = 2$ taking for mean and oscillating forces the values indicated above (Figure 57). It is seen that there is substantial amplification only for small thicknesses to diameter ratios.

In applying these well known linear methods to the dynamic ovalling problem, there should be serious concern for the overlooked modulation of the modal forces by hydroelastic effects. For want of a better means to predict these effects one can only suggest that the trends established in the beam response modes might be applied to shell ovalling: i.e., static cylinder values of oscillating modal forces should be multiplied by a factor of about 2.5 to account for hydroelastic

STATIC

$$\bar{S}_2 = 0$$

$$(\bar{C}_2 = 0.67)$$

DYNAMIC

FREQUENCY $F_D/U = 0.2$

AMPLITUDE $S'_2 \doteq 0.14$

FIGURE 55. SHELL MODAL FORCE $S_2(T)$

LINEAR THEORY

$$\frac{A'_N}{C'_N} = \frac{1}{2K_N^2 \left[\frac{1}{2} \mu \left(1 + \frac{1}{N^2} \right) + \frac{\epsilon}{N} \right]} \frac{1}{\sqrt{\left(1 - \frac{K^2}{K_N^2} \right)^2 + G^2}}$$

RATIO OF OSCILLATING TO MEAN STRESS

$$\frac{\sigma'_N}{\bar{\sigma}_N} = \frac{C'_N}{\bar{C}_N} \frac{1}{\sqrt{\left(1 - \frac{K^2}{K_N^2} \right)^2 + G^2}}$$

FIGURE 56. DYNAMIC RESPONSE ESTIMATES - GRP PIPE

FIGURE 57. RATIO OF OSCILLATING TO MEAN STRESS (MODE 2)

amplification; and a fluid damping term should be included in the modal response equations, following the beam model. This procedure is recommended only by way of analogy with what is known about beam response: no quantitative verification of hydroelastic shell response levels appears to exist. In studying a CWP design concept where shell wall ovaling resonances are indicated experimental work involving scale models would be necessary in order to provide quantitative results for response levels.

3.4 Design Criteria

Hydroelastic loads have been described in detail in the preceeding sections. Design load procedures are summarized here in the form of step by step examples first for the beam model and then for the shell model. The example CWP's are taken to be 30 foot constant diameter 3300 feet long cylinders with a surface roughness of 10^{-3} diameters. For demonstration purposes assume one design has a natural frequency of 0.06 rad/sec while the second has a natural frequency of 0.15 rad/sec. The OTEC plant is assumed anchored off Punta Tuna, Puerto Rico, and the current profile (Figure 58) can be approximated by

$$U = 0.6 + 5.4e^{-Z/833} + 0.014Ze^{-Z/500}$$

where U is in ft/sec and Z is in feet. The CWP will be suspended at a depth below the surface which depends on the particular concept. For this exercise the CWP is assumed to extend to the surface, and the current loads will be in excess of those seen in actual practice.

3.4.1 Beam Model Loads

Beam model analyses require lift and drag forces along the CWP. Loads are specified in terms of a mean drag force and

FIGURE 58. CURRENT VELOCITY PROFILE FOR PUNTA TUNA

a fluctuating lift (normal) force. Care should be taken to avoid structural resonance in the in-line direction where the (small) fluctuating drag term has a frequency twice the vortex shedding frequency.

1. Find mean current profile velocity

$$\bar{U} = \int_0^{3300} \frac{U(z)}{3300} dz = 3.0 \text{ ft/sec}$$

2. Calculate Reynolds number and roughness Reynolds number

$$\bar{Re} \approx 9 \times 10^6 \quad Re_K \approx 9000$$

3. Select a baseline mean drag coefficient from Figure 19

$$C_{D_o} = 0.95$$

4. Select a Strouhal number from Figure 22

$$S_o = 0.3$$

5. Calculate single shedding frequency for entire pipe using mean velocity

$$\omega_o = 0.19 \text{ rad/sec}$$

6. Check for vortex shedding lock-on to cylinder frequency (i.e., resonance)

Design 1- No

Design 2 - Yes

$$\frac{\omega_n}{\omega_o} = 0.3$$

$$\frac{\omega_n}{\omega_o} = .78$$

7. For no lock-on multiply mean drag coefficient by 1.2 to cover fluctuating drag

Design 1 $\bar{C}_D = 1.14$

For lock-on multiply mean drag coefficient by amplification factor in Figure 33. For oscillations on the order of a diameter a value of 2 should suffice

Design 2 $\bar{C}_D = 2.0$

8. Calculate drag forces/length along pipe (Figure 59) and total drag

$$\frac{dF_o}{dz} = \frac{1}{2} \rho U^2(z) D C_D$$

$$F_o = 29 C_D \int_0^{3300} U^2(z) dz$$

Design 1

Design 2

$$F_o = 1.3 \times 10^6 \text{ lbs} \quad 2.5 \times 10^6 \text{ lbs}$$

9. Select a baseline lift coefficient from RMS curves of Figure 22

$$C_{L_o} = 0.27$$

For 95% confidence choose 2 σ value

$$C_{L_o} = 0.54$$

10. For no lock on use baseline value

Design 1 $C_L = 0.54$

For lock on use amplification curve of Figure 39

Design 2 $C_L = 1.2$

FIGURE 59. CWP DRAG LOADS

11. Calculate lift force amplitude/unit length along pipe (Figure 60)

$$A_L = \frac{1}{2} \rho U^2(Z) D C_L$$

12. For no-lock on apply lift force/length at vortex shedding frequency

$$\text{Design 1 } f = A_L \cos 0.19t$$

For lock on apply lift force at CWP natural frequency

$$\text{Design 2 } f = A_L \cos 0.15t$$

13. Total lift forces can be found by integrating along the pipe (loads are fully correlated)

$$\text{Design 1 } f(t) = 6.9 \times 10^5 \cos 0.19t$$

$$\text{Design 2 } f(t) = 1.5 \times 10^6 \cos 0.15t$$

These unsteady loads are seen to be on the order of the mean drag loads.

3.4.2 Shell Model Loads

Shell model analyses require specification of the cross-sectional pressure distributions along the pipe. Pressure is specified in terms of a mean and fluctuating component.

1. Construct a single mean pressure coefficient distribution from Table 1 and multiply by dynamic pressure at depth (Z) (Figure 61)

$$P - P_\infty = \frac{1}{2} \rho U^2(Z) C_p(\theta)$$

2. Select baseline RMS pressure fluctuations from Figure 15. For 95% confidence choose 2σ value

$$P' = \frac{1}{2} \rho U^2(Z) C_p'$$

FIGURE 60. CWP LIFT LOADS

FIGURE 61. CWP CIRCUMFERENTIAL PRESSURE DISTRIBUTION

3. For no lock on use baseline fluctuating value superimposed on mean pressures (Figure 61). For lock on use amplification curve of Figure 39 and superimpose amplified fluctuating loads on the mean pressures.
4. Apply mean and fluctuating loads in the form $\bar{C}_p + C'_p \cos \omega t$. For no lock on use shedding frequency

Design 1 $(P - P_\infty)_{90^\circ} = -0.74 + 0.07 \cos 0.19t$

For lock on use pipe natural frequency

Design 2 $(P - P_\infty)_{90^\circ} = -0.74 + 0.15 \cos 0.15t$

Fluctuating loads are applied 180° out of phase on top and bottom of cylinder crosssection

Design 2 $(P - P_\infty)_{90^\circ} = -0.74 + 0.15 \cos (0.15t + \pi)$

4.0 COMPARISON OF ALTERNATE HYDRODYNAMIC LOADS

Surface and internal wave induced currents are sources of additional loads on the CWP and are considered in this section along with a discussion of possible effects due to internal pipe flows.

4.1 Surface Waves

Ocean waves exist to sufficient depths to affect the CWP. Wave induced forces on the CWP are complicated, but various analytical methods and empirical techniques based on experimental data are available to estimate the loads. The forces that arise are (1) the induced current flow around the structure which results in a drag (2) an inertia force due to the fluid acceleration (the added mass effect), and (3) a diffraction effect resulting in a low frequency periodic drift force. The interaction is coupled; however, estimates can be made based on a separation of effects.

Morison, et al.³³ assumed that the total force is obtained by adding linearly the drag component and the inertial force term with the assumption that the ratio of wavelength to CWP diameter is large so that diffraction effects are small:

$$F(z) = \frac{1}{2} \rho D C_D U |U| + \frac{\pi D^2}{4} \rho C_M \frac{\partial U}{\partial t} \quad (41)$$

where ρ is fluid density, C_D is drag coefficient, C_M is mass coefficient, and D is cylinder diameter. Note that the drag force reverses sign due to the oscillatory nature of the waves. Actually the drag and inertial terms are out of phase, but their scalar addition represents a maximum estimate.

The problem of wave force prediction is reduced to one of determination of the local fluid velocity U and acceleration $\partial U / \partial t$ due to wave motion and the drag and mass coefficients C_D and C_M . Appropriate wave theories give the velocities and

accelerations, and the drag and mass coefficients are obtained mainly through experiments. Keulegan and Carpenter³³ performed laboratory experiments on smooth circular cylinders immersed in standing waves and found that C_D was a function of the parameter $U_M T/D$ where U_M is the maximum horizontal particle velocity of the wave, T is the wave period, and D the cylinder diameter. More recently, Sarpkaya³⁴ has made measurements of C_D and C_M in oscillating flows at high Reynolds numbers on both smooth and rough cylinders. His values will be used in the present study. First consider the relative importance of the drag and inertia terms by considering the nondimensional form of Equation (41).

If

$$U(t) = U_M \cos \theta, \quad \theta = \frac{2\pi t}{T}$$

Then

$$\frac{F(z)}{\frac{1}{2}\rho U_M^2 D} = -C_D \cos \theta |\cos \theta| + \frac{\pi^2}{K} C_M \sin \theta \quad (42)$$

where the Keulegan-Carpenter number is related to the wave-height by

$$K = \frac{U_M T}{D} = \frac{\pi H}{D}$$

The Keulegan-Carpenter regimes³⁵ are identified as

$$\begin{aligned} K &< 10 \text{ inertia dominated} \\ 10 &< K < 30 \text{ drag } \sim \text{ inertia} \\ K &> 30 \text{ drag dominated} \end{aligned}$$

Since the instantaneous particle velocity is of the same order as typical steady currents, and C_D and C_M are $O(1)$, the wave induced loads are comparable to current loads in the upper portions of the pipe before the wave induced currents decay substantially. For typical CWP dimensions and even waves of significant height of 80', $K < 10$ so that the CWP will generally

be in the inertia dominated regime, and D/λ will be large enough such that wave diffraction becomes important.

Variations of cylinder drag coefficients as a function of surface roughness are shown in Figure 62 for various values of K . The underestimate in using steady flow values is evident. For $K/D \leq 10^{-3}$ the variation of C_D with K is small and may be neglected so that a single curve may be used for all values of K . For very small values of K in the inertia dominated regime, uncertainties in C_D will not significantly influence the total load estimate.

Variations of the inertia coefficient C_M with relative roughness (Figure 63) show a stronger dependence on K ; however with $K < 10$ for the OTEC CWP, an upper bound is given by the smooth cylinder value. Furthermore, inclusion of diffraction effects³⁶ reduces C_M . Hence the smooth cylinder value yields a conservative estimate of the inertia forces due to waves.

The maximum transverse lift coefficient defined by $C_L = \text{LIFT} / \frac{1}{2} \rho U_M^2 D$ showed little variation with K/D and Re within the data scatter so that a single curve has been chosen (Figure 64). The regimes for Reynolds number independence based on Sarpkaya's data for various K as a function of K/D are shown in Figure 65. For harmonic flows $Re > 10^6$ implies Reynolds number independence for C_D , C_M and C_L .

A summary of the recommended approach for calculating OTEC CWP surface wave loading is,

- Reynolds number independence assumed
- Calculated K based on mean U_M for wave velocity profile

FIGURE 62. DRAG COEFFICIENT IN HARMONIC FLOW
(REYNOLDS NUMBER INDEPENDENCE REGION)

- FIGURE 63. INERTIA COEFFICIENT IN HARMONIC FLOW
(REYNOLDS NUMBER INDEPENDENCE REGION)

$$S = \frac{FD}{U_M} = 0.22$$

$$C_L = \frac{\text{LIFT}}{\frac{1}{2} \rho U_M^2 D}$$

FIGURE 64. MAXIMUM TRANSVERSE LIFT COEFFICIENT IN HARMONIC FLOW

FIGURE 65. REYNOLDS NUMBER INDEPENDENCE REGION FOR HARMONIC FLOW

- C_D - use $K = 20$ curve for all $K \leq 20$, $Re \geq Re_I$
- C_M - use smooth cylinder value of 1.8 for all $K, K/D, Re \geq Re_I$
- $(C_L)_M$ - use single curve shown in Figure 60.
- Loads - use local velocity

4.2 Internal Waves

Internal waves are present wherever there is vertical density stratification. They contribute additional components in the environmental current field, and isopycnal (lines of constant density) displacements may change the ambient stratification. The effects of these velocity components are similar to steady currents, though they vary over time scales of minutes to hours and vertical length scales of tens to hundreds of meters. In most cases the effects of internal waves can be modeled by adding a velocity component that is varying in time and depth to existing calculations. Typical values for internal wave parameters³⁷ are

- Maximum amplitude of 1 - 100m
- Period of 4 min - 25 hr
- Wave propagation speed 0.05 - 1 m/s
- Wave lengths 0(kms)
- Currents normal to propagation < 0.01 m/s;
in direction of propagation 0.05 - 0.1 m/s

The induced currents are seen to be quite small compared to ocean currents or wave induced currents and can therefore be neglected. However, the period of the waves may induce low period modes of the CWP, and this should be considered in the structural analysis.

4.3 Internal Flows

Internal flows may influence the CWP and pump operation in a number of ways. The lower static pressure due to the

internal flow, friction losses and differences in internal and external water densities could lead to buckling instabilities for thin wall structures. Forces exerted by the fluid flow due to the water hammer effect can cause pipe deflections. Internal flows may cause buckling or oscillations of the pipe due to instabilities induced when the pipe and flow interact.

Consideration of Bernoulli's equation inside and outside the pipe results in the following expression for the pressure difference across the CWP wall at a depth z due to internal flow and difference in density between the ambient and cold water in the pipe

$$\Delta P(\theta) = \int_z^L (\rho_i - \rho_o) g Dz + \frac{1}{2} \rho_i U_i^2 + q_o C_p(\theta) \quad (43)$$

where

ρ_i = water density inside CWP

ρ_o = ambient water density

q_o = ambient dynamic pressure

$C_p(\theta)$ = local pressure coefficient distribution

The Westinghouse study⁹ showed that Equation (43) contributes 65-80% of the total pressure difference while viscous losses through wall friction contribute 2-10% and screens and pumps contribute 15-30%. Typical ΔP 's are of the order 0.01 to 0.1 atmosphere.

Internal flow induced instabilities³⁸ are estimated using the critical flow velocity for a cantilevered pipe

$$V_C = \frac{\pi}{2L} \left(\frac{EI}{\rho A} \right)^{\frac{1}{2}} \quad (44)$$

Internal flows approaching this value may cause buckling instabilities. For a given cross section, the critical velocities for various materials are

<u>Material</u>	<u>V_c (fps)</u>
Steel	143
GRP	37
Rubber	1.2

The section moment of inertia may be different for the different materials, but the value is indicative that flow induced instabilities are of main concern for very compliant materials.

An additional effect of internal pipe flows is the reduction of the fundamental natural frequency of the pipe³⁸ which again is insignificant for the steel pipe but may be an important effect for the more compliant structures. For example, the reduction of the natural frequency for a pinned-pinned pipe as a function of internal flow velocity is shown in Figure 66. If compliant materials are used, the structural analysis must consider this effect.

The water hammer effect is particularly serious during shut down operations. The rate of change of momentum per unit area of the water column is ρcU where ρ is the water density, c is the speed of sound and U is the column velocity. If the cold water flow could be shut down instantaneously, the pressure developed is $2.5 \times 10^6 \text{ N/m}^2$ or about 25 atmospheres. Thus, the rate at which the water can be shut off is dictated by allowable loads on the piping and pump system. Such effects could be induced by heaving motion of the platform/CWP system (depending on seawater system design).

FIGURE 66. INFLUENCE OF FLOW VELOCITY ON FUNDAMENTAL BENDING FREQUENCY OF A PINNED-PINNED PIPE (BLEVINS 1977)

5.0 EXPERIMENTAL REQUIREMENTS

Engineering design load criteria have been established from cylinder flow data and analytical models and are considered adequate for design purposes. Additional experiments would be valuable to increase confidence in the design loads and the prediction of CWP hydroelastic response. In some instances it may be possible to remove conservatism in the design load specifications.

Although cylinder flows have been studied for many years, there still exist a large number of unknowns because of the complicated nature of the flowfield, because of the many parameters (Reynolds number, roughness, cylinder motion, shear, and geometry) which can influence the loads and because of simulation difficulties. The effects considered most uncertain roughly in order of decreasing uncertainty are:

- Ovallling effects on pressures
- End effects on loads
- Stratification effects on loads
- Shear flow effects on loads
- Hydroelastic response of representative CWPs
- Fluctuating loads at high Reynolds number

Being the most uncertain does not necessarily imply that the information is most needed. For example, pressure fluctuations are of little interest if the CWP does not respond dynamically in the shell mode. Requirements for additional engineering data in decreasing order of importance for OTEC are:

- Hydroelastic response of representative CWPs
- Shear flow effects on loads
- Ovallling effects on pressures (for highly flexible walls)
- Fluctuating loads at high Reynolds number
- Stratification effects on loads
- End effects on loads

An experiment designed to investigate the hydroelastic response of scaled CWP concepts in uniform and shear flows would be especially useful to validate existing design tools.

6.0 SUMMARY

An analytical program was carried out to prescribe current load design criteria for the OTEC CWP. Baseline models of circumferential pressure distribution, drag and lift force and load spectra were developed for uniform flows. The effects of current shear, stratification, and cylinder motion on the baseline models were estimated to provide the final design criteria summarized in Section 3.4 of this report. Hydrodynamic forces due to surface waves, internal waves and internal flows were estimated for comparison.

6.1 Conclusions

The conclusions reached during the study are:

- Sufficient cylinder flow test data are available to provide engineering design loads for a rigid OTEC cold water pipe in uniform flows. Mean circumferential pressure distributions and drag curves were prescribed from a critical review of the literature. Pressure fluctuations and lift coefficients were found to be normally distributed in time so fluctuating loads can be described with the desired degree of confidence from the RMS fluctuating components available in the literature and some previous SAI tests. Both smooth and rough cylinders were considered, although the OTEC CWP will most certainly be rough in the hydrodynamic sense leading to simple periodic loading functions correlated over the length of the CWP.
- Cylinder motion amplifies the vortex induced loads by as much as a factor of 2.6 if the oscillating frequency is near the vortex shedding frequency. Lock-on is assumed to occur within ± 30 percent of the natural shedding frequency. Correlation lengths increase with amplitude of motion if not already correlated. Although these conclusions have been developed for laterally oscillating cylinders, it is assumed that they hold for ovalling cylinders as well.

- Ocean current shear gradients from the OTEC environmental package are not expected to dramatically alter the uniform flow loads. The recommended approach is to compute force coefficients based on an average Reynolds number and apply the local dynamic pressures to calculate the loads. The loads are taken to be correlated over the length of the pipe. There is a limited amount of data on which to base these conclusions.
- Sea water density stratification should not have a significant impact on the uniform flow loads.
- Surface wave loads can be of the same order of magnitude as current loads and must be included in the CWP design.
- Internal wave currents are small compared to flow currents and thus are of possible importance only if their periods are sufficiently short to excite CWP dynamic response.
- Internal flow pressure loads appear to be small with the possible exception of flow transients due to plant motion (vertical heave) or shut down (water hammer). However, all hydrodynamic loads should be considered in the CWP design to insure internal flow induced instabilities are not important.

6.2 Recommendations

As a result of the analytical study of hydrodynamic design loads, the following recommendations are made to verify the CWP hydroelastic response to vortex excitation or to provide engineering loads data to provide increased confidence in design recommendations.

- A small scale laboratory test program should be conducted to investigate the response of alternate CWP designs to vortex induced loads. Test parameters should include velocity shear, pipe stiffness, attachment stiffness, internal flow, roughness and end conditions. Computer codes developed to aid in the OTEC CWP design should be tested against the scale model results.

- An analytic sensitivity study should be performed to determine the relative importance of various hydrodynamic load parameters. Where critical parameters are identified, more detailed investigation of the design criteria should be undertaken possibly including experimental programs. Shear flows are one possibility, another is pressure fluctuations on flexible wall cylinders.

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