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AXIAL VIBRATIONS
OF A FLUID FILLED PIPE

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Technical Note No. 1

Prepared for

National Science Foundation (ASRA Directorate)
1800 G Street
Washington, D.C. 20550

Grant No. PFR 78-15049

November 1978

50272 -101

REPORT DOCUMENTATION PAGE	1. REPORT NO. NSF/RA-780463	2.	3. Recipient's Accession No. PB293838
4. Title and Subtitle Axial Vibrations of a Fluid Filled Pipe (Technical Note No.1)			5. Report Date November 1978
			6.
7. Author(s) I. Nelson			8. Performing Organization Rept. No.
9. Performing Organization Name and Address Weidlinger Associates, Consulting Engineers 110 East 59th Street New York, New York 10022			10. Project/Task/Work Unit No.
			11. Contract(C) or Grant(G) No. (C) (G) PFR7815049
12. Sponsoring Organization Name and Address Applied Science and Research Applications (ASRA) National Science Foundation 1800 G Street, N.W. Washington, D.C. 20550			13. Type of Report & Period Covered Technical Note
			14.

15. Supplementary Notes

16. Abstract (Limit: 200 words)

Fluid-carrying pipelines are an essential part of modern civilization and may cross seismically active regions where they may be subject to damage from earthquake induced ground shaking. The pipe motion may be in the lateral or axial directions or both. Since studies have shown that axial stresses may be an order of magnitude or more than bending stresses, axial motion is given first attention. Based on the assumption that a certain fraction of the mass of the fluid participates in the pipe response during axial motion, this study is an attempt to answer the question of fluid pipe interaction when the only mechanism for coupling is the fluid viscosity. The report includes sections on problem formulation, vanishingly small frequencies, and high frequency response. It is concluded that for any practical range of parameters, the effect of an internal fluid, coupled to a pipe through viscosity only, on the longitudinal (axial) vibrations of the pipe is negligible.

17. Document Analysis a. Descriptors

Axial flow	Pipelines	Seismic waves
Fluid flow	Vibration	Equations
		Formulations

b. Identifiers/Open-Ended Terms

c. COSATI Field/Group

18. Availability Statement	19. Security Class (This Report)	21. No. of Pages
NTIS		41
	20. Security Class (This Page)	22. Price PC A03 / MF A01

(See ANSI–Z39.18) See Instructions on Reverse OPTIONAL FORM 272 (4–77) (Formerly NTIS–35) Department of Commerce

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AXIAL VIBRATIONS OF A FLUID FILLED PIPE

Introduction

Pipelines carry fluids important to modern civilization. Many cross seismically active regions where they may be subjected to damage from earthquake induced ground shaking. The pipe motion may be in the lateral or axial directions, or both. Novak and Hindy (2), for example, show that axial stresses may be an order of magnitude or more than bending stresses. Thus, axial motion should be given first attention. Nelson and Weidlinger (1) modeled a pipeline as a discrete system of rigid links connected to each other, and to the ground, by springs and dashpots. The model is a natural one for a pipe consisting of stiff segments connected by relatively soft joints. They showed that the peak relative motion across a joint may be estimated using spectral techniques, where the frequency and damping ratio may be expressed in terms of the assumed mass, spring and dashpots coefficients of the system.

Leaving aside for now the much more difficult problem of estimating the soil-pipe interaction coefficients (4), the question remains as to the effect of the enclosed fluid on the mass and damping coefficients. Many authors have studied solid-fluid interaction phenomena when the relative motion is normal to the walls of the solid. When the relation motion is parallel to the walls of the pipe, i.e., axial motion, some authors [e.g., (3)] have assumed a certain fraction of the mass of the fluid participates in the pipe response. However, little analytical or experimental work is available to estimate these effects on a more rational basis. The current note is an attempt to answer the question of fluid pipe interaction when the only mechanism for coupling is the fluid viscosity.

Problem Formulation

Consider a single segment of a long straight pipe attached via joint springs and dashpots (k_p and c_J) to adjacent links which are held fixed, Fig. 1. The fluid (which cannot leak out of the joints) is free to move in the pipe. For simplicity, the average (over-time) fluid velocity is zero. Alternatively, if the steady flow is laminar, the current dynamic solution may be superimposed upon the quasistatic fluid motion.

The center pipe segment, of mass m_p, is excited by a harmonic force $F_o e^{i\omega t}$. The equation of motion of the pipe segment (assumed rigid) is thus

$$m_p \ddot{x} + 2c_J \dot{x} + 2k_p x = - f_w + F_o e^{i\omega t} \tag{1}$$

where x is the axial displacement of the segment, and where

$$f_w = 2\pi a \int_0^{\ell} \tau_{rz}(a, z, t) dz \tag{2}$$

is the force exerted by the fluid on the pipe.

The fluid is assumed to be Newtonian, and to move only parallel to the pipe axis, so that

$$\tau_{rz} = \mu \frac{\partial v}{\partial r} \tag{3}$$

where μ and $v(r, z, t)$ are the fluid viscosity and velocity, respectively. In cylindrical coordinates, the equation of motion of a fluid particle is

$$\rho \dot{v} = \frac{\partial \sigma_z}{\partial z} + \frac{\partial \tau_{rz}}{\partial r} + \frac{\tau_{rz}}{r} \tag{4}$$

Assuming $\ell >> a$, end effects are negligible, so that near the center the longitudinal variation will be small, i.e., $\partial\sigma_z/\partial z \simeq 0$. Combining Eqs. (3) and (4), and restricting ourselves to steady state solutions $v(r, t) = \bar{v}(r)e^{i\omega t}$

$$\frac{d^2\bar{v}}{dr^2} + \frac{1}{r}\frac{d\bar{v}}{dr} - i\omega\frac{\rho}{\mu}\bar{v} = 0 \tag{5}$$

where ordinary derivatives have replaced partial derivatives since the time variation is already accounted for, and the z variation neglected.

Equation (5) is recognized as Bessel's equation of order zero with complex arguments. Since the velocity must remain finite at $r = 0$, the solution of Eq. (5) may be written as

$$\bar{v}(r) = C\,J_o\left(i^{3/2} r\sqrt{\frac{\rho\omega}{\mu}}\right) \tag{6a}$$

or

$$\bar{v}(r) = C\left[\mathrm{ber}\left(r\sqrt{\frac{\rho\omega}{\mu}}\right) + i\,\mathrm{bei}\left(r\sqrt{\frac{\rho\omega}{\mu}}\right)\right] \tag{6b}$$

In all subsequent development, form Eq. (6a) will be used.

At the pipe wall the fluid velocity and pipe velocity must be equal, i.e.,

$$\bar{v}(a) = C\,J_o\left(i^{3/2} a\sqrt{\frac{\rho\omega}{\mu}}\right) = i\omega x_o \tag{7}$$

where x_o is the amplitude of the steady state pipe displacement. The shear stress at the pipe wall may be found via Eq. (3), so that Eq. (2) leads to

$$\bar{f}_w = \frac{f_w}{x_o}e^{-i\omega t} = 2\pi a\ell\omega\sqrt{\rho\omega\mu}\;i^{1/2}\left[\frac{J_1(\zeta)}{J_o(\zeta)}\right] \tag{8}$$

where $dJ_o(\zeta)/d\zeta = -J_1(\zeta)$ has been used, and where

$$\zeta = i^{3/2} a\sqrt{\frac{\rho\omega}{\mu}} \tag{9}$$

Thus, the steady state force required to cause a steady state displacement x_o is, by Eq. (1),

$$\frac{F_o}{x_o}(\omega) = \left\{2k_p - \omega^2 m_p + \mathrm{Re}[\bar{f}_w]\right\} + i\left\{2\omega c_J + \mathrm{Im}[\bar{f}_w]\right\} \tag{10}$$

where Re and Im refer to the real and imaginary parts.

Letting

$$\bar{f}_w = - m_w \omega^2 \beta + i\omega\,\gamma \tag{11}$$

where m_w is the entire mass of the enclosed fluid, and combining Eqs.(8) and (11),

$$\beta = -\frac{\mathrm{Re}[\bar{f}_w]}{m_w \omega^2} = -\frac{2}{|\zeta|}\,\mathrm{Re}[zz] \tag{12}$$

and

$$\gamma = \frac{\mathrm{Im}[\bar{f}_w]}{\omega} = \frac{2m_w\omega}{|\zeta|}\,\mathrm{Im}[zz] \tag{13}$$

where the complex quantity

$$zz = i^{1/2}\,\frac{J_1(\zeta)}{J_o(\zeta)} \tag{14}$$

Noting Eq. (11), Eq. (10) becomes

$$\frac{F_o}{x_o}(\omega) = \left[2k_p - \omega^2(m_p + \beta m_w)\right] + i\omega[2c_J + \gamma] \tag{15}$$

Thus, physically, the quantity β is the fraction of the enclosed fluid mass which should be included in computing the mass of the pipe-fluid system, while γ is the apparent increase in damping caused by the fluid.

Often, the natural frequency of the undamped system, and the fraction of critical damping are of greater interest. For the fluid filled pipe, the former is

$$\omega_w = \sqrt{\frac{2k_p}{m_p + \beta m_w}} = \omega_p / \sqrt{1 + \beta\, m_w/m_p} \tag{16}$$

where ω_p is the natural frequency of the empty pipe.

The fraction of critical damping

$$\xi' = \frac{2c_J + \gamma}{2\omega_w(m_p + \beta m_w)} = \frac{2c_J + \gamma}{2\omega_p m_p\sqrt{1 + \beta\, m_w/m_p}} \tag{17}$$

If $\beta \frac{m_w}{m_p} << 1$, then $\omega_w \approx \omega_p$ and

$$\xi' \approx \xi_p + \xi_w \tag{18}$$

where ξ_p is the fraction of critical damping of the empty pipe, and where

$$\xi_w = \frac{\gamma}{2\omega_p m_p} = \frac{\omega\, m_w}{\omega_p m_p |\zeta|}\, \mathrm{Im}[z\dot{z}] \tag{19}$$

is the added fraction of critical damping caused by the internal viscous fluid.

Vanishingly Small Frequencies

It is instructive to consider the limiting case of vanishingly small frequencies $\omega \to 0$, or $|\zeta| \to 0$. When $|\zeta| << 1$, the Bessel functions may be approximated by the first two terms, or

$$J_o(\zeta) = 1 - \frac{\zeta^2}{4} \quad \text{and} \quad J_1(\zeta) = \frac{\zeta}{2} - \frac{\zeta^3}{16} \tag{20}$$

Thus, for $|\zeta| << 1$

$$\frac{J_1(\zeta)}{J_o(\zeta)} = \frac{\zeta}{2}\left(1 - \frac{\zeta^2}{8}\right)\left(1 - \frac{\zeta^2}{4}\right)^{-1} = \frac{\zeta}{2}\left(1 - \frac{\zeta^2}{8}\right)\left(1 + \frac{\zeta^2}{4}\right) = \frac{\zeta}{2}\left(1 + \frac{\zeta^2}{8}\right) \tag{21}$$

so that Eq. (15) becomes

$$\frac{F_o}{x_o}(\omega) = \left[2k_p - \omega^2(m_p + m_w)\right] + i\omega\left[2c_J + \frac{m_w^2\omega^2}{8\pi\mu\ell}\right] \quad , \ \omega \to 0 \tag{22}$$

Equation (22) shows that at vanishingly small frequencies, the entire mass of the enclosed fluid moves with the pipe (acts as a virtual mass), and that there is an increase in damping which increases with increasing frequency. One should not conclude from Eq. (22) that the added damping varies inversely with viscosity, since Eq. (22) was derived by assuming $|\zeta| = a\sqrt{\rho\omega/\mu} \to 0$.

High Frequency Response

The high frequency response may be obtained from the asymptotic expansions of the Bessel functions $J_o(\zeta)$ and $J_1(\zeta)$

$$J_o(\zeta) = \sqrt{\frac{2}{\pi\zeta}} \left[\cos(\zeta - \frac{\pi}{4}) + \frac{1}{8\zeta} \sin(\zeta - \frac{\pi}{4})\right] \tag{23a}$$

$$J_1(\zeta) = \sqrt{\frac{2}{\pi\zeta}} \left[\cos(\zeta - \frac{3\pi}{4}) - \frac{3}{8\zeta} \sin(\zeta - \frac{3\pi}{4})\right] \tag{23b}$$

For $|\zeta| >> 10$, the second term in each of Eqs. (23) may be dropped. In the limiting case it may be shown that

$$\lim_{|\zeta| \to \infty} \left[\frac{J_1(\zeta)}{J_o(\zeta)}\right] = + i \tag{24}$$

so that the fluid mass fraction and added damping ratio become

$$\left.\begin{aligned} \beta &= \frac{1}{a} \sqrt{\frac{2\mu}{\rho\omega}} = \frac{\sqrt{2}}{|\zeta|} << 1 \\ \xi_w &= \frac{\beta}{2} \frac{m_w}{m_p} \frac{\omega}{\omega_p} = \frac{m_w}{a\, m_p \omega_p} \sqrt{\frac{\omega\mu}{2\rho}} \end{aligned}\right\} \qquad |\zeta| = a \sqrt{\frac{\rho\omega}{\mu}} >> 10 \tag{25}$$

Results and Conclusions

Both the fluid mass fraction β and the added percent of critical damping caused by the internal fluid ξ_w are plotted versus the input frequency ω in Fig. 2, assuming $a = 1$ ft (0.3 m), $\omega_p = 100\ \text{sec}^{-1}$, and $m_w/m_p = 1.0$. The solid line corresponds to water at 70° F (21° C) [$\mu/\rho = 10^{-5}$ ft^2/ sec (9.3 x 10^{-7} m^2/sec)], while the dashed line represents crude oil at 20° F (-7° C) [$\mu/\rho = 3 \times 10^{-4}$ ft^2/ sec (2.8 x 10^{-5} m^2/sec)]. For both fluids $\beta \to 1$ as $\omega \to 0$, as was predicted by Eq. (22). Figure 2 shows that the high frequency results, Eqs. (25), apply for ω as small as 0.01 sec^{-1}, even for crude oil. At $\omega = \omega_p$, β has fallen off to 4.5×10^{-4} and 2.45×10^{-3} for water and oil, respectively.

While added damping on the order of one or more percent does occur at sufficiently high frequencies, at $\omega = \omega_p$, ξ_w is only 0.022% and 0.12% for water and oil, respectively. The larger damping at ultra high frequencies is irrelevent since the dynamic response would be essentially zero at these frequencies in any case. Of course, the actual values β and ξ_w will vary with the assumed parameters.

Nevertheless, one must conclude that for any practical range of parameters, the effect of an internal fluid, coupled to a pipe through viscosity only, on the longitudinal vibrations of the pipe is negligible.

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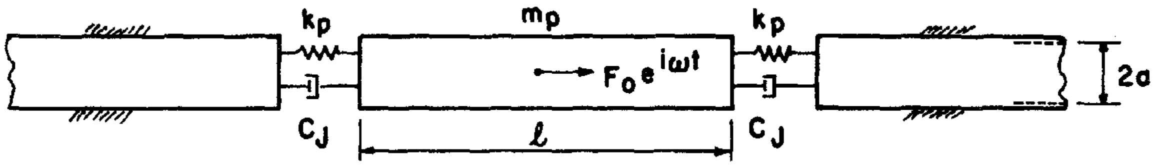

FIG. 1 GEOMETRY OF PROBLEM

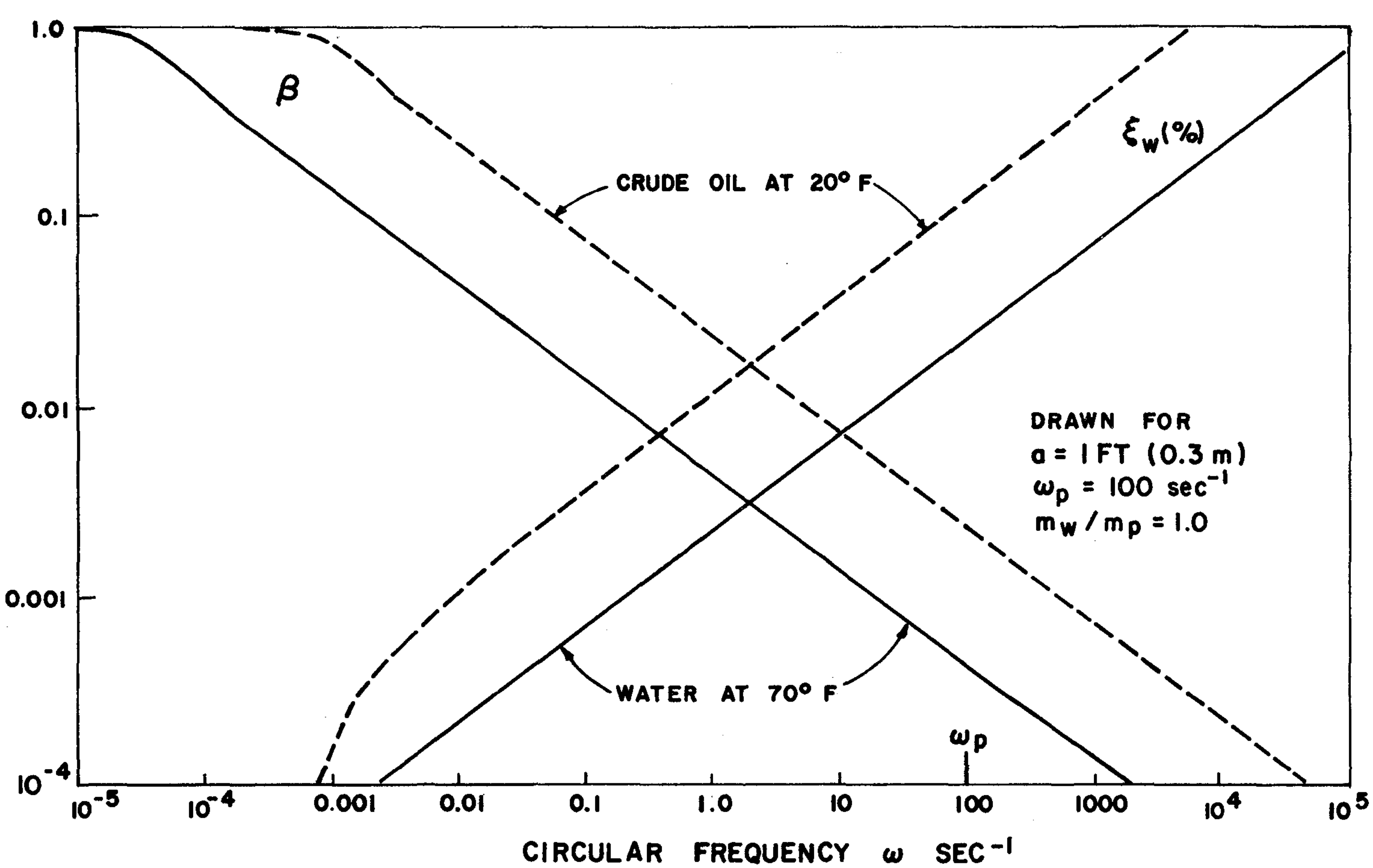

FIG. 2 FLUID MASS FRACTION β AND ADDED PERCENT OF CRITICAL DAMPING ξ_w VERSUS FREQUENCY

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