Hydrodynamically Induced Loads on Components Submerged in High-Level Waste Storage Tanks

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HYDRODYNAMICALLY INDUCED LOADS ON COMPONENTS
SUBMERGED IN HIGH-LEVEL WASTE-STORAGE TANKS

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ABSTRACT

This paper addresses the effects of added mass on components submerged in fluids. In particular, as new equipment is designed for installation in the double-shell waste-storage tanks at the Hanford Site near Richland, Washington, the equipment and the tank must be evaluated for the anticipated loads. Seismically induced loads combined with loadings from other sources must be considered during this evaluation.

A literature review shows that, for components in fluids confined to a narrow annulus or without a free surface, drastic reductions in response to seismic excitation are predicted by two-dimensional analysis. This phenomenon has been supported by testing. The reductions are explained in terms of mass coupling and buoyancy effects. For equipment submerged in fluids having a free surface and large annulus, practice suggests that it is appropriate to lump the added-mass terms with the component to address the hydrodynamic effects adequately. As in the case of a narrow annulus, this practice will reduce the natural frequency of the submerged component, but generally will increase the loads.

This paper presents the structural evaluations of submerged components using computer models that employ mock fluid elements that determine the appropriateness of considering fluid added-mass and buoyancy effects. The results indicate that if a free surface exists and the submerged component has a wide fluid annulus about it, then the added mass should be lumped with the model, and buoyancy effects are not significant. The component then may be considered to be in an air environment, and the stresses are calculated from the application of standard response spectrum procedures.

INTRODUCTION

The interaction of fluid in a containment with the equipment inserted and submerged in the fluid, under the action of seismic loads, will be addressed in this paper. Of particular interest is the treatment of hydrodynamic added mass from a 42-in. circular tube inserted at the center of a large waste-storage tank. The tanks are 75 ft in diameter and are filled with waste to an approximate height of 35 ft. The capacity of the tank is limited to 1,000,000 gal.

The added mass reduces the natural frequency of submerged structures; this phenomenon is well established in the literature. Considering a cylinder submerged in another to form a narrow fluid annulus, Fritz (1972) shows mass-coupling equations that, when applied to seismic excitation, show considerable response reduction over a simple spring-mass cylinder in air having the added mass lumped with the cylinder mass. Such effects were demonstrated with analysis and test by Scavuzzo (1979) for a box structure in a narrow fluid annulus. On the other hand, Goya (1989) shows no such effect in equations modeling the response of an intake tower submerged in a large body of water.

The equipment under discussion was treated as a cantilever beam suspended in fluid and constrained at the top. The first approach to the seismic qualification of the equipment was to lump the hydrodynamic added mass with the equipment mass. The component then was considered to be in an air environment and was analyzed with standard response spectrum procedures. This approach, however, resulted in large seismic stresses that controlled the support design for the equipment. A justifiable reduction in the seismic loads could result in considerable savings for the support structure.

This paper represents the results of analysis containing the closed form solutions as well as analyses using finite-element methodology intended to verify the contribution of added mass. The results confirm that the initial design approach was appropriate. The other contribution to the loads and stresses in

hydrodynamic analysis is the sloshing effect caused by fluid motions washing over the stationary equipment. This effect is addressed in detail in a paper by Rezvani, et al. (1993).

APPROACH

To fully address the effects of added mass and buoyancy on equipment submerged in the storage tanks, the following six cases are considered. First, a single-degree-of-freedom system is selected. The closed form solutions are considered for three different masses:

1) The equipment mass only (air environment)

2) The equipment mass and the virtual added mass coupled together

3) The equipment mass and the virtual added mass lumped together.

The second analysis used finite-element methodology. The three cases considered are listed below.

4) One-dimensional model featuring a horizontal slice of the equipment in a full tank and without a free surface.

5) Two-dimensional model featuring the equipment in a tank with a free surface and 8 ft in diameter, i.e., equipment is confined to a narrow annulus.

6) Two-dimensional model featuring the tube in a tank with a free surface and 75 ft in diameter, i.e., equipment diameter is small relative to containment diameter and provides a large annulus.

For all six cases, the input was a lateral harmonic excitation at the base of the structure, equivalent to a sine sweep test, which shakes a tank, and a tube extending down into the fluid. This area has been under investigation for a long time.

ANALYSIS

Mass-Coupling Equations

The container considered is rigid and completely filled with fluid. The container base is excited laterally with an input acceleration of $\ddot{x}_o$ (see Figure 1).

From Blevins (1979) for the case of a stationary tank,

$$m_s(\ddot{x} - \ddot{x}_o) = \sum F_x = -c\dot{x} - kx - m_v\ddot{x} + m_b\ddot{x}_o, \qquad (1)$$

where

x = Relative displacement of structural mass, m_s, with respect to the containment

m_v = Virtual or added mass

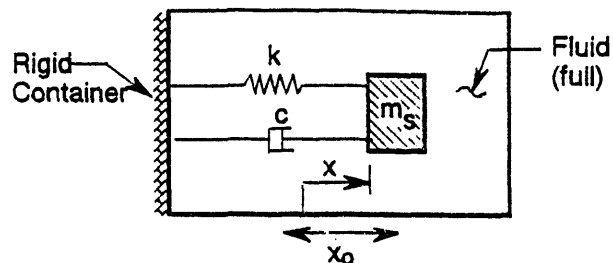m_b = Fluid mass displaced by m_s

m_s = Component mass.

Figure 1. Rigid Fluid Container with Coupled Component.

The $m_b\ddot{x}_o$ term is the buoyancy term from the bulk fluid acceleration $\ddot{x}_o$. Equation (1) is put into the form common to seismic response spectrum analysis by dividing each term by $(m_s + m_v)$ and rearranging:

$$\ddot{x} + 2\zeta\omega_n\dot{x} + \omega_n^2 x = -\psi\ddot{x}_o , \qquad (2)$$

where

$$2\zeta\omega_n = \frac{c}{m}, \quad \omega_n^2 = \frac{k}{m},$$

$$m = m_s + m_v, \quad \psi = \frac{m_s - m_b}{m}. \qquad (3)$$

In a response spectrum analysis (Clough and Penzien 1975, Chapter 27), the maximum displacement and spring force are

$$x = \psi\frac{S_a(\omega_n,\zeta)}{\omega_n^2} = \frac{m_s - m_b}{k}S_a , \qquad (4a)$$

$$f = kx = (m_s - m_b)S_a, \qquad (4b)$$

where $S_a(\omega_n,\zeta)$ is the spectral acceleration at natural frequency ω_n and damping ζ. Equation (4b) clearly shows that the added mass is not involved with the spectral acceleration loading and that the buoyancy term reduces the loading.

The fluid force terms $m_v\ddot{x}$ and $m_b\ddot{x}_o$ in Equation (1) are readily identified in Equation (14) from Fritz (1972) for the case of a cylinder in a fluid-filled annulus. Response spectrum analysis of a more general structural configuration can be carried out with the fluid coupling element STIF38 in ANSYS[1] (1989), which employs Fritz's notation.

Damping

Considering fluid viscous effects, equivalent damping for a simplified pump model has been estimated to be about 1% with drag data for cylinders. Since structural damping is substantially greater, the viscous effects are neglected.

Single-Degree-of-Freedom (SDOF) Steady Response

Steady response to base excitation is calculated for a single degree of freedom with ω as the forcing frequency (see Figure 2).
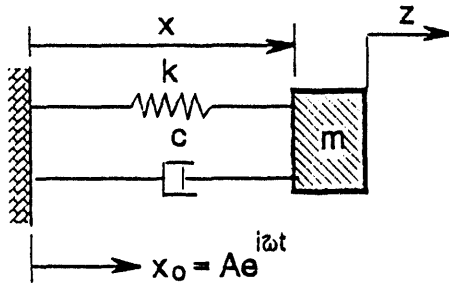

Figure 2. Single-Degree-of-Freedom Model.

Consider Equation (2) and the definitions of x_0, x, and z as given below:

$$x_o = A_o\, e^{i\omega t} \qquad x = A e^{i\omega t} \qquad z \equiv x_o = B e^{i\omega t}$$

Three mass cases are selected:

(1) Air (only structural entrapped mass is considered):
$$m = m_s$$
$$\psi = 1.$$

(2) Mass coupling (structural entrapped mass and virtual [added] mass are coupled):
$$m = m_s + m_v$$
$$\psi = \frac{m_s - m_b}{m_s + m_v}$$

(3) Lumped (structural entrapped mass and virtual [added] mass are lumped together):
$$m = m_s + m_v$$
$$\psi = 1.$$

The resulting steady responses to harmonic excitation are displayed by plotting displacements and reactions in the form of magnitudes and phases of $\dfrac{z}{x_o}$ and $\dfrac{kx}{-m_s \ddot{x}_o}$ vs $\dfrac{\omega}{\omega_a}$,

where

$$\omega_a^2 \equiv k/m_s.$$

Necessary equations for absolute displacement and base reaction in all three cases are

$$\frac{z}{x_o} = \frac{B}{A_o} = \frac{1-(1-\psi)\omega^2/\omega_n^2+2i\zeta\omega/\omega_n}{1-\omega^2/\omega_n^2+2i\zeta\omega/\omega_n} = T_\psi e^{i\theta_z}$$

and

$$\frac{kx}{-m\ddot{x}_0} = \frac{\omega^2\,A}{\omega^2\,A_o} = \frac{\psi}{1-\omega^2/\omega_n^2+2i\zeta\omega/\omega_n} = \psi R_d e^{i\theta_R}.$$

Here, T_ψ is the usual force transmissibility when $\psi = 1$, while θ_z and θ_R are the appropriate phase angles.

If we let $\omega_a^2 = k/m_s$, we must adjust ω/ω_n and the reaction equation for the three cases considered individually, as follows:

(1) Air:
$$\frac{kx}{-m_s\ddot{x}_o} = R_d e^{i\theta} \qquad \omega/\omega_n = \omega/\omega_a.$$

(2) Mass coupling:
$$\frac{kx}{-m_s\ddot{x}_o} = \frac{kx}{-(m_s-m_v)\ddot{x}_o}\frac{m_s+m_v}{m_s} = \frac{m_s-m_b}{m_s}R_d e^{i\theta}$$

$$\omega/\omega_n = \frac{\omega}{\omega_n}\frac{\omega_a}{\omega_n} = \sqrt{\frac{m_s}{m_s+m_v}}\frac{\omega}{\omega_a}$$

(3) Lumped:
$$\frac{kx}{-m_s\ddot{x}_o} = \frac{m_s-m_v}{m_s}R_d e^{i\theta} \qquad \frac{\omega}{\omega_n} = \sqrt{\frac{m_s}{m_s+m_v}}\frac{\omega}{\omega_a}.$$

The resulting normalized displacement and reactions vs. normalized frequency are plotted in the results and conclusions section below.

One-Dimension Finite-Element Modeling

The ANSYS STIF61 and STIF81 Fourier elements are used to model a one-dimensional horizontal slice of a configuration with a 40-in.-diameter probe centered in a 75-ft-diameter tank (see Figure 3).

The probe was modeled as a double cantilever with the lower end free to move in the fluid slice and was given a natural frequency of 1 Hz in air. All probe mass is concentrated at the end. Stiffness proportional damping is selected as 7% at 1 Hz.

The fluid-element mesh refinement is graduated, from a 0.3-ft radial thickness at the probe, increasing with a 1.5 factor outward. This choice was based on the assumption that the linear STIF81 elements give a multilinear fit to the fluid Stokes velocity curve given by Fritz (1972) with an error of about 2%. The STIF81 fluid nodes are coupled to STIF61 cantilever nodes in the

radial direction. Similarly, the outer fluid nodes are restrained in the radial direction. The ANSYS STIF81 input directions provide water material properties.

The cantilever reaction used is shown below (see ANSYS Section 2.26.2):

$$F_n = m_s \ddot{x}_o = m_s \omega^2 x_o \qquad \omega = 2\pi f \qquad x_0 = 1 \ .$$
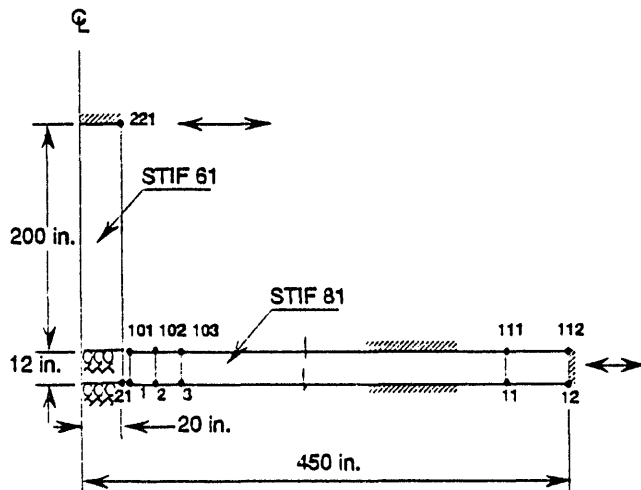

Figure 3. One-Dimensional Finite-Element Model.

The first variable in POST26 output is the frequency. It also serves as the normalized frequency, since $f_a = 1$. Similarly the displacement amplification is the cantilever node displacement. The frequency shift for added mass that should be observed in the results is

$$f = f_a \sqrt{\frac{m_s}{m_s + m_v}} = \sqrt{\frac{2}{3}} f_a = 0.816 \times 1\,\text{Hz} = 0.816\,\text{Hz} \ .$$

The resulting normalized displacement amplitudes and reactions vs. normalized frequency are plotted in the results and conclusions section below.

Two-Dimensional Narrow Tank

A 42-in.-diameter cylinder extending into an 8-ft-diameter tank is analyzed with a free surface (see Figure 4).

The structural masses and stiffnesses produced $f_a = 1.903\,\text{Hz}$. Stiffness proportional damping was set at 7% for this frequency.

Excitation frequencies are normalized to f_a with a 0.525 (1/1.903) factor for normalized displacement amplitude and reactions. The results are plotted in the results and conclusions section below. Amplification is taken as the displacement at an elevation where the probe's relative displacement is the same as a single degree of freedom oscillator with the same natural

frequency. This is where the displacement is ψ_1^{-1} times the tip displacement. Here, ψ_1 is the participation factor for the free cantilever mode with a unit tip displacement.
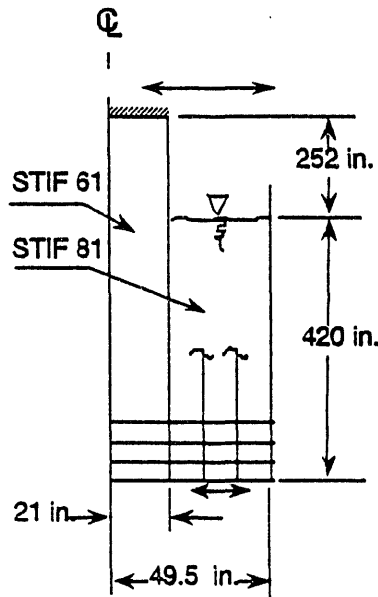

Figure 4. Two-Dimensional Finite-Element Model; The Equipment Submerged in a Narrow Open Annulus.

For base reaction moment normalization, the base moment that would be generated statically in air testing is used:

$$M_s = mh\omega^2 x_o \qquad x_o = 1 \ .$$

where mh is an equivalent mass moment found from a modal analysis of the free cantilever:

$$M = mhS_a \qquad mh = \frac{M}{S_a} \ .$$

Thus, with ANSYS notation for the Fourier element output,

$$M_n = \frac{\pi(RF_y + M_z)}{mh} \ .$$

Two-Dimensional Wide Tank

The 8-ft-diameter model is readily extended to the scale of a million-gallon tank. The outside wall is flexible, with an impulse mode of about 8.5 Hz. The slosh-mode natural frequency is 0.19 Hz. Horizontal meshing uses the 1.5 factor for element-size expansion to a maximum of a 60-in. width.
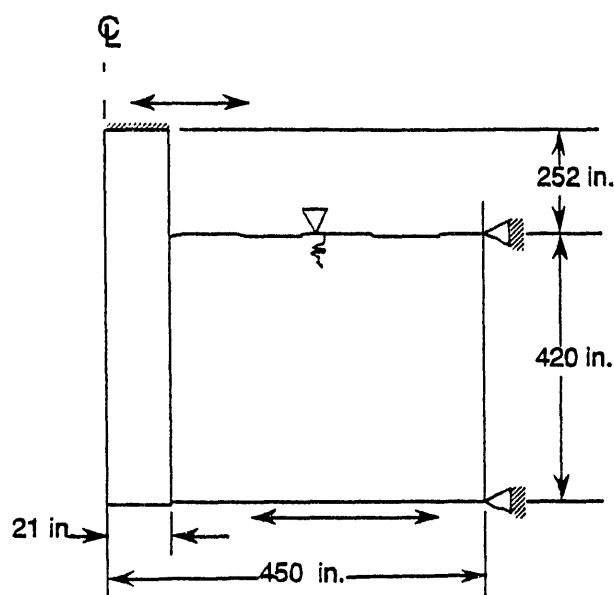
Figure 5. Two-Dimensional Finite-Element Model:
The Equipment Submerged in a Wide Open Annulus.

The added mass reduced frequency is estimated:

$$f = \sqrt{\frac{m_s}{m_s + m_a}} f_a = \sqrt{\frac{2}{3}} \, 1.903 = 1.553 \, \text{Hz} \; .$$

Displacement amplification is taken at an elevation of 168 in.

Modal analysis of the fluid-structure system was also carried out to see whether more than one mode contributed to probe bending. There was only one significant mode at 1.539 Hz. The shape is shown in Figure 6.

Some surface wave motion can be observed. The resulting normalized displacements and reactions vs. normalized frequency are plotted in the results and conclusions section below. The normalization factors are calculated with the same procedure used for the two-dimensional narrow tank.

RESULTS AND CONCLUSIONS

Six different configurations were analyzed to find the response of the systems subjected to harmonic base excitation. The responses of interest were the displacement amplitudes and the moment reactions as a function of frequency. The results, normalized, are plotted in Figures 7 and 8. Figure 7 shows the normalized displacement amplitudes and phase angles vs. forcing frequency normalized with respect to natural frequency.
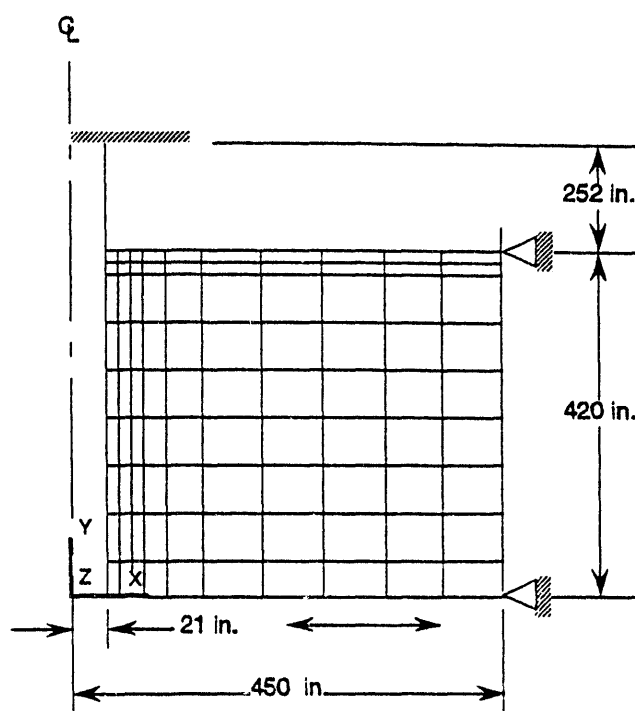

Figure 6. Surface Wave Motion for 1.539 Hz Mode,
For Dimensions and Boundary Conditions See Figure 5.

The six configurations evaluated are listed below:

1) The response for equipment in an air environment, i.e., no contribution from interaction with fluid

2) The response for an equipment mass plus the virtual added mass coupled with the equipment mass

3) The response for an equipment mass plus the virtual added mass lumped with the equipment mass

4) The response for a one-dimensional finite-element model of equipment in a full, rigid tank without free surface

5) The response for a two-dimensional finite-element model of equipment in a narrow-annulus, rigid tank with free surface

6) The response for a two-dimensional finite-element model of equipment in a tank with a wide annulus relative to the equipment diameter and with free surface.
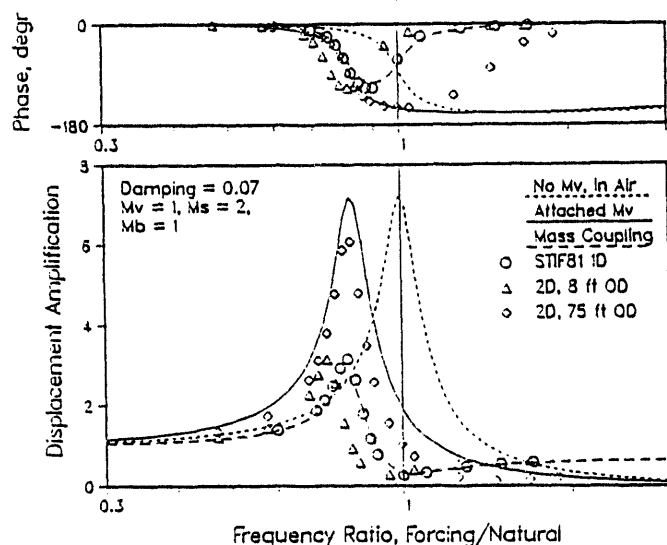
Figure 7. Normalized Displacement Amplitude vs. Normalized Forcing Frequencies.

Figure 8 shows the normalized moment reactions vs. forcing frequency normalized with respect to natural frequency.
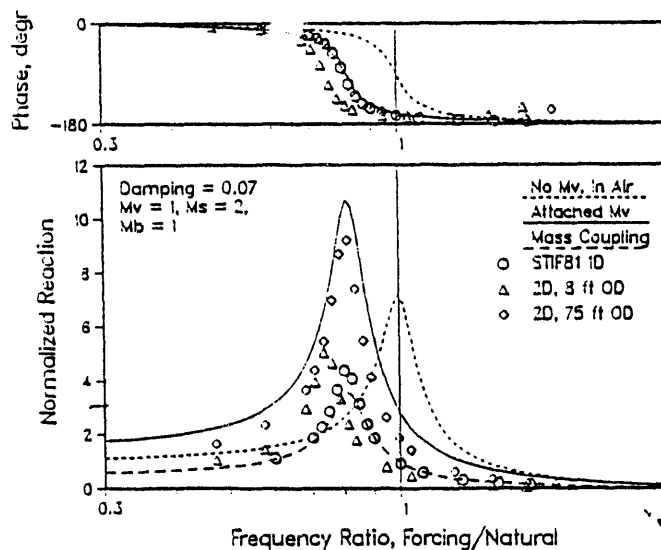

Figure 8. Normalized Moment Reaction vs. Normalized Forcing Frequencies.

Inspection of Figure 7 shows that the displacement amplitudes for Case 3 have the same values as those for Case 1. However, the added virtual mass has decreased the forcing frequency and thus shifted the curve for Case 3 to the left. Cases 3 and 6 are almost identical except that the displacement amplitudes for Case 6 are lower. Cases 3, 4, and 5 are practically identical; marked reductions in amplitude are noted. The same conclusions addressed

above could be cited for Figure 8. From Figure 8, it becomes obvious that for the case under investigation - submerged equipment in a full tank with a wide open annulus - the virtual added mass should be lumped with the equipment mass. Then the structure could be analyzed as if it were in an air environment as in the initial approach.

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