WHC-SA-2616-FP

# A Parametric Study of Double-Shell Tank Response to Internal High-Frequency Pressure Loading

Prepared for the U.S. Department of Energy
Office of Environmental Restoration and
Waste Management

Westinghouse
Hanford Company Richland, Washington

Hanford Operations and Engineering Contractor for the
U.S. Department of Energy under Contract DE-AC06-87RL10930

Conf-950740--40

WHC-SA-2616-FP

# A Parametric Study of Double-Shell Tank Response to Internal High-Frequency Pressure Loading


J. P. Strehlow
G. Abatt
Westinghouse Hanford Company

R. Baliga
K. Choi
J. S. Shulman
ADVENT Engineering Services, Inc.



Date Published
February 1995

To Be Presented at
1995 ASME-PVP Conference
Honolulu, Hawaii
July 23-27, 1995

To Be Published in
*Proceedings*

Prepared for the U.S. Department of Energy
Office of Environmental Restoration and
Waste Management

Westinghouse Hanford Company P.O Box 1970 Richland, Washington

Hanford Operations and Engineering Contractor for the
U.S. Department of Energy under Contract DE-AC06-87RL10930

## DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

# A PARAMETRIC STUDY OF DOUBLE-SHELL TANK RESPONSE TO INTERNAL HIGH-FREQUENCY PRESSURE LOADING

R. Baliga, K. Choi and J. S. Shulman
ADVENT Engineering Services, Inc.
3 Crow Canyon Court, Suite 100
San Ramon, California

J. P. Strehlow and G. Abatt
Westinghouse Hanford Company
P. O. Box 1970
Richland, Washington

## ABSTRACT

The double-shell waste tank 241SY101 (SY101) is a 3,785,400-liter tank used to store radioactive waste at the Hanford Site near Richland, Washington. The tank waste has formed two layers of sludge in the tank; a convective and a nonconvective layer. Ongoing reactions in the waste cause a buildup of hydrogen molecules that become trapped within the nonconvective layer of the waste. Various means of preventing the buildup of hydrogen molecules in the nonconvective layer have been investigated, including the use of a sonic probe that would transmit high-frequency acoustic pressure waves into the nonconvective layer of the waste. During the operation of the sonic probe, the pressure waves transmitted from the probe induce pressure time history loading on the inside surface of the primary tank.

For low-frequency fluid-structure interaction loads, such as those associated with seismic events, the convective and impulsive effects of the waste-filled tank are well documented. However, for high-frequency loading, such as that associated with acoustic pressure waves, interactions between the waste and the primary tank are not understood. The pressure time history is represented by a harmonic function with a frequency range between 30 and 100 Hz. Structural analyses of the double-shell tank have been performed that address the tank's response to the sonic probe acoustic pressure loads.

This paper addresses the variations in the tank response as a function of percent waste mass considered to be effective in the dynamic excitation of the tank. It also compares results predicted by analyses that discretely model the liquid waste and presents recommendations for the simplified effective mass approach. Also considered in the parametric study is the effect of damping on the tank response for the same pressure loading.

## INTRODUCTION

A sonic probe, eccentrically located in the SY101 tank, would transmit continuous acoustic pressure waves into the waste medium (nonconvective layer) during its operation. These waves would induce pressure time history characterized by steady-state harmonic time dependence. The primary tank liner in contact with the nonconvective waste medium, would be subjected to this time history load.

The purpose of this analysis was to assess the effects of this harmonic pressure loading on the various tank components. The structural analysis of the SY101 tank subjected to the sonic probe loading included: (1) developing a detailed three-dimensional finite-element model of the tank with different percentages of liquid waste mass lumped on the primary liner of the tank, and (2) analyzing the tank for the sonic probe pressure loading. To account for the effects of acoustic rebounding, the sonic probe pressure wave responses were factored by 2.0. This assumed that the wave boundary was perfectly rigid and the reflected wave was in phase with the incident wave. To address the uncertainty in the structural frequencies calculated, a $\pm 15.0\%$ tolerance was applied to the harmonic forcing frequencies of 30, 70, and 100 Hz. The pressure loading at the various locations on the primary tank liner was conservatively considered in phase. Three different modal analyses associated with the 50, 25, and 0% liquid mass lumped with the primary tank mass were performed and compared.

## MODEL DEVELOPMENT

The structural analysis of the SY101 tank included developing a 3-D finite-element model of the tank using the computer program ANSYS[1]. Because of the ANSYS program limitations on the model size, a 3-D symmetric (180°) tank model was developed instead of modeling a full 360°. The finite-element model included the primary tank, concrete dome, wall, and base slab. The primary tank liner, dome, wall, and base slab were modeled with quadrilateral shell elements. The primary tank liner was connected at the base to the concrete base slab by rigid elements, and the tank wall was connected directly to the base slab. The base slab was constrained in the global x, y, and z directions, and the tank wall was restrained in the radial direction.

It is not anticipated that the concrete dome, wall and base slab have significant dynamic response to the sonic pressure loading. Therefore, the concrete section of the tank modeled with a 0.0 density. A 512 master degrees of freedom selected to characterize the overall dynamic behavior of the primary tank liner only. A schematic of the SY101 tank and the ANSYS model of the tank are provided in Figures 1 and 2.

## SONIC PROBE PRESSURE LOAD

The steady-state rotary motion of the sonic probe generated acoustic pressure waves through the nonconvective layer of the waste media (up to 200 in. from the bottom of the primary tank liner), which were transmitted to the primary tank liner in the form of a surface pressure time history. The pressure time history is represented by the harmonic forcing function: $F(t) = F_o(\omega,r)\cos(\omega t)$ where $F_o(\omega,r)$ and $\omega$ represent the amplitude and frequency, respectively, of the forcing function $F(t)$. The force amplitude, $F_o(\omega,r)$, varies depending on the frequency of excitation, $\omega$, and distance of the equipment from the sonic probe. From the previous study, three sets of pressure profiles are considered with the following parameters that give pressure intensity values at a given distance from the probe:

- 2.0 mm bubbles and viscosity of 100 cP
- 2.0 mm bubbles and viscosity of 10,000 cP
- 0.2 mm bubbles and viscosity of 10,000 cP.

Each set had three pressure profiles corresponding to sonic probe excitation frequencies of 30, 70, and 100 Hz. For the tank, the 30, 70, and 100-Hz pressure profiles corresponding to the 0.2 mm bubbles and the viscosity of 10,000 cP were found to be governing and were used to calculate the pressure intensity.

The distance between the sonic probe and the primary tank liner surface varied depending on the location of the tank liner. Figure 3 shows the acoustic pressure profiles used in the evaluation of the primary tank liner. Table 1 gives the tabular values at control points (0 to 20 m from the sonic probe).

## STRUCTURAL ANALYSIS

The SY101 tank was analyzed for the pressure loads using the mode superposition technique to perform the steady-state harmonic analysis. Separate ANSYS analyses (i.e., displacement pass, POST26 analysis, stress pass) were performed for each of the three frequencies of excitation. Following the three separate ANSYS analyses, POST1 postprocessing was performed to obtain the enveloped stresses in the various tank components for the various load cases. Modal analysis was performed to calculate the natural frequencies and mode shapes of the tank. A total of 512 natural frequencies were calculated for the entire model. Modal analysis was performed for three different models (representing 50, 25, and 0% fluid masses). After the modal analysis, a displacement pass was performed to calculate the displacements at the master degrees of freedom. Constant modal damping values of 2 and 5% were used in the analysis. Although the sonic probe excitation

[1]ANSYS is a registered trademark of the Swanson Computer Systems, Houston, Pennsylvania.

frequencies were exactly defined at 30, 70, and 100 Hz, the structural frequencies of the tank may vary because of the uncertainties in modeling and stiffness characteristics of the tank. It was, therefore, decided that a range of excitation frequencies (±15% around the specified probe excitation frequencies) to accommodate these uncertainties should be considered. All 512 modes were considered in obtaining the cumulative response for each excitation frequency range (i.e., 25.5 to 34.5, 59.5 to 80.5, and 85 to 115 Hz).

A POST26 analysis was performed for each excitation frequency range on completion of the displacement pass. Displacement amplitudes and phase angles were obtained at all nodes that were declared master degrees of freedom. The POST26 analysis created output of the displacement amplitude and phase angle as a function of the excitation frequency. The excitation frequencies corresponding to the peak displacement amplitudes at the various locations on the primary tank liner were identified, and stress passes were performed at these excitation frequencies. The plots of displacement amplitudes against the forcing frequencies are shown in Figures 3 through 12. The stress pass expand the reduced displacement solution (obtained from the displacement pass) to the full-degree-of-freedom set. With the expanded solution, strains, stresses, nodal forces, and reaction forces were calculated. The stress pass were performed at the governing frequencies identified from the POST26 output. The solutions are computed at phase angles 0 and 90°. The ANSYS general postprocessor, POST1, was used to obtain the response of the various tank components. Such responses included: stresses in the primary liner, concrete dome, wall, and base; and displacements at critical locations. The (real and imaginary) responses computed in the stress pass at all governing frequencies, within each excitation frequency range, were combined by the SRSS (square root sum of the squares) method to determine response amplitude were these governing frequencies. The computed response amplitudes are compared and the maximum responses used in the structural evaluation.

The maximum element stresses in the primary liner at the various governing frequencies are tabulated in Tables 2 through 4. Table 5 shows the comparison of the stresses in the primary liner resulting from the sonic probe pressure loading at 70 Hz between 2% damping and 5% damping for the 0% added mass case. Large differences in the nodal and element stresses in the primary liner were noticed. These differences were attributed to the element sizes in the convective region of the primary liner. However, the primary liner was evaluated using the maximum nodal stresses in each tank component (e.g., 1/2 in., 3/8 in. plates, etc.). Table 6 shows the maximum nodal stresses in the concrete section of the SY101 tank. Tables 2 through 4 indicate that 79 or 80 Hz is the governing frequency for the case with the 0% added mass, while 25.5 Hz is the governing frequency for the case with 25% or more added mass. The added mass can lower the natural frequencies of the critical modes and affect the stress responses significantly. Table 5 shows that the ratios of the maximum stresses from the 2 and 5% damping cases at the governing frequency are around 2.0. This was close to the expected amplification (2.5) between the 2 damping values for a single degree of freedom system.

## ANALYSIS RESULTS

Structural analysis of the SY101 double-shell tank was performed to address the response of the tank resulting from the sonic probe acoustic pressure loads. For low-frequency fluid-structure interaction loads, such as those associated with seismic events, the convective and impulsive effects of the waste-filled tank are well documented. However, for high-frequency loading, such as that associated with acoustic pressure waves, interactions between the waste and the primary tank are not well understood. The pressure time history was represented by a harmonic function with a frequency range between 30 and 100 Hz. Therefore, the analysis included the various load cases associated with the forcing frequencies, the added mass, and the damping. This was necessary to address the variations in the tank response as a function of percent waste mass considered effective in the dynamic excitation of the tank. As part of the parametric study, 3 added mass cases are considered in the analysis (50, 25, and 0% added mass). All three analyses considered 2% structural damping. Additionally, two analyses were performed for 70 and 100 Hz pressure loading with 0% added mass and 5% structural damping. The results of the analyses showed the governing load case was the one with a pressure loading having 70 Hz exciting frequency, 0% added mass, and 2% structural damping.

To demonstrate the effect of the structural damping on the structural response of the tank, the maximum membrane and membrane + bending stresses resulting from the 70-Hz pressure loading are plotted for 2 and 5% damping in Figures 13 through 18. These figures indicate that the stresses are reduced by a factor of 2.0 when the structural damping is increased from 2 to 5%. To further demonstrate the effect of the added mass on the structural response of the tank, the maximum principal stresses resulting from the three different frequency pressure loadings (i.e., 30, 70, 100 Hz) as a function of added mass are plotted in Figures 19 through 21. The first mode frequency of the tank with the 0% added mass was 6.23 Hz. When 50% waste mass was added to the tank liner, the first mode frequency was shifted from 6.23 to 0.89 Hz. When 25% waste mass was added to the tank liner, the first mode frequency shifted from 6.23 to 1.25 Hz. With 0% added mass, the dominant frequency of the tank was approximately equal to 79 Hz. With 25% added mass, the dominant frequency of the tank was approximately equal to 25.5 Hz. For a 30-Hz pressure loading case, 25% added mass was dominating more than the 0% added mass. For a 30-Hz pressure loading case, the maximum stress amplification between the 25% added mass and the 0% added mass was approximately 4.5. For the same loading case, the maximum stress amplification between the 0% added mass and the 50% added mass was approximately 2.0. For 70 Hz pressure loading case, 0% added mass was dominating more than the 25% added mass. For 70 and 100 Hz pressure loading cases, the maximum stress amplification between the 0% added mass and the 50% added mass was more than 100. For the same loading cases, the maximum stress amplification between the 50% added mass and the 25% added mass was negligible. Even though the 25% added mass case was governing at 30-Hz frequency, the maximum stress in this case was only 1,000 $lbf/in^2$, which was 1/10 of the maximum stress with 0% added mass and 70 Hz frequency loading.

## CONCLUSIONS

The following conclusions can be drawn from the structural analysis of the SY101 for the sonic probe pressure loading.

- The stresses in the primary tank are within the code allowable limits when the tank is subjected to the sonic pressure loading at a 30 Hz frequency.

- The stresses in the primary tank are within the code allowable limits when the tank is subjected to the sonic pressure loading at 70 and 100 Hz frequencies, and the percent added waste mass effective with the tank liner is in the range of 25 to 50% of the total waste mass in the tank.

- The stresses in the concrete section of the tank (i.e., dome, wall, and base slab) are negligible for any pressure loading between 30 and 100 Hz. The percent added waste mass and the percent structural damping does not have any effect on the structural response of the concrete containment of the tank. Therefore, in a double-shell tank such as SY101, the concrete section of the tank need not be included in the finite-element model to evaluate the hydrodynamic effect on the primary tank liner.

With the present analyses, it is difficult to justify the actual percent of added mass as effective with the tank liner in the dynamic excitation of the tank. The stresses reported in this paper for the 0% added mass case exceed the code allowable limits. However, it is unreasonable to assume no waste mass is excited with the primary tank when the sonic pressure loading is applied. Therefore, it is concluded that more analysis is required to study the actual behavior of the primary tank with the liquid waste when the tank liner is subjected to the high-frequency sonic probe pressure loading. More analyses will be performed to demonstrate the liquid mass effect on the tank.

**TABLE 1. SONIC PROBE PRESSURE AMPLITUDE AS A FUNCTION OF DISTANCE.**

| Distance (Meters) | Pressure (Pascal) | | |
|---|---|---|---|
| | 30 Hz | 70 Hz | 100 Hz |
| 0.5 | 4,552 | 31,446 | 64,165 |
| 1 | 2,900 | 20,555 | 40,320 |
| 2 | 1,941 | 13,034 | 23,118 |
| 3 | 1,544 | 9,608 | 15,358 |
| 4 | 1,308 | 7,519 | 10,827 |
| 5 | 1,147 | 6,080 | 7,885 |
| 6 | 1,027 | 5,018 | 5,861 |
| 7 | 933 | 4,200 | 4,418 |
| 8 | 857 | 3,553 | 3,365 |
| 9 | 793 | 3,029 | 2,583 |
| 10 | 738 | 2,598 | 1,996 |
| 12 | 649 | 1,939 | 1,208 |
| 14 | 579 | 1,468 | 741 |
| 16 | 522 | 1,123 | 460 |
| 18 | 474 | 865 | 287 |
| 20 | 434 | 671 | 181 |

TABLE 2. MAXIMUM ELEMENT STRESSES IN THE 241SY101 TANK PRIMARY LINER WITH 50% ADDED MASS AND 2% DAMPING (ALL STRESSES ARE IN $LBF/IN^2$).

| Item | Freq Range (Hz) | SXT | SYT | SHT | SXM | SYM | SHM | SXB | SYB | SHB |
|---|---|---|---|---|---|---|---|---|---|---|
| 30 Hz | 25.5 | 126 | 125 | 15 | 8 | 89 | 15 | 137 | 75 | 15 |
| | 27.0 | 87 | 85 | 9 | 5 | 60 | 9 | 95 | 54 | 10 |
| | 30.0 | 51 | 48 | 5 | 3 | 34 | 6 | 56 | 32 | 6 |
| | 31.5 | 43 | 40 | 8 | 4 | 29 | 8 | 46 | 27 | 8 |
| | 32.0 | 44 | 42 | 8 | 3 | 31 | 7 | 47 | 29 | 7 |
| 70 Hz | 60.0 | 57 | 58 | 22 | 5 | 45 | 22 | 62 | 43 | 22 |
| | 61.0 | 53 | 51 | 18 | 4 | 44 | 18 | 57 | 41 | 18 |
| | 66.0 | 43 | 42 | 22 | 5 | 35 | 21 | 47 | 33 | 21 |
| | 67.0 | 43 | 45 | 21 | 5 | 35 | 21 | 47 | 33 | 21 |
| | 68.0 | 43 | 45 | 19 | 5 | 35 | 19 | 46 | 33 | 19 |
| | 70.0 | 39 | 41 | 13 | 3 | 32 | 13 | 43 | 31 | 13 |
| | 80.0 | 28 | 28 | 24 | 4 | 24 | 24 | 30 | 22 | 24 |
| 100 Hz | 86.0 | 35 | 37 | 15 | 3 | 28 | 15 | 37 | 27 | 15 |
| | 87.0 | 33 | 34 | 14 | 3 | 28 | 14 | 35 | 27 | 14 |
| | 88.0 | 33 | 35 | 14 | 4 | 28 | 14 | 35 | 27 | 14 |
| | 89.0 | 35 | 39 | 13 | 4 | 29 | 13 | 35 | 26 | 13 |
| | 90.0 | 35 | 39 | 13 | 4 | 29 | 13 | 35 | 25 | 13 |
| | 100. | 25 | 26 | 11 | 2 | 20 | 11 | 26 | 19 | 11 |
| | 103. | 24 | 26 | 15 | 3 | 20 | 15 | 25 | 18 | 14 |

TABLE 3. MAXIMUM ELEMENT STRESSES IN THE 241SY101 TANK PRIMARY LINER WITH 25% ADDED MASS AND 2% DAMPING (ALL STRESSES ARE IN LBF/IN$^2$).

| Item | Freq Range (Hz) | SXT | SYT | SHT | SXM | SYM | SHM | SXB | SYB | SHB |
|---|---|---|---|---|---|---|---|---|---|---|
| 30 Hz | 25.5 | 1069 | 943 | 39 | 68 | 625 | 37 | 1049 | 394 | 37 |
| | 26.0 | 975 | 874 | 33 | 64 | 584 | 32 | 970 | 370 | 33 |
| | 30.0 | 328 | 319 | 66 | 20 | 221 | 66 | 333 | 158 | 67 |
| | 30.5 | 386 | 384 | 84 | 25 | 268 | 84 | 392 | 190 | 85 |
| | 31.0 | 386 | 382 | 75 | 22 | 267 | 75 | 396 | 195 | 76 |
| | 33.0 | 221 | 223 | 40 | 14 | 159 | 41 | 232 | 128 | 41 |
| 70 Hz | 60.0 | 109 | 100 | 34 | 11 | 84 | 33 | 118 | 74 | 32 |
| | 61.0 | 99 | 96 | 39 | 14 | 81 | 38 | 107 | 67 | 37 |
| | 62.0 | 100 | 98 | 49 | 20 | 78 | 49 | 109 | 73 | 49 |
| | 63.0 | 118 | 124 | 49 | 20 | 93 | 49 | 122 | 87 | 48 |
| | 65.0 | 99 | 101 | 31 | 11 | 82 | 31 | 108 | 76 | 31 |
| | 66.0 | 94 | 91 | 30 | 9 | 77 | 30 | 102 | 70 | 29 |
| | 70.0 | 78 | 80 | 50 | 13 | 63 | 50 | 85 | 60 | 50 |
| | 79.0 | 56 | 54 | 49 | 8 | 46 | 49 | 61 | 43 | 48 |
| | 80.0 | 56 | 55 | 47 | 9 | 49 | 46 | 61 | 46 | 46 |
| 100 Hz | 86.0 | 64 | 63 | 27 | 7 | 58 | 27 | 70 | 55 | 27 |
| | 91.0 | 57 | 59 | 37 | 9 | 51 | 37 | 62 | 48 | 37 |
| | 92.0 | 56 | 59 | 44 | 10 | 49 | 43 | 61 | 47 | 43 |
| | 93.0 | 61 | 67 | 45 | 11 | 50 | 45 | 63 | 47 | 44 |
| | 94.0 | 63 | 69 | 40 | 10 | 52 | 40 | 63 | 48 | 40 |
| | 100. | 53 | 56 | 18 | 4 | 43 | 18 | 54 | 41 | 17 |
| | 111. | 41 | 45 | 38 | 7 | 34 | 38 | 42 | 32 | 38 |

TABLE 4. MAXIMUM ELEMENT STRESSES IN THE 241SY101 TANK PRIMARY LINER WITH 0% ADDED MASS AND 2% DAMPING (ALL STRESSES ARE IN LBF/IN$^2$).

| Item | Freq Range (Hz) | SXT | SYT | SHT | SXM | SYM | SHM | SXB | SYB | SHB |
|---|---|---|---|---|---|---|---|---|---|---|
| 30 Hz | 25.5 | 143 | 228 | 23 | 20 | 208 | 24 | 151 | 187 | 24 |
| | 30.0 | 150 | 257 | 51 | 42 | 235 | 52 | 190 | 212 | 52 |
| | 31.0 | 152 | 262 | 92 | 75 | 239 | 93 | 200 | 216 | 94 |
| | 31.5 | 154 | 237 | 89 | 70 | 216 | 90 | 159 | 194 | 91 |
| | 32.0 | 157 | 231 | 67 | 52 | 210 | 68 | 142 | 188 | 69 |
| | 35.0 | 166 | 265 | 43 | 28 | 241 | 43 | 185 | 217 | 44 |
| 70 Hz | 62.0 | 1958 | 4585 | 1514 | 351 | 4148 | 1496 | 2013 | 3712 | 1498 |
| | 63.0 | 2452 | 4762 | 1382 | 286 | 4302 | 1356 | 2566 | 3843 | 1362 |
| | 70.0 | 3002 | 7143 | 287 | 162 | 6281 | 284 | 3060 | 5839 | 285 |
| | 72.0 | 4040 | 9809 | 671 | 322 | 8749 | 660 | 4212 | 8087 | 671 |
| | 79.0 | 8418 | 9931 | 802 | 235 | 8334 | 794 | 8551 | 7557 | 786 |
| | 80.0 | 8178 | 9819 | 761 | 251 | 8512 | 748 | 8354 | 7660 | 740 |
| 100 Hz | 86.0 | 2772 | 2662 | 546 | 169 | 2281 | 543 | 3007 | 2094 | 540 |
| | 98.0 | 6097 | 5657 | 2392 | 366 | 4005 | 2394 | 6608 | 3537 | 2397 |
| | 99.0 | 6060 | 5670 | 2338 | 373 | 4031 | 2341 | 6567 | 3894 | 2345 |
| | 100. | 4961 | 4688 | 1892 | 325 | 3669 | 1897 | 5376 | 3515 | 1901 |

TABLE 5. COMPARISON OF MAXIMUM ELEMENT STRESS IN THE 241SY101 TANK PRIMARY LINER BETWEEN 2% DAMPING AND 5% DAMPING AND WITH 0% ADDED MASS (ALL STRESSES ARE IN LBF/IN$^2$).

| Item | Freq Range (Hz) | SXT | SYT | SHT | SXM | SYM | SHM | SXB | SYB | SHB |
|---|---|---|---|---|---|---|---|---|---|---|
| 70 Hz | 62.0 | 1492 | 2742 | 741 | 175 | 2478 | 738 | 1562 | 2217 | 737 |
| 5% Damping | 63.0 | 1518 | 2840 | 714 | 161 | 2562 | 711 | 1578 | 2299 | 711 |
| | 64.0 | 1564 | 2971 | 645 | 142 | 2675 | 645 | 1612 | 2408 | 644 |
| | 67.0 | 1893 | 3799 | 480 | 100 | 3385 | 478 | 1943 | 3080 | 476 |
| | 70.0 | 2451 | 5145 | 418 | 139 | 4544 | 415 | 2552 | 4174 | 412 |
| | 72.0 | 2842 | 5727 | 494 | 182 | 5037 | 490 | 2979 | 4635 | 487 |
| | 75.0 | 3275 | 5028 | 567 | 192 | 4372 | 562 | 3304 | 4003 | 557 |
| | 79.0 | 4065 | 4836 | 577 | 169 | 4217 | 571 | 4152 | 3729 | 565 |
| | 80.0 | 3861 | 4730 | 571 | 163 | 4212 | 564 | 3959 | 3719 | 557 |
| 70 Hz | 62.0 | 1958 | 4585 | 1514 | 351 | 4148 | 1496 | 2013 | 3712 | 1498 |
| 2% Damping | 63.0 | 2452 | 4762 | 1382 | 286 | 4302 | 1356 | 2566 | 3843 | 1362 |
| | 64.0 | 2154 | 3640 | 900 | 183 | 3283 | 877 | 2302 | 2926 | 883 |
| | 67.0 | 1957 | 4110 | 569 | 119 | 3643 | 566 | 1969 | 3340 | 563 |
| | 70.0 | 3002 | 7143 | 287 | 162 | 6281 | 284 | 3060 | 5839 | 285 |
| | 72.0 | 4040 | 9809 | 671 | 322 | 8749 | 660 | 4212 | 8087 | 671 |
| | 75.0 | 4618 | 6441 | 755 | 273 | 5525 | 750 | 4572 | 5169 | 750 |
| | 79.0 | 8418 | 9931 | 802 | 235 | 8334 | 794 | 8551 | 7557 | 786 |
| | 80.0 | 8178 | 9819 | 761 | 251 | 8512 | 748 | 8354 | 7660 | 740 |

TABLE 6. MAXIMUM NODAL STRESSES IN THE 241SY101 TANK CONCRETE SECTION WITH 0% ADDED MASS AND 2% DAMPING (ALL STRESSES ARE IN LBF/IN$^2$).

| Item | Freq Range (Hz) | SXT | SYT | SHT | SXM | SYM | SHM | SXB | SYB | SHB |
|---|---|---|---|---|---|---|---|---|---|---|
| 70 Hz | 79.0 | 27 | 11 | 14 | 22 | 8 | 18 | 18 | 11 | 24 |
| | 80.0 | 27 | 10 | 13 | 22 | 8 | 17 | 18 | 11 | 22 |

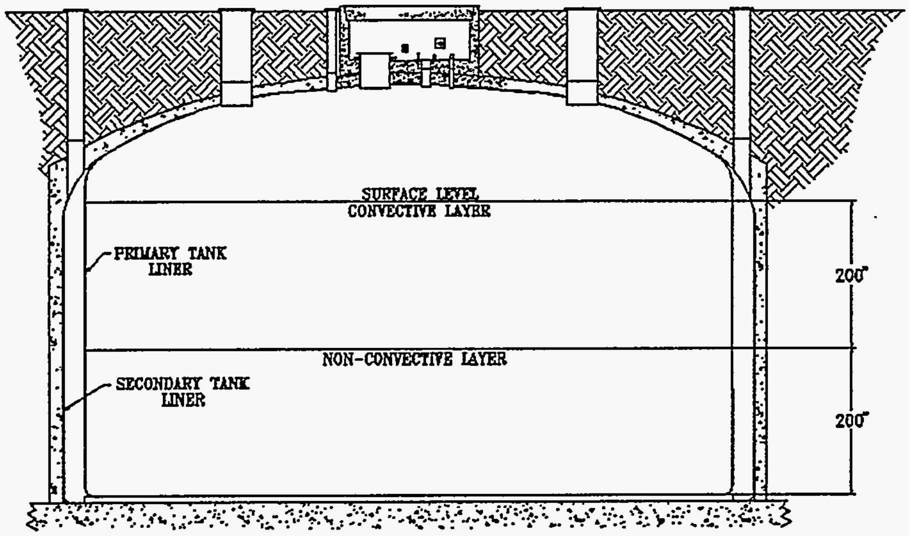


FIGURE: 1. A SCHEMATIC OF THE 241SY101 TANK.

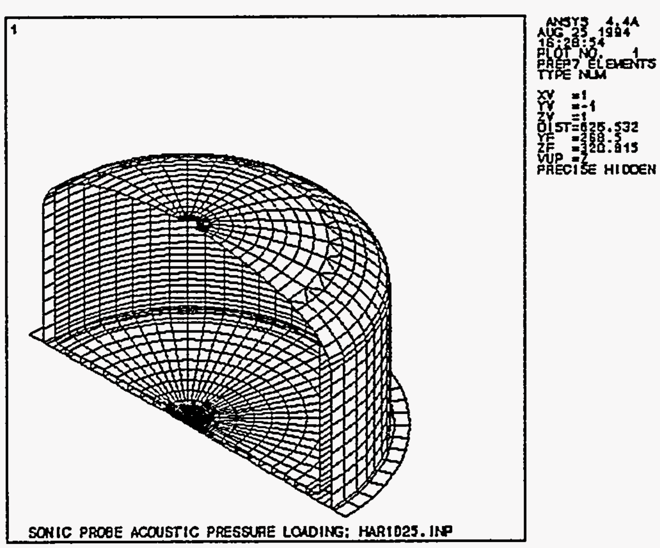


FIGURE 2. 3-D FINITE-ELEMENT MODEL OF THE 241SY101 TANK.

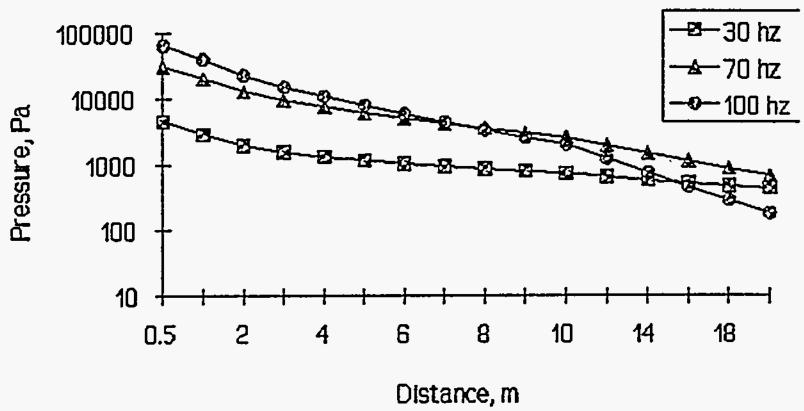


FIGURE 3. ACOUSTIC PRESSURE PROFILES.

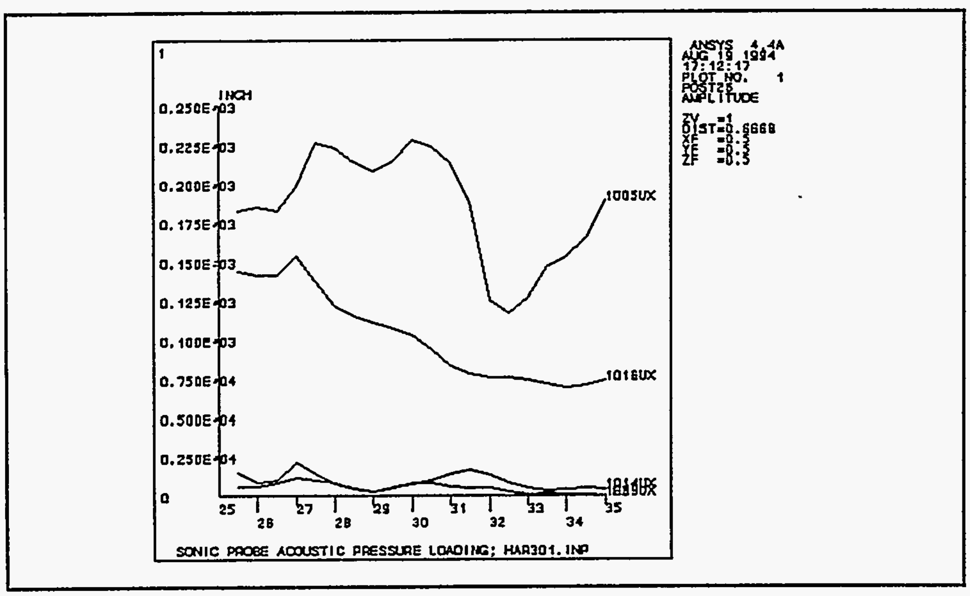


FIGURE 4. DISPLACEMENT AMPLITUDE RESPONSE (UX)
(FORCING FREQUENCY RANGE: 25 Hz - 34.5 Hz; 50% ADDED MASS).

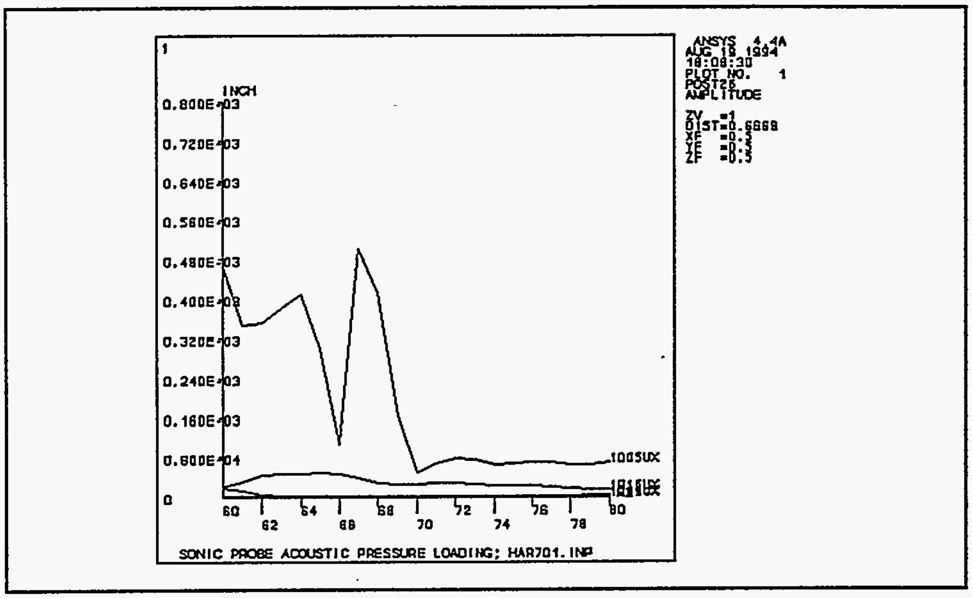


FIGURE 5.. DISPLACEMENT AMPLITUDE RESPONSE (UX)
(FORCING FREQUENCY RANGE: 59.5 Hz - 80.5 Hz; 50% ADDED MASS).

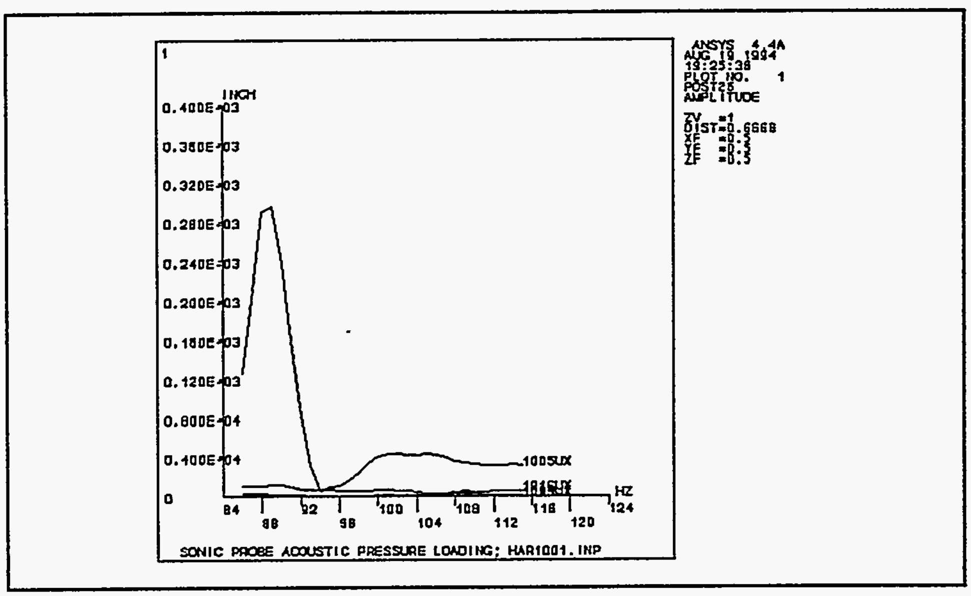


FIGURE 6. DISPLACEMENT AMPLITUDE RESPONSE (UX)
(FORCING FREQUENCY RANGE: 85 Hz - 115 Hz; 50% ADDED MASS.

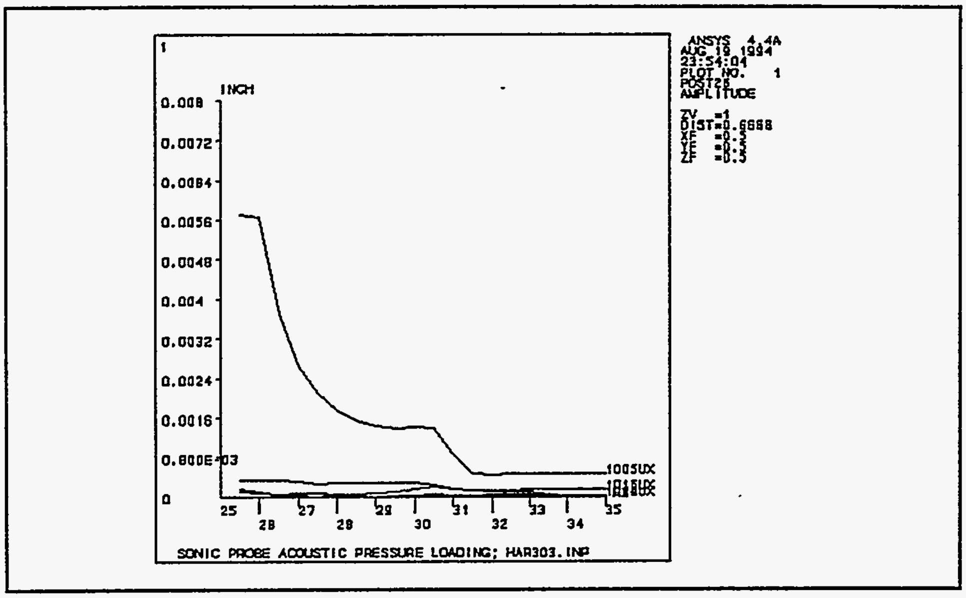


FIGURE 7. DISPLACEMENT AMPLITUDE RESPONSE (UX)
(FORCING FREQUENCY RANGE: 25.5 Hz - 34.5 Hz; 25% ADDED MASS).

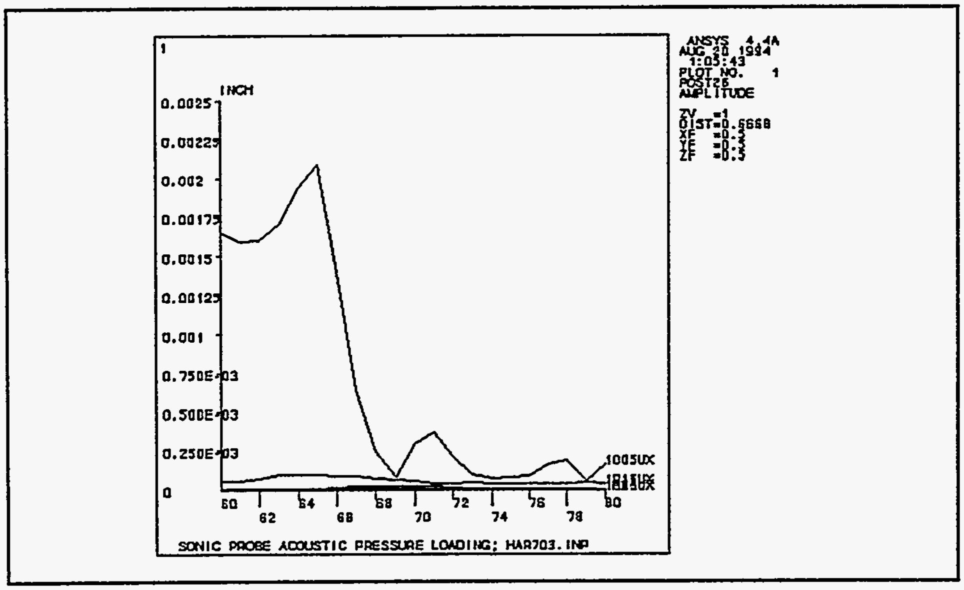


FIGURE 8. DISPLACEMENT AMPLITUDE RESPONSE (UX)
(FORCING FREQUENCY RANGE: 59.5 Hz - 80.5 Hz; 25% ADDED MASS).

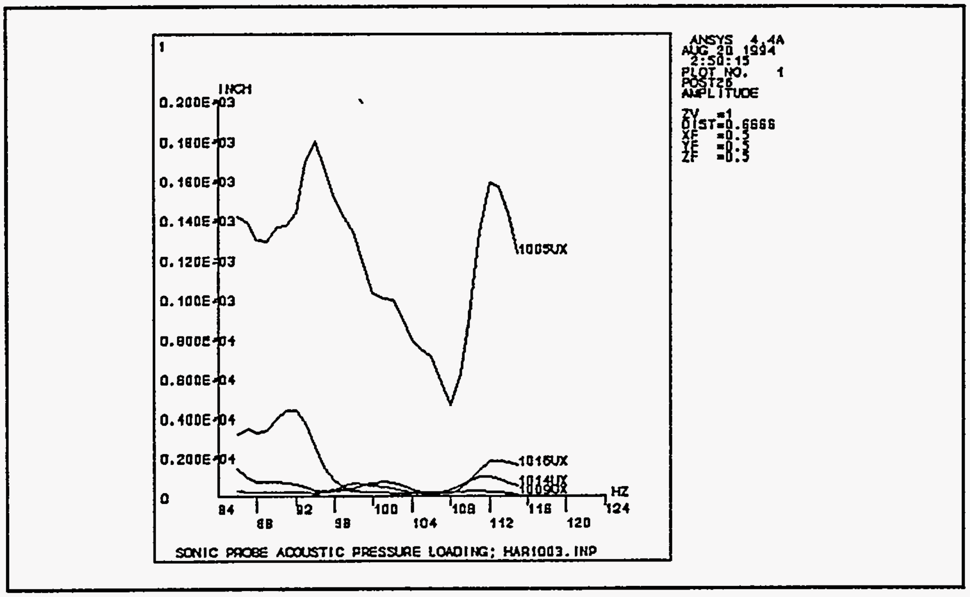


FIGURE 9. DISPLACEMENT AMPLITUDE RESPONSE (UX)
(FORCING FREQUENCY RANGE: 85 Hz - 115 Hz; 25% ADDED MASS).

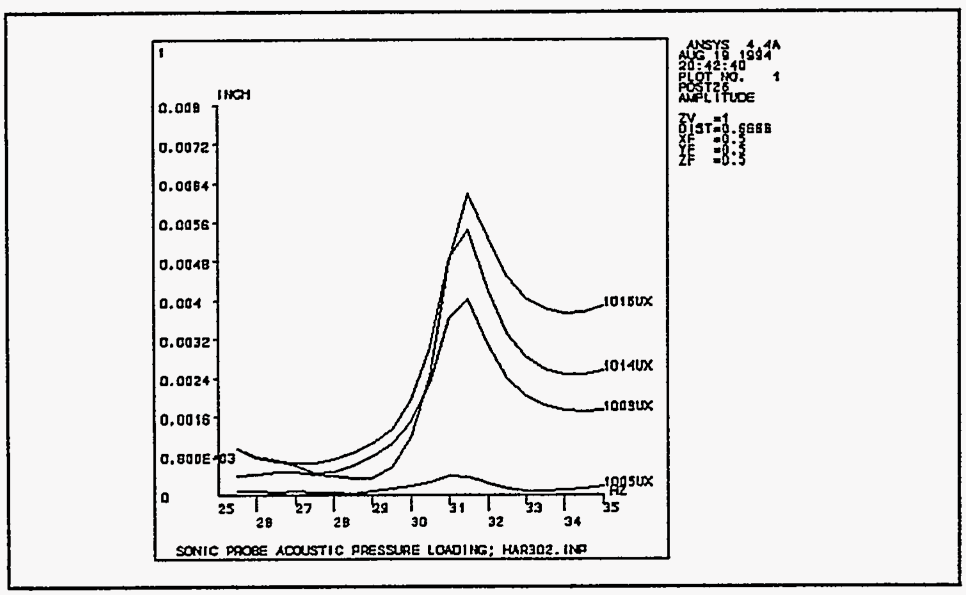


FIGURE 10. DISPLACEMENT AMPLITUDE RESPONSE (UX)
(FORCING FREQUENCY RANGE 59.5 Hz - 80.5 Hz; 0% ADDED MASS).

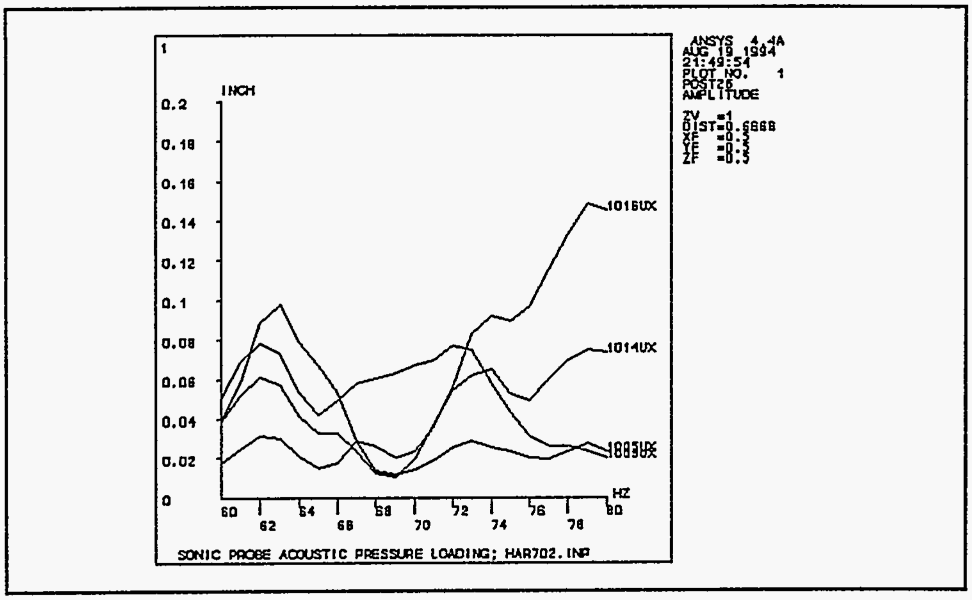


FIGURE 11. DISPLACEMENT AMPLITUDE RESPONSE (UX)
(FORCING FREQUENCY RANGE 59.5 Hz - 80.5 Hz; 0% ADDED MASS).

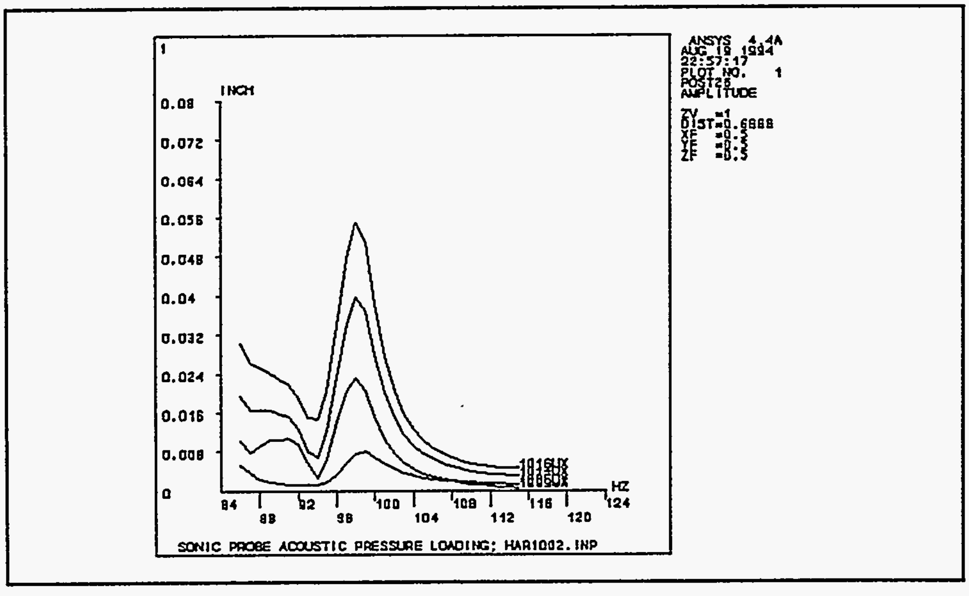


FIGURE 12. DISPLACEMENT AMPLITUDE RESPONSE (UX)
(FORCING FREQUENCY RANGE: 85 Hz - 115 Hz; 0% ADDED MASS).

## NOTATIONS

For Tables 2 through 5

| | | |
|---|---|---|
| SXT | = | centroidal longitudinal stress in the top fiber of the element |
| SYT | = | centroidal circumferential stress in the top fiber of the element |
| SHT | = | centroidal shear stress in the top fiber of the element |
| SXM | = | centroidal longitudinal stress in the mid-fiber of the element |
| SYM | = | centroidal circumferential stress in the mid-fiber of the element |
| SHM | = | centroidal shear stress in the mid-fiber of the element |
| SXB | = | centroidal longitudinal stress in the bottom fiber of the element |
| SYB | = | centroidal circumferential stress in the bottom fiber of the element |
| SHB | = | centroidal shear stress in the bottom fiber of the element |

For Table 6

| | | |
|---|---|---|
| SXT | = | nodal longitudinal stress in the top fiber of the element |
| SYT | = | nodal circumferential stress in the top fiber of the element |
| SHT | = | nodal shear stress in the top fiber of the element |
| SXM | = | nodal longitudinal stress in the mid-fiber of the element |
| SYM | = | nodal circumferential stress in the mid-fiber of the element |
| SHM | = | nodal shear stress in the mid-fiber of the element |
| SXB | = | nodal longitudinal stress in the bottom fiber of the element |
| SYB | = | nodal circumferential stress in the bottom fiber of the element |
| SHB | = | nodal shear stress in the bottom fiber of the element |

## REFERENCES

ADVENT, 1994, *Structural Analysis and Evaluation of the 241SY101 tank for Sonic Probe Loadings to Support SY101 Hydrogen Mitigation*, ADVENT Calculation No. 94007.4.1, Rev. 0, ADVENT Engineering Services, San Ramon, California.

ASME, 1989, *Boiler and Pressure Vessel Code, Nuclear Power Plant Components,* Section III, Division 1, American Society of Mechanical Engineers, New York, New York.

Blevins, R. D., 1990, *Flow-Induced Vibration*, Second Edition, Van Nostrand Reinhold, New York, New York.

SASI, 1989, *ANSYS Reference Manual*, Swanson Analysis Systems, Inc., Houston, Pennsylvania.