SEISMIC DESIGN FOR STEAM GENERATORS A MULTIPLE SYSTEM APPROACH

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SUMMARY

Certain classes of nuclear power stations such as CANDU-PHW plants have multiple reactor coolant systems. The typical CANDU 600MW(e) plant has 4 such systems functioning within each reactor building, with the major components in each coolant loop being the steam generator. These steam generators are clustered about the reactor and are supported independently, but in similar fashion by the boiler room floor. Although the component steam generators were designed as identical units, it was found that differences in operating conditions and variations in lateral supports caused a detuning effect between the 4 generators, with a consequent decrease in the effective mass ratio and an increase in response. It is the purpose of this paper to extend the concept of mass ratio to multiple system installations.

To study the effect of mass ratio on multiple system installations, a 5 degree-of-freedom model was created, consisting of one mass M, uncoupled frequency Ω, and 4 component oscillators of mass εM, individual frequency ω_j $(j=1,4)$, representing the primary structure and steam generators, respectively, such that in the limit, as the mass ratio $\varepsilon \rightarrow 0$, the main mass is effectively decoupled from the system and the small oscillators respond independently to the movement of the mass M as determined by standard floor response spectra. The equations of motion were normalized with respect to the mean frequency $\bar{\omega}$ of the steam generators and the response of the system was investigated for the condition of internal resonance defined by the approximate equality $\Omega \approx \omega_j$ $(j=1,4)$. The equations of motion were solved by modal analysis and direct and modal time integration, with the bulk of the parametric study carried out by means of modal analysis. The parametric study focused on the mass ratio, the detuning $\pm\Delta_j$ $(j=1,4)$ of the component oscillators, and the detuning $\pm\lambda$ of the primary mass relative to the mean frequency $\bar{\omega}$ of the cluster. By defining limits of the detuning range from 0% to $\pm 20\%$ of $\bar{\omega}$, graphs of minimum and maximum response for the range of $0.001 \leq \varepsilon \leq 0.1$ were determined. Within each range of detuning, the response of the oscillators depended on the infinite number of possible variations of Δ_j. The response was investigated, using both deterministic and random distribution of the component frequencies.

Guided by the results of the parametric study, it was concluded that the coupled building/steam generator analysis, carried out on only one system, with the weight of the remaining three systems lumped into the building structure, would provide conservative but reasonable results; whereas, an analysis modelling all 4 systems would not only lead to a more extensive and complicated analysis but would also underestimate the response. A description of the coupled analysis and representative results of the steam generator accelerations and displacement are presented in the paper.

1. Introduction

Certain classes of nuclear power stations such as CANDU-PHW plants have multiple reactor coolant loops. The typical CANDU 600MW(e) plant has 2 such systems functioning within each reactor building, with two of the major components in each coolant loop being the steam generators. These steam generators are clustered about the reactor and are supported independently, but in similar fashion by the boiler room floor. Although the component steam generators were designed as identical units, it was found that differences in operating conditions and variations in lateral supports caused a detuning effect between the 4 generators, with a consequent decrease in the effective mass ratio and an increase in response. It is the purpose of this paper to extend the concept of mass ratio to multiple system installation.

2. Statement of the Problem

Figure 1 is a representation of the CANDU Primary Heat Transport System. The elements that comprise each of the 2 coolant loops are the steam generators or boilers, the primary system pumps and headers. The steam generators are supported by columns and braced by lateral supports (not shown) against the interior concrete structure. To obtain an insight into the dynamic response of this multiple system installation, the 4 steam generators and the supporting building are idealized as a 5 mass, discrete system, as shown in Fig. 2. The building is represented by one mass M with the steam generators designated by masses which are each a fraction ε of M. The stiffnesses of the springs are as shown on the drawing.

The undamped equations of motion and of the system shown in Fig. 2 can be written as:

$$M\ddot{z} + (K + \sum_{j=1}^{4} k_j)z - \sum_{j=1}^{4} k_j x_j = - M\ddot{u}(t) \qquad 1(a)$$

$$m_i \ddot{x}_i + k_i x_i - k_i z = - m_i \ddot{u}(t) \qquad 1(b)$$

$$(i=1,\ldots,4)$$

Introducing the following substitutions

$$\tau = \bar{\omega}\, t \qquad 2(a)$$

$$\Omega^2 = K/M \qquad (b)$$

$$m_j = \varepsilon M \qquad (c)$$

$$\omega_j^2 = k_j/m_j \qquad (d)$$

$$\Delta_j = \omega_j/\bar{\omega} - 1 \qquad (e)$$

$$\lambda = \Omega/\bar{\omega} - 1 \qquad (j=1,\ldots 4,) \qquad (f)$$

Equations 1 may be re-written as follows:

$$\ddot{z} + \left\{ (1+\lambda)^2 + \varepsilon \sum_{j=1}^{4} (1+\Delta_j)^2 \right\} z \qquad 3(a)$$

$$- \varepsilon \sum_{j=1}^{4} (1+\Delta_j)^2 x_j \qquad = - \ddot{u}/\bar{\omega}^2$$

$$\ddot{x}_i + (1+\Delta_i)^2 x - (1+\Delta_i)^2 z \qquad = - \ddot{u}/\bar{\omega}^2 \qquad (b)$$

$(j=1,...4,)$

or in matrix notation as:

$$\ddot{X} + \left[K\right] X = - F \qquad 4$$

Viscous damping proportional to the K matrix may be added to equation 4 as follows:

$$\ddot{X} + \alpha \left[K\right] \dot{X} + \left[K\right] X = - F \qquad 5$$

It is of interest to note that as $\varepsilon \to 0.0$ eq. (1(a)) or mass M decouples from the system and the component oscillators represented by m_j may be excited to large amplitude response for the condition of internal resonance defined by $\lambda \approx \Delta_j$. For a single secondary system, Newmark [1] has provided upper bound response equations and Hadjian [2] has provided guidelines, based on mass ratio ε and frequency ratio λ, for systems requiring coupled or uncoupled analysis.

3. Approximate Analysis

Operating on the homogeneous system of eq. (3), the eigenvalues, eigenvectors, and participation factors may be extracted to carry out the usual spectral analysis. The modal acceleration vector may be determined as follows:

$$\left\{\bar{a}\right\}_j = \Gamma_j A_j \left\{\emptyset\right\}_j \qquad (6)$$

where $\left\{\bar{a}\right\}_j$ is the absolute modal acceleration vector, Γ_j the modal participation factor, A_j $(\hat{\omega}_j)$ the spectrum acceleration value, which is a function of the modal frequency $\hat{\omega}_j$ and modal damping, and $\left\{\emptyset\right\}_j$ which represents the mode shape. Since the system is investigated for the condition of internal resonance $\Omega \approx \omega_i$ $(i=1,...,4)$ it will be assumed, following the arguments used by Gelman [3], that A_j $(\hat{\omega}_j)$ is constant for all j and equal to $A(\bar{\omega})$. The maximum accelerations of each mass point can be obtained by absolute summation, Newmark [1]

$$a_i = A(\bar{\omega}) \sum_{j=1}^{5} \left| \Gamma_j \emptyset_{ij} \right| \qquad (7)$$

Realizing that $\Sigma \left| \Gamma_j \emptyset_{ij} \right|$ corresponding to mass M is unity, the response of $m_j (j=1,...,4)$ relative to M may be obtained by normalizing eq. (7).

$$a_i/A(\overline{\omega}) = \sum_{j=1}^{5} |\Gamma_j \emptyset_{ij}| \qquad (m_1, \ldots, m_4) \qquad (8)$$

Eq. (8) is a function of ε and the frequency detuning Δ_j and λ. To present the results of this study, the Δ_j are grouped within ranges $|\Delta_j| \leq R$ where R defines the maximum range of detuning. Within each range of detuning R, an infinite number of combinations may occur. However, it is possible to select the combination of Δ_j that produce maximum and minimum response. This was done and the results for $\lambda=0.0$, $0.001 \leq \varepsilon \leq 0.1$, and $R \leq 0.20$ are shown in Figure 4. For a given R=0.05, the histogram for 100 sets of Δ_j uniformly distributed in R, is shown in Figure 3.

Examining the results of Figure 4 it is seen that the upper bound response is asymptotic to the results predicted by Newmark [1] for a single oscillator attached to a mass $M(1+3\varepsilon)$. For completely uniform components $\Delta_j=0.0$, the results approach that of a single oscillator of mass $4\varepsilon M$ attached to M. The difference in response between identical components and those that may have large respective differences (say 20%) is 100%. That is detuning, within limits, actually increases response. Figure 3 gives an indication of response when frequencies are randomly selected within the range $R \leq 0.05$. The mean response is approximately 40% in excess of the response for R=0.0.

To provide verification of these results, the direct time integration of the equations from which Figures 3 and 4 were derived was carried out for various initial conditions. Figure 5 shows the response of the main Mass M and one of the component oscillators subjected to initial displacements. Shown in Table 1a are the parameters used and the response predicted from equation 8. Close agreement is evident. Different combinations of initial conditions, including pulse accelerations, were computed and the results proved essentially the same.

4. Direct Integration Analysis

The response of the system shown in Figure 2 to actual earthquake records will depend first upon the structural coefficients of the differential equations and second upon the nature of the earthquake ground motion. To investigate the actual response, the system as represented by eq. (5), was integrated in terms of its physical and normal coordinates for 3 representative earthquake records - Golden Gate N10E, Taft S69E, El Centro SOOE. Typical results in terms of the physical coordinates, for Taft, are shown in Figure 6. The numerical data associated with this Figure are shown in Table 2.

Intuitively, the longer the duration at high intensity shaking, the greater is the probability of the phase relationship of key modes aligning to cause maximum response. A comparison of the amplitude response of the secondary masses relative to the primary mass for Golden Gate, representative of a sharp short duration earthquake, and Taft, representative of a high level, long duration earthquake, showed an increase of 20% for Taft.

Difference in response due to changes in structural coefficients were much more pronounced. The structural coefficients can be classified into 4 groups (a) mass ratio ε, (b) component frequencies λ, Δ_j, (c) damping α, (d) the mean frequency $\overline{\omega}$. Different runs were carried out for the mass ratio ε. Using Taft, $\varepsilon=0.05$ and $\varepsilon=0.001$, 2% damping, all other parameters constant showed first, that the maximum response of the primary mass remained sensibly constant (i.e. no vibration absorber effect) and second that the responses of the component oscillators increased by a factor of just over 5 but less than the factor of $\sqrt{50}$ expected from interpolation of the results shown in Figure 4. Various combinations of component frequencies were analysed within a given range R. It was found that the ratio of maximum to minimum response given in Figure 4 was maintained. For example, the increase in response of a component oscillator subjected to Golden Gate, with all Δ_j=0.0 and Δ_j arranged for maximum response, range R=0.03, was 1.58 in close agreement with Figure 4. With damping values corresponding approximately to 0%, 2% and 4% modal damping, it was found that the response of the primary mass is essentially similar to that given by the response spectrum for the appropriate damping values but that the response of the component oscillators is dramatically affected. For example, for the El Centro record it was found that relative to the primary mass, the component experiences an increase of 8.6 for zero damping, 3.5 for 2% damping and 2.8 for 4% damping. The importance of the mean frequency $\bar{\omega}$ becomes self evident when it is recognized that in the limit as $\bar{\omega}\to\infty$ no dynamic amplification of the external excitation exists and the response becomes a static solution.

Interpolation of the direct integration results with the approximate analysis represented by equation 6, 7 and 8 will only yield consistent results if a_j and $A(\bar{\omega})$ are relative acceleration values. Since relative acceleration spectra are not readily available, and the combination rules of relative and absolute accelerations are not defined, it is recommended that eq. (8) and the results of Figure 4 be applied with the use of published response spectra recognizing that within certain ranges of the structural coefficients the results will be conservative.

5. Steam Generator Analysis

Guided by the parametric analysis and the results presented in Figure 4, the analysis on the actual steam generators was based on one unit interacting with the building and the mass of the remaining three added to the structure. It is to be noted that if four boilers had been modelled, there would be no guarantee that a slight perturbation would not produce higher response - the worst effect in this case being a factor of 2. The combined steam generator/building model consisted of a steam generator represented by 12 nodes and a building model consisting of 6 nodes (or 13 nodes depending on whether or not soil interaction had to be considered for that particular site). Preceeding the combined analysis, the equipment and building models were analysed separately and all possible internal resonance conditions were identified. If the preliminary parametric studies showed that internal resonance was indeed feasible, the frequencies in question of the two sub-systems were paired to be equal. The combined system was then run and the response recorded. Composite modal damping was based on a weighted average of strain energies in each material with a value of 5% critical assigned to the structure and 2% to the steam generator. For response calculation, paired modes were added by absolute sum before combining by the root sum of squares procedure. Other modes within the 10% frequency rule [4] were added by absolute sum only if interaction was actually possible. Figure 7 shows the acceleration response of the two most significant modes.

6. Conclusions

A general accepted axiom in structural dynamics is that detuning of the secondary structure with respect to the primary structure will reduce response. For multiple secondary systems, however, detuning, within limits, will actually increase response. This paper has carried out a parametric analysis on an idealized configuration of four secondary and one primary system and has confirmed that the increase for a 'worst' to 'best' combination of detuning can be a factor of approximately two. Generalizing, this means that the effective mass of the equipment, interacting with the primary structure, reduces to that of only one secondary system. For cases where 'worst' combination approach provides undue economic penalty, and the range of frequency variation of the components can be determined, this paper provides approximate procedures to give amplified response limits as a function of mass ratio and frequencies.

References

[1] Newmark N.M, 'Earthquake Response Analysis of Reactor Structures', Nuclear Engineering and Design 20 (1972) 303-322.

[2] Hadjian A.H., 'On the Decoupling of Secondary Systems for Seismic Analysis'. 6th World Conference on Earthquake Engineering, India, Jan. 1977.

[3] Gelman A.P., 'Determining Interaction Effects in the Seismic Analysis of Components' 2nd National Congress on Pressure Vessels and Piping, ASME 1975, 75-PVP-50.

[4] U.S. Atomic Energy Commission, Regulatory Guide 1.92 (1974) 'Combination of Modes and Spatial Components in Seismic Response Analysis'.

Nomenclature

$\bar{a}_j$	Modal accelerations	R	Range of detuning
a_j	Absolute accelerations	X	Coordinate vector
k_j	Stiffness secondary system	α	Damping coefficient
m_j	Mass secondary system	ϵ	Mass ratio m/M
t	Time	λ	Detuning M
$\ddot{u}(t), u''(\tau)$	Ground acceleration	τ	Normalized time
$x(t), x(\tau)$	Coordinates m_j	$\emptyset$	Normal coordinates
$z(t), z(\tau)$	Coordinates M	ω	Uncoupled frequencies m_j
A_j	Spectrum acceleration	$\bar{\omega}$	Mean frequency
F	Force vector	$\hat{\omega}_j$	Modal frequencies
K	Stiffness primary system	Γ_j	Participation factor
$\bar{K}$	Stiffness matrix	Δ_j	Detuning m_j
M	Primary mass	Ω	Uncoupled frequency M

TABLE #1

NUMERICAL DATA ASSOCIATED WITH FIGURE #5

$\varepsilon = 0.01$

$R = 0.05$

$\bar{\omega} = 13.0$ Hz

$\alpha = 0.00$ (No Damping)

Mass	Detuning λ, Δ_j	Response Equation 8	Normalized Int. Response	Natural Frequencies Mode	$\hat{\omega}/\bar{\omega}$
Primary	0.0	1.0	1.00	1	0.90020
Component #1	0.0370	6.495	5.98	2	0.96809
" #2	0.0228	6.659	6.00	3	1.00414
" #3	-0.0461	7.919	7.52	4	1.03042
" #4	-0.0098	7.198	6.70	5	1.11020

TABLE #2

NUMERICAL DATA ASSOCIATED WITH FIGURE #6

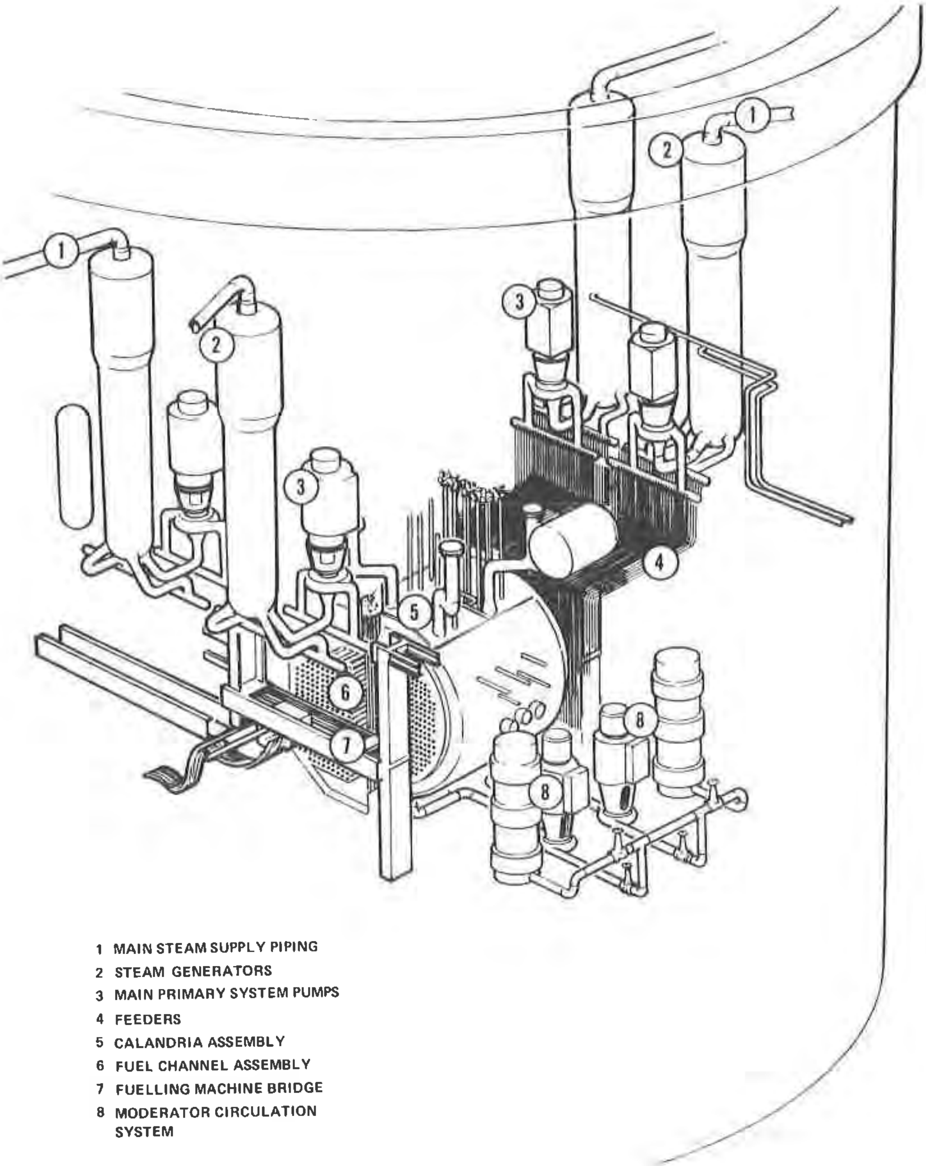
$\varepsilon = 0.01$

$R = 0.05$

$\bar{\omega} = 13.0$ Hz

$\alpha = 0.04$ (2% damping)

Mass	Detuning λ, Δ_j	Response Equation 8	Normalized Int. Response	Natural Frequencies Mode	$\hat{\omega}/\bar{\omega}$
Primary	0.0	1.0	1.00	1	0.91139
Component #1	0.0457	5.78	3.48	2	0.99495
" #2	0.0418	5.85	3.48	3	1.03130
" #3	0.0251	6.13	3.56	4	1.04386
" #4	-0.0178	7.25	5.17	5	1.12364



1 MAIN STEAM SUPPLY PIPING
2 STEAM GENERATORS
3 MAIN PRIMARY SYSTEM PUMPS
4 FEEDERS
5 CALANDRIA ASSEMBLY
6 FUEL CHANNEL ASSEMBLY
7 FUELLING MACHINE BRIDGE
8 MODERATOR CIRCULATION SYSTEM

FIGURE 1 CANDU PRIMARY HEAT TRANSPORT SYSTEM

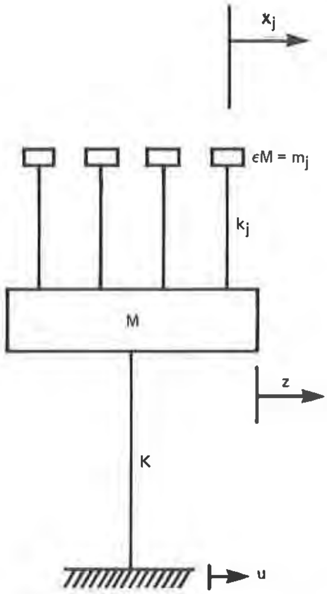


FIGURE 2 MULTIPLE SYSTEM MODEL

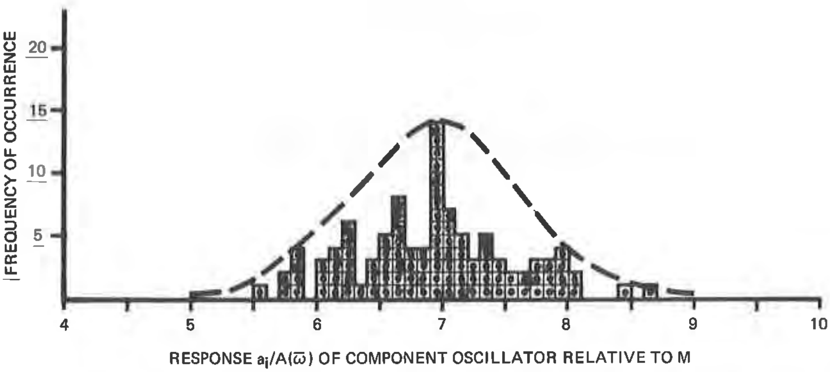


FIGURE 3 RESPONSE HISTOGRAM TO RANDOM DISTRIBUTION OF Δ_i (R = 0.05)

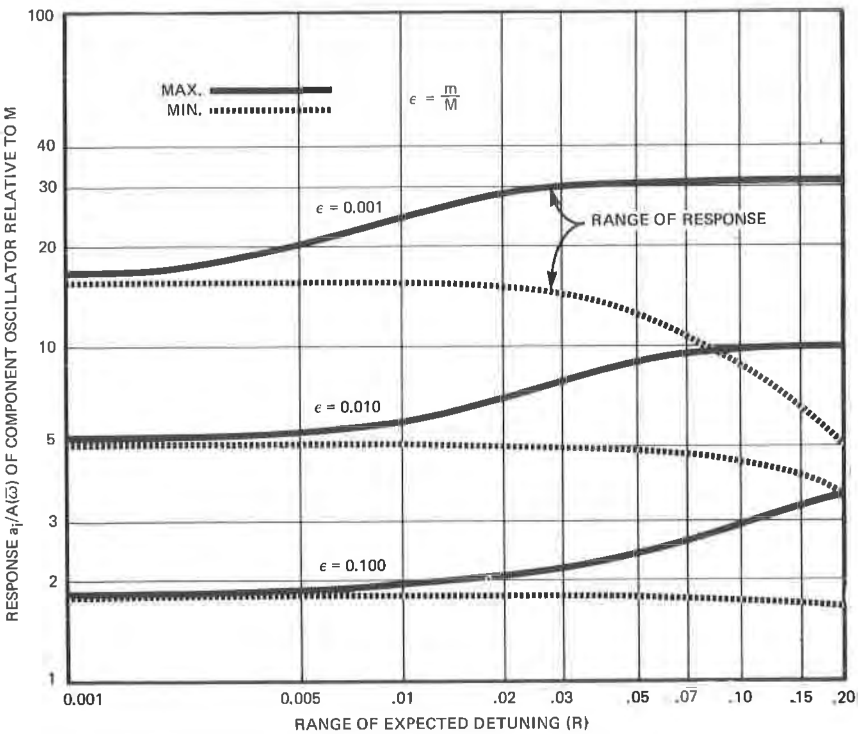


FIGURE 4 MAXIMUM AND MINIMUM RESPONSE VS. RANGE OF EXPECTED DETUNING

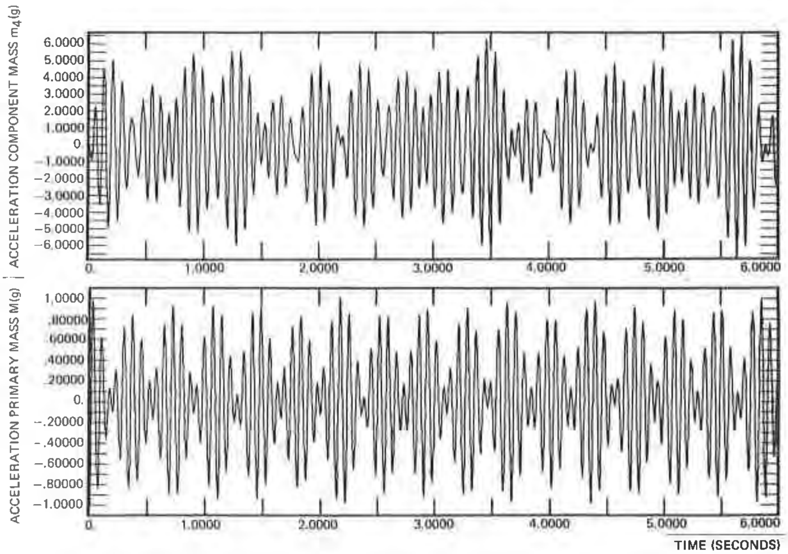


FIGURE 5 FREE VIBRATION RESPONSE (0% DAMPING)

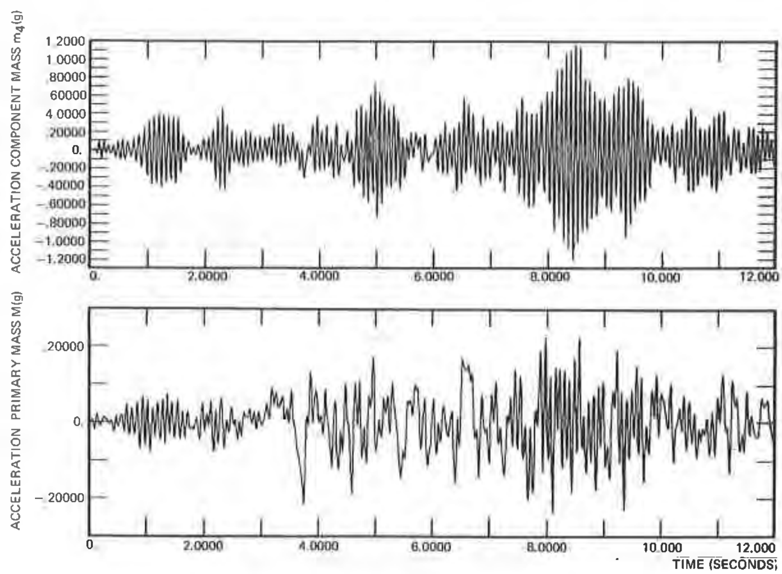
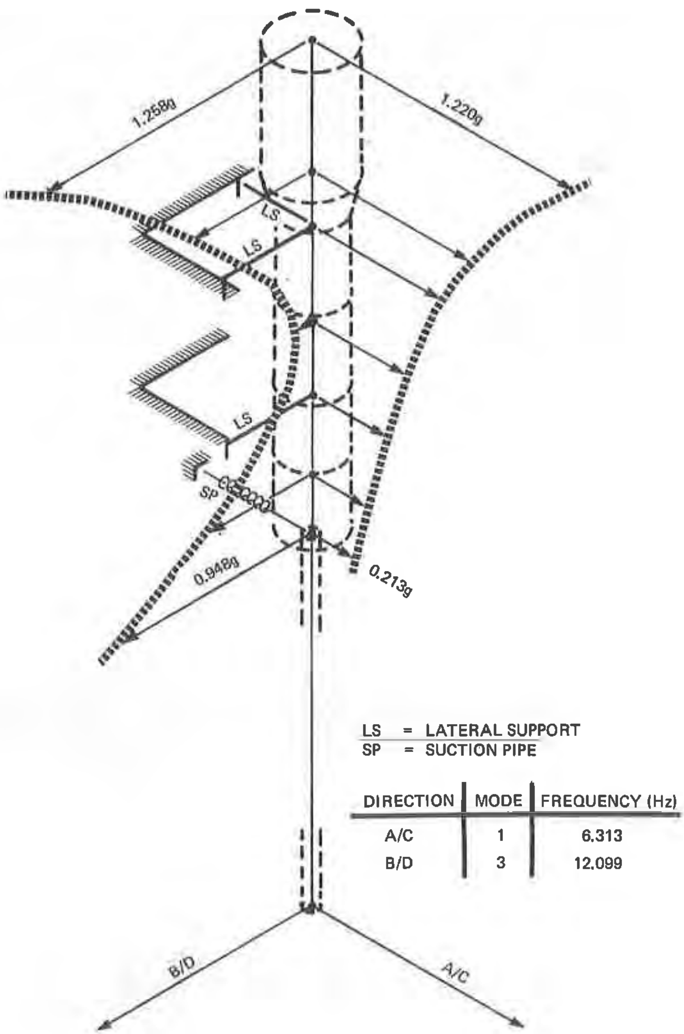


FIGURE 6 RESPONSE OF SYSTEM TO TAFT S69E (2% DAMPING)



DIRECTION	MODE	FREQUENCY (Hz)
A/C	1	6.313
B/D	3	12.099

FIGURE 7 MAXIMUM ACCELERATION MODES — COUPLED STEAM GENERATOR/BUILDING MODEL