

PB81-218497

REPORT NO.
UCB/EERC-81/05
APRIL 1981

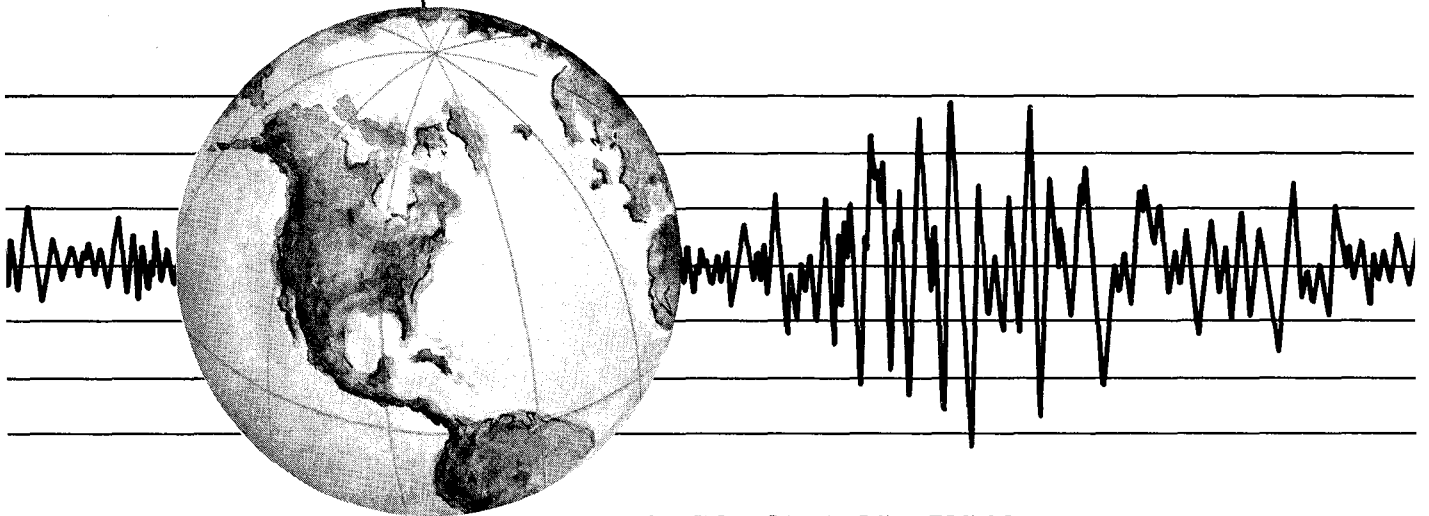
EARTHQUAKE ENGINEERING RESEARCH CENTER

DYNAMIC RESPONSE OF LIGHT EQUIPMENT IN STRUCTURES

by

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Report to the National Science Foundation



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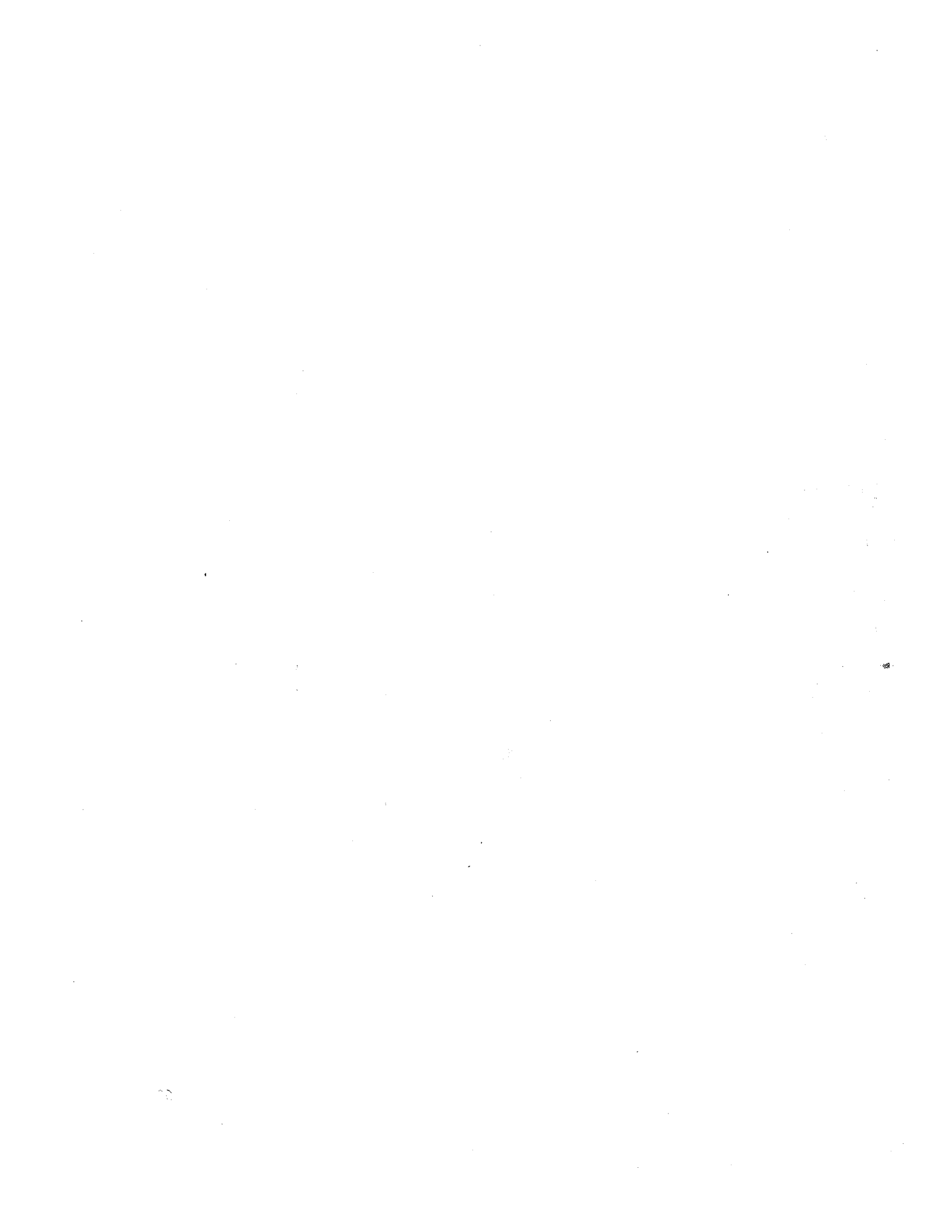
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REPORT DOCUMENTATION PAGE	1. REPORT NO. NSF/ENG-81005	2.	3. Recipient's Accession No. PBB# 218497
4. Title and Subtitle Dynamic Response of Light Equipment in Structures			5. Report Date April 1981
7. Author(s) A. Der Kiureghian, J.L. Sackman and B. Nour-Omid			6.
9. Performing Organization Name and Address Earthquake Engineering Research Center University of California, Berkeley 47th Street and Hoffman Blvd. Richmond, California 94804			8. Performing Organization Rept. No. UCB/EERC-81/05
12. Sponsoring Organization Name and Address National Science Foundation 1800 G Street, N.W. Washington, D. C. 20550			10. Project/Task/Work Unit No.
15. Supplementary Notes			11. Contract(C) or Grant(G) No. (C) (G) CME-7923651 ENG-7905906
			13. Type of Report & Period Covered
			14.

16. Abstract (Limit: 200 words)

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18. Availability Statement: Release Unlimited	19. Security Class (This Report)	21. No. of Pages
	20. Security Class (This Page)	22. Price



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A report on research sponsored by
the National Science Foundation

Report No. UCB/EERC-81/05
Earthquake Engineering Research Center
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ACKNOWLEDGMENT

This research work was sponsored by the National Science Foundation under Grants No. CME-7923651 and ENG-7905906. This sponsorship is gratefully acknowledged.



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INTRODUCTION

The dynamic analysis of light equipment in structures is of wide engineering interest. Control equipment, pressure vessels, pumps, etc., in power plants, communication and control items in transportation vehicles, and vital equipment in certain lifeline systems are common examples. In theory, the equipment can be incorporated in a dynamic model of the combined system as a part of the structure and analyzed by a conventional method. However, this approach in practice leads to two difficulties. First, for light equipment, the mass and stiffness matrices of the combined system will have elements with vastly different magnitudes, resulting in numerical difficulties in the dynamic analysis. Second, in design situations, it is often required to study many equipment items attached to a structure or several alternative attachment configurations or locations of a single item. Clearly, such an approach can be prohibitively costly because a multitude of dynamic models of the combined system must be constructed and analyzed. These difficulties can be overcome if, instead of using a conventional approach, an alternative procedure is followed taking advantage of the mismatch between the properties of the light equipment and the structure.

In the first part of this report, closed form expressions are derived for the dynamic properties of the combined equipment-structure system in terms of those of the structure alone and of the equipment alone. These are obtained using a single-degree-of-freedom model for the equipment and a multi-degree-of-freedom one for the structure. This development is based on a perturbation method that takes advantage of the intuitively obvious fact that, for light equipment, the dynamic properties of the combined system are not too different from those of the original structure. The effect of interaction between the equipment and the structure, which is especially significant when the equipment frequency coincides with one of the structure frequencies, is included in this analysis. By this approach, the numerical difficulties alluded to above are avoided and the computational effort becomes trivial.

Once the modal properties of the combined equipment-structure system are determined, any dynamic analysis procedure employing the mode-superposition method, e.g., the time-

history integration method for a deterministic input or the random vibration method with a power spectral density or a response spectrum description of the input, can be used to evaluate the response of the equipment. However, in using the random vibration method, a peculiar problem arises when the equipment is light and has a frequency close to one of the structure frequencies. Then, closely spaced modes occur in the combined system, the responses of which are known to be highly correlated (3). The central issue for an accurate evaluation of the equipment response then focuses on a proper accounting of this correlation in the superposition of modal responses.

In the second part of this report, a mode-superposition procedure for the response of the equipment including the effect of closely spaced modes is developed. First, the random vibration approach for response to a stationary input described through a power spectral density function is presented. This forms the basis for development of the response spectrum method in the second portion which is applicable under a set of broad conditions. In this method, which is of great interest in earthquake engineering, the equipment response is obtained directly in terms of information readily available to the designer; namely, the dynamic properties of the structure alone, the dynamic properties of the equipment alone, and the design response spectrum describing the input into the structure. These results should prove to be especially effective in design applications where a multitude of analyses is required.

The method presented here is more comprehensive and at the same time is computationally more efficient than any of the methods currently used. The existing methods (e.g., Refs. 6, 8, 9 and 12) invariably ignore one or more of the problems associated with the response of light equipment in structures, e.g., interaction between equipment and structure, effect of coincidence of frequencies, and correlation between closely spaced frequencies. Most common among the existing methods is the floor spectrum procedure where the response spectrum associated with the equipment attachment point in the structure is first determined, ignoring the equipment-structure interaction, and then used as the input into the equipment (e.g., Ref. 8). With the present results, the conventional floor spectrum is produced simply by setting the

mass of the equipment to zero, resulting in an algorithm simpler than any current one. Furthermore, with the present method it becomes possible to efficiently generate floor spectra which include the effect of equipment-structure interaction.

In the final part of this report, numerical examples are presented in order to illustrate the accuracy and the simplicity of the method. These include comparisons between the dynamic properties of the combined equipment-structure system obtained by an exact analysis and by the present method, and comparisons between the equipment response as obtained from time-history integrations and from the response spectrum formulation developed herein. In general, excellent agreement is obtained between the results from the two analyses.

DYNAMIC PROPERTIES OF EQUIPMENT-STRUCTURE SYSTEM

Consider an n -degree-of-freedom structure with mass, damping, and stiffness matrices \mathbf{M} , \mathbf{C} , and \mathbf{K} having elements m_{ij} , c_{ij} , and k_{ij} , $i, j = 1, 2, \dots, n$, respectively. Assume the structure has classical modes with natural frequencies ω_i , modal vectors $\Phi_i = [\phi_{1i} \ \phi_{2i} \ \dots \ \phi_{ni}]^T$, and modal damping coefficients ζ_i , where a superposed T denotes a transpose. Consider an equipment item, modeled as a single-degree-of-freedom oscillator of mass m_e , natural frequency ω_e , and damping coefficient ζ_e , attached to the structure such that it is only affected by motion in the k -th structure degree of freedom. The combined system, see Fig. 1, is an $n+1$ -degree sys-

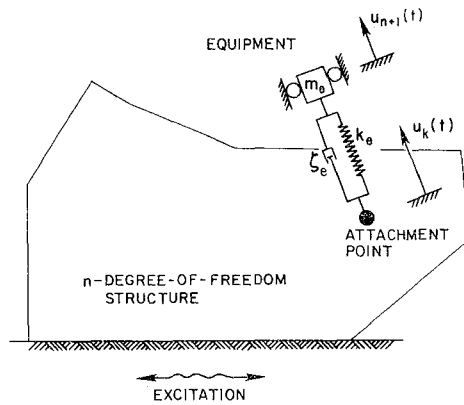


Figure 1. Equipment-Structure System

tem with $n+1$ modes. In general, the equipment frequency can be close or equal, i.e., nearly or perfectly tuned, to a natural frequency of the structure, say ω_l . Then, two closely spaced modes will occur in the combined system which require special treatment. It is the objective here to determine the dynamic properties of the combined system, including the case where tuning occurs, in terms of the properties of the two subsystems when the equipment mass is small compared to that of the structure. It is subsequently shown that for such conditions the modes of the combined system are closely related to those of the structure alone. Each structure mode, with contributions from other modes, produces a corresponding mode in the combined system. In addition, all the structure modes together with the equipment generate a "new" mode of the combined system. For convenience, the "new" mode will be identified by the subscript zero with the other modes retaining their original numbers. This process is conceptually illustrated in Fig. 2.

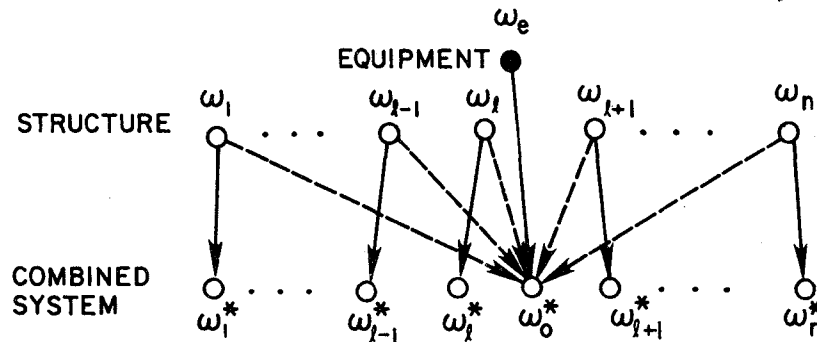


Figure 2. Schematic Representation of Modal Frequencies

The system of equations which determine the i -th ($i=0,1,\dots,n$) frequency and mode shape of the combined system may be obtained from the known properties of the two subsystems as

$$\mathbf{K}^* \Phi_i^* = \omega_i^{*2} \mathbf{M}^* \Phi_i^* \quad (1)$$

where a superscript asterisk has been used to denote values associated with the combined system. In the above, \mathbf{K}^* and \mathbf{M}^* are the stiffness and mass matrices for the combined system, and ω_i^* and Φ_i^* are the i -th natural frequency and mode shape, given by

$$\mathbf{K}^* = \left[\begin{array}{cccc|c} k_{11} & \dots & k_{1k} & \dots & k_{1n} & 0 \\ \cdot & & \cdot & & \cdot & \cdot \\ \cdot & & \cdot & & \cdot & \cdot \\ k_{k1} & \dots & k_{kk} + m_e \omega_e^2 & \dots & k_{kn} & -m_e \omega_e^2 \\ \cdot & & \cdot & & \cdot & \cdot \\ \cdot & & \cdot & & \cdot & \cdot \\ k_{n1} & \dots & k_{nk} & \dots & k_{nn} & 0 \\ \hline 0 & \dots & -m_e \omega_e^2 & \dots & 0 & m_e \omega_e^2 \end{array} \right]$$

$$\mathbf{M}^* = \left[\begin{array}{cccc|c} m_{11} & \dots & m_{1k} & \dots & m_{1n} & 0 \\ \cdot & & \cdot & & \cdot & \cdot \\ \cdot & & \cdot & & \cdot & \cdot \\ m_{k1} & \dots & m_{kk} & \dots & m_{kn} & 0 \\ \cdot & & \cdot & & \cdot & \cdot \\ \cdot & & \cdot & & \cdot & \cdot \\ m_{n1} & \dots & m_{nk} & \dots & m_{nn} & 0 \\ \hline 0 & \dots & 0 & \dots & 0 & m_e \end{array} \right]$$

and

$$\Phi_i^* = \left\{ \begin{array}{c} \phi_{1i}^* \\ \cdot \\ \cdot \\ \phi_{ki}^* \\ \cdot \\ \cdot \\ \phi_{ni}^* \\ \hline \phi_{n+1,i}^* \end{array} \right\}$$

where the $n+1$ -th degree of freedom has been assigned to the equipment motion. Partitioning the matrices as shown and expanding Eq. 1, one obtains

$$\mathbf{K} \left\{ \begin{array}{c} \phi_{1i}^* \\ \cdot \\ \cdot \\ \phi_{ki}^* \\ \cdot \\ \cdot \\ \phi_{ni}^* \end{array} \right\} + \left\{ \begin{array}{c} 0 \\ \cdot \\ \cdot \\ m_e \omega_e^2 (\phi_{ki}^* - \phi_{n+1,i}^*) \\ \cdot \\ \cdot \\ 0 \end{array} \right\} = \omega_i^{*2} \mathbf{M} \left\{ \begin{array}{c} \phi_{1i}^* \\ \cdot \\ \cdot \\ \phi_{ki}^* \\ \cdot \\ \cdot \\ \phi_{ni}^* \end{array} \right\} \quad (2a)$$

and

$$-m_e \omega_e^2 (\phi_{ki}^* - \phi_{n+1,i}^*) = \omega_i^{*2} m_e \phi_{n+1,i}^* \quad (2b)$$

From Eq. 2b,

$$\phi_{n+1,i}^* = - \frac{\omega_e^2}{\omega_i^{*2} - \omega_e^2} \phi_{ki}^* = \alpha_i \phi_{ki}^* \quad (3)$$

where $\alpha_i = \phi_{n+1,i}^* / \phi_{ki}^*$ is the modal amplification factor of the equipment relative to the attachment point. Substituting this in Eq. 2a, yields

$$\mathbf{K} \begin{pmatrix} \phi_{1i}^* \\ \vdots \\ \phi_{ki}^* \\ \vdots \\ \phi_{ni}^* \end{pmatrix} + \begin{pmatrix} 0 \\ \vdots \\ m_e \phi_{ki}^* \omega_i^{*2} \omega_e^2 / (\omega_i^{*2} - \omega_e^2) \\ \vdots \\ 0 \end{pmatrix} = \omega_i^{*2} \mathbf{M} \begin{pmatrix} \phi_{1i}^* \\ \vdots \\ \phi_{ki}^* \\ \vdots \\ \phi_{ni}^* \end{pmatrix} \quad (4)$$

The single nonzero term in the second vector on the left hand side is small for light equipment and, therefore, will only slightly modify the frequencies and mode shapes of the original structure. The well known Rayleigh's quotient establishes that first-order errors in mode shapes result in second-order errors in frequencies (5). Thus, as a first approximation, it is assumed here that the portions of modal vectors corresponding to structural degrees of freedom retain their shapes after the equipment is attached; i.e. it is assumed that $\phi_{mi}^* = \phi_{mi}$, for $m = 1, 2, \dots, n$ and for $i \neq 0$. This will lead to a second-order approximation in the frequencies which is desirable for the computation of the response, since it can be shown that accuracy in frequencies is more critical than that in mode shapes. It should be pointed out that this approximation scheme is only appropriate when the major contribution to Φ_i^* , the i -th mode shape of the combined equipment-structure system, comes from Φ_i , the i -th mode shape of the structure. This precludes applying this approximation to cases where the equipment is tuned to a cluster of closely spaced structure modes. In such situations, all of the modes of the cluster will participate in a significant way in generating the mode shape of the combined system. The method presented here can be extended to deal with this problem.

Premultiplying Eq. 4 by Φ_i^T , one obtains

$$\omega_i^2 M_i + m_e \phi_{ki}^2 \frac{\omega_i^{*2} \omega_e^2}{\omega_i^{*2} - \omega_e^2} = \omega_i^{*2} M_i \quad (5)$$

where $M_i = \Phi_i^T \mathbf{M} \Phi_i = \omega_i^{-2} \Phi_i^T \mathbf{K} \Phi_i$ is the i -th modal mass of the structure. This yields

$$(1 + \beta_i) \left(\frac{\omega_i^*}{\omega_i} \right)^4 - 2 \left(1 + \frac{\beta_i + \gamma_i}{2} \right) \left(\frac{\omega_i^*}{\omega_i} \right)^2 + 1 = 0 \quad (6)$$

where $\beta_i = (\omega_i^2 - \omega_e^2)/\omega_e^2$ and $\gamma_i = m_e/(M_i/\phi_{ki}^2)$ are the *detuning parameter* and the *effective mass ratio*, respectively, for mode i . These two parameters play important roles in the subsequent analysis. The solution of Eq. 6 is

$$\left(\frac{\omega_i^*}{\omega_i} \right)^2 = \begin{cases} \frac{1 + \frac{\beta_i + \gamma_i}{2} - \left[\left(1 + \frac{\beta_i + \gamma_i}{2} \right)^2 - (1 + \beta_i) \right]^{1/2}}{1 + \beta_i}, & \beta_i < 0 \\ \frac{1 + \frac{\beta_i + \gamma_i}{2} + \left[\left(1 + \frac{\beta_i + \gamma_i}{2} \right)^2 - (1 + \beta_i) \right]^{1/2}}{1 + \beta_i}, & \beta_i \geq 0 \end{cases} \quad (7)$$

where the proper root has been chosen according to the sign of β_i so as to produce ω_i^* near ω_i . (The other root, which yields ω_i^* near ω_e , is the contribution to the "new" mode. This contribution is subsequently retrieved by imposition of orthogonality between the modes.) For $\beta_i = 0$, either root is acceptable; here, the second one is chosen for definiteness. Substituting Eq. 7 in Eq. 3, the modal amplification factor is obtained as

$$\alpha_i = \begin{cases} \frac{1}{\frac{\beta_i + \gamma_i}{2} - \left[\left(1 + \frac{\beta_i + \gamma_i}{2} \right)^2 - (1 + \beta_i) \right]^{1/2}}, & \beta_i < 0 \\ \frac{1}{\frac{\beta_i + \gamma_i}{2} + \left[\left(1 + \frac{\beta_i + \gamma_i}{2} \right)^2 - (1 + \beta_i) \right]^{1/2}}, & \beta_i \geq 0 \end{cases} \quad (8)$$

It is noted that the magnitude of α_i can be very large for small β_i which occurs in the case of near or perfect tuning.

The new mode, denoted by Φ_0^* , is obtained through the requirement of orthogonality with the previous modes. Let

$$\Phi_0^* = \begin{Bmatrix} \phi_{10}^* \\ \cdot \\ \cdot \\ \phi_{n0}^* \\ 1 \end{Bmatrix} = \begin{Bmatrix} \Phi_0 \\ 1 \end{Bmatrix} \quad (9)$$

be the new mode shape, scaled so that $\phi_{n+1,0}^* = 1$. Orthogonality requires that $\Phi_i^{*T} \mathbf{M}^* \Phi_0^* = 0$, for $i \neq 0$. Using \mathbf{M}^* , Φ_i^* , and Φ_0^* as described above, one then obtains

$$\Phi_i^{*T} \mathbf{M}^* \Phi_0^* = \Phi_i^T \mathbf{M} \Phi_0 + m_e \alpha_i \phi_{ki} = 0, \quad i = 1, 2, \dots, n \quad (10)$$

which gives

$$\Phi_i^T \mathbf{M} \Phi_0 = -m_e \alpha_i \phi_{ki}, \quad i = 1, 2, \dots, n \quad (11)$$

Collecting all such equations in a matrix form,

$$\Phi^T \mathbf{M} \Phi_0 = - \begin{Bmatrix} m_e \alpha_1 \phi_{k1} \\ \cdot \\ \cdot \\ m_e \alpha_n \phi_{kn} \end{Bmatrix} \quad (12)$$

where $\Phi = [\Phi_1 \ \Phi_2 \ \dots \ \Phi_n]$ is the modal matrix of the structure alone. It is easy to show that $\Phi \mathbf{D}^{-1}$ is the inverse of $\Phi^T \mathbf{M}$, where \mathbf{D} is a diagonal matrix with elements M_i . Premultiplying Eq. 12 by $\Phi \mathbf{D}^{-1}$, therefore, yields

$$\Phi_0 = - \Phi \mathbf{D}^{-1} \begin{Bmatrix} m_e \alpha_1 \phi_{k1} \\ \cdot \\ \cdot \\ m_e \alpha_n \phi_{kn} \end{Bmatrix} = - \begin{Bmatrix} \sum_i \alpha_i \gamma_i \phi_{1i} / \phi_{ki} \\ \cdot \\ \cdot \\ \sum_i \alpha_i \gamma_i \phi_{ni} / \phi_{ki} \end{Bmatrix} \quad (13)$$

Note that all structure modes contribute to the generation of the new mode. However, observe that generally the shape of the new mode will be dominated by that structure mode which has a frequency closest to the equipment frequency, i.e. the l -th mode, because its modal amplification factor, α_l , is the largest. To determine the frequency of the new mode, Eqs. 9 and 13 are used in conjunction with Eq. 2(b) giving

$$\omega_0^* = \left(1 + \sum_i \alpha_i \gamma_i\right)^{1/2} \omega_e \quad (14)$$

Observe that the frequency of the new mode is only slightly removed from the equipment frequency, with the difference being contributed by all structure modes. Clearly, the largest contribution to this difference comes from the l -th mode.

To investigate the nature of the error in the mode shapes and frequencies, a norm of the second vector on the left-hand side of Eq. 4 is examined in comparison with the corresponding norms of the two remaining vectors. This is accomplished by premultiplying Eq. 4 by Φ_l^T . For light equipment, the scalars thus obtained from the two remaining vectors are of order $\omega_l^2 M_l$, whereas that from the former vector is of order $\omega_l^2 M_l \gamma_l \omega_e^2 / (\omega_l^{*2} - \omega_e^2)$. The ratio of these norms, r , which is a measure of the modification of frequencies and mode shapes of the structure resulting from the addition of the light equipment, can be written as

$$r = \frac{\gamma_l}{\left(\frac{\omega_l^*}{\omega_l}\right)^2 (1 + \beta_l) - 1} \quad (15)$$

where the ratio of frequencies is given in terms of β_l and γ_l in Eqs. 7. It is easy to show that in the extreme cases of gross detuning, i.e. when $|\beta_l|$ is large, and of perfect tuning, i.e. when β_l is zero, the modification factor r is of order γ_l and $\sqrt{\gamma_l}$, respectively. A plot of $|r/\sqrt{\gamma_l}|$ versus β_l is shown in Fig. 3 for $\gamma_l = 0.001, 0.01$ and 0.05 . Observe that for small γ_l , the modification term rapidly increases very near tuning, i.e. when $\beta_l \rightarrow 0$. This leads to a formal definition of gross detuning as values of β_l for which $|r/\sqrt{\gamma_l}| \ll 1$. From Eq. 15, this definition can be expressed as $|\beta_l| \gg \sqrt{\gamma_l}$.

It is observed from Eq. 15 that errors in the estimated mode shapes and frequencies are largest for modes with frequencies closest to the equipment frequency. To achieve a uniform accuracy, it is necessary to improve the results for modes having frequencies near the equipment frequency. For simplicity, it is assumed herein that all but the l -th mode are well spaced from the equipment frequency. Therefore, a refinement of only mode l need be considered.

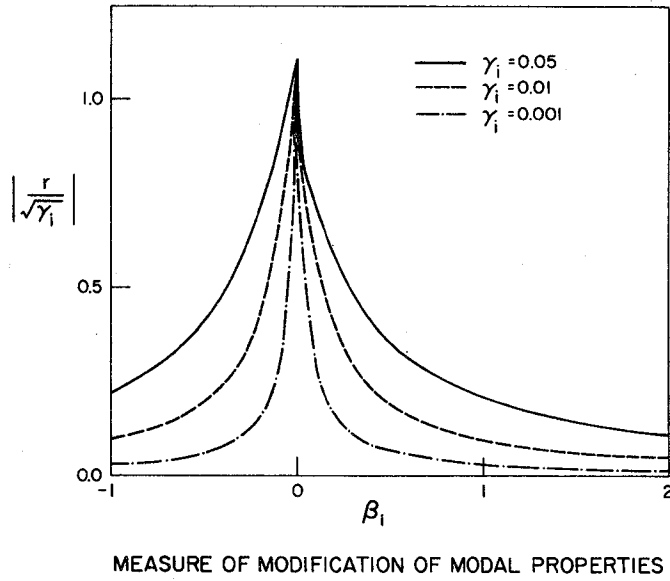


Figure 3. Measure of Modification of Modal Properties

To construct a refined shape for the l -th mode, the same procedure is used as for the new mode above, i.e. the refined l -th mode is obtained by imposition of orthogonality with all other modes, including the new mode. This results in

$$\Phi_l^* = \begin{pmatrix} \sum_{i \neq l} \alpha_i \gamma_i \phi_{li} / \phi_{ki} - \frac{1}{\alpha_l} \phi_{li} / \phi_{kl} \\ \vdots \\ \sum_{i \neq l} \alpha_i \gamma_i \phi_{ni} / \phi_{ki} - \frac{1}{\alpha_l} \phi_{ni} / \phi_{kl} \\ -1 \end{pmatrix} \quad (16)$$

where, for the sake of uniformity with Φ_0^* , this mode has been scaled such that $\phi_{n+1,l}^* = -1$. In this derivation, only the dominant terms in Φ_0^* , i.e. those arising from the l -th mode, have been included. This is consistent with the order of accuracy in other modes.

The refined shape of the l -th mode could be used to obtain a more refined estimate of ω_l^* from Eq. 2(b). However, it is clear from Rayleigh's quotient that the second-order correction in the mode shape will result in only a third-order correction in the frequency, which is neglected for uniformity in approximation.

It is assumed herein that the damping in the equipment can be modeled by a viscous damper connecting the equipment to the attachment point, see Fig. 1. Since the structure is assumed to have modal damping, intuitively it would appear that for light damping the combined system will also very nearly have modal damping. The preceding perturbation scheme can be used to determine the damping coefficients for the combined system in terms of those of the two subsystems. To this end, the modal damping relation

$$2\zeta_i^* \omega_i^* M_i^* = \Phi_i^{*T} \mathbf{C}^* \Phi_i^* \quad (17)$$

is used, where \mathbf{C}^* , the damping matrix of the combined system, is given by

$$\mathbf{C}^* = \left[\begin{array}{cccc|c} c_{11} & \dots & c_{1k} & \dots & c_{1n} & 0 \\ \cdot & & \cdot & & \cdot & \cdot \\ \cdot & & \cdot & & \cdot & \cdot \\ \cdot & & \cdot & & \cdot & \cdot \\ c_{k1} & \dots & c_{kk} + 2\zeta_e \omega_e m_e & \dots & c_{kn} & -2\zeta_e \omega_e m_e \\ \cdot & & \cdot & & \cdot & \cdot \\ \cdot & & \cdot & & \cdot & \cdot \\ \cdot & & \cdot & & \cdot & \cdot \\ c_{n1} & \dots & c_{nk} & \dots & c_{nn} & 0 \\ \hline 0 & \dots & -2\zeta_e \omega_e m_e & \dots & 0 & 2\zeta_e \omega_e m_e \end{array} \right]$$

Using the preceding in conjunction with the expressions of β_i and γ_i as well as the identity

$$2\zeta_i \omega_i M_i = \Phi_i^T \mathbf{C} \Phi_i, \text{ Eq. 17 gives}$$

$$\zeta_i^* = \begin{cases} \frac{\sqrt{1+\beta_i} \zeta_i + (1-\alpha_i)^2 \gamma_i \zeta_e}{\sqrt{1+\beta_i} (1+\alpha_i^2 \gamma_i)} \frac{\omega_i}{\omega_i^*}, & i = 1, 2, \dots, n \\ \frac{\sum_j \sqrt{1+\beta_j} \alpha_j^2 \gamma_j \zeta_j + \left(1 + \sum_j \alpha_j \gamma_j\right)^2 \zeta_e}{1 + \sum_j \alpha_j^2 \gamma_j} \frac{\omega_e}{\omega_i^*}, & i = 0 \end{cases} \quad (18)$$

In arriving at the result for ζ_i^* , the unrefined shape of the l -th mode was used for reasons of simplicity and consistency in approximation. Note that the ratios of frequencies in the preceding relations are close to unity and can be discarded.

To obtain the response of the combined system to specified input excitations, it is necessary to compute modal masses and participation factors. Using the mode shapes previously obtained, these are determined in closed form as

$$\begin{aligned}
 M_i^* &= \Phi_i^{*T} \mathbf{M}^* \Phi_i^* \\
 &= \begin{cases} (1 + \alpha_i^2 \gamma_i) M_i, & i \neq l, 0 \\ \left[1 + \alpha_l^2 \gamma_l \left(1 + \sum_{j \neq l} \alpha_j^2 \gamma_j \right) \right] \frac{M_l}{(\alpha_l \phi_{kl})^2}, & i = l \\ \left(1 + \sum_j \alpha_j^2 \gamma_j \right) m_e, & i = 0 \end{cases} \quad (19)
 \end{aligned}$$

and

$$\begin{aligned}
 \Gamma_i^* &= \frac{\Phi_i^{*T} \mathbf{M}^* \mathbf{R}^*}{M_i^*} \\
 &= \begin{cases} \frac{\Gamma_i + \alpha_i \gamma_i r_{n+1}^* / \phi_{kl}}{1 + \alpha_i^2 \gamma_i}, & i \neq l, 0 \\ \frac{\alpha_l \phi_{kl} \Gamma_l - \alpha_l^2 \gamma_l \left(\sum_{j \neq l} \alpha_j \phi_{kj} \Gamma_j - r_{n+1}^* \right)}{1 + \alpha_l^2 \gamma_l \left(1 + \sum_{j \neq l} \alpha_j^2 \gamma_j \right)}, & i = l \\ \frac{\sum_j \alpha_j \phi_{kj} \Gamma_j - r_{n+1}^*}{1 + \sum_j \alpha_j^2 \gamma_j}, & i = 0 \end{cases} \quad (20)
 \end{aligned}$$

where $\Gamma_i = \Phi_i^T \mathbf{M} \mathbf{R} / M_i$ is the participation factor associated with the i -th mode of the structure, in which $\mathbf{R} = [r_1 \ r_2 \ \cdots \ r_n]^T$ and $\mathbf{R}^* = [r_1 \ r_2 \ \cdots \ r_n \ r_{n+1}^*]^T$ are the conventional influence vectors coupling the input to the degrees of freedom of the structure and of the combined system, respectively.

PERFECTLY TUNED EQUIPMENT

It is instructive to examine the preceding results in the special case when the equipment is perfectly tuned to the l -th mode and grossly detuned with respect to all other modes. Using Eqs. 7 and 14, the modal frequencies in this case reduce to

$$\omega_i^* \approx \begin{cases} \omega_i, & i \neq l, 0 \\ \left(1 + \frac{\sqrt{\gamma_l}}{2}\right) \omega_l, & i = l \\ \left(1 - \frac{\sqrt{\gamma_l}}{2}\right) \omega_e, & i = 0 \end{cases} \quad (21)$$

where $\omega_l = \omega_e$ and terms only up to order $\sqrt{\gamma_l}$ have been retained. The corresponding results for modal amplification factors, obtained from Eqs. 8, are

$$\alpha_i \approx \begin{cases} -\frac{1}{\beta_i}, & i \neq l, 0 \\ -\frac{1}{\sqrt{\gamma_l}}, & i = l \end{cases} \quad (22)$$

These factors together with γ_i determine the mode shapes as

$$\Phi_i^* \approx \begin{cases} \phi_{li} \\ \vdots \\ \phi_{ni} \\ \frac{\phi_{ki}}{\beta_i} \end{cases}, \quad i \neq l, 0; \quad \Phi_l^* \approx \begin{cases} \sqrt{\gamma_l} \frac{\phi_{li}}{\phi_{kl}} - \sum_{i \neq l} \frac{\gamma_i}{\beta_i} \frac{\phi_{li}}{\phi_{ki}} \\ \vdots \\ \sqrt{\gamma_l} \frac{\phi_{nl}}{\phi_{kl}} - \sum_{i \neq l} \frac{\gamma_i}{\beta_i} \frac{\phi_{ni}}{\phi_{ki}} \\ -1 \end{cases}; \quad \Phi_0^* \approx \begin{cases} \sqrt{\gamma_l} \frac{\phi_{li}}{\phi_{kl}} + \sum_{i \neq l} \frac{\gamma_i}{\beta_i} \frac{\phi_{li}}{\phi_{ki}} \\ \vdots \\ \sqrt{\gamma_l} \frac{\phi_{nl}}{\phi_{kl}} + \sum_{i \neq l} \frac{\gamma_i}{\beta_i} \frac{\phi_{ni}}{\phi_{ki}} \\ +1 \end{cases} \quad (23)$$

The corresponding expressions for damping coefficients from Eqs. 18 are

$$\zeta_i^* \approx \begin{cases} \zeta_i, & i \neq l, 0 \\ \frac{\zeta_l + \zeta_e}{2}, & i = l, 0 \end{cases} \quad (24)$$

The modal masses from Eqs. 19 reduce to

$$M_i^* \approx \begin{cases} M_i, & i \neq l, 0 \\ 2m_e, & i = l, 0 \end{cases} \quad (25)$$

Note that the small modal masses associated with the l -th and 0-th modes result from the scaling used for these modes, i.e. from the scaling of their last components to 1 and -1 , respectively. The expressions for the participation factors from Eqs. 20 are

$$\Gamma_i^* \approx \begin{cases} \Gamma_i, & i \neq l, 0 \\ \frac{1}{2} \left(\frac{\Gamma_l}{\sqrt{\gamma_l}} \phi_{kl} - \sum_{j \neq l} \frac{\Gamma_j}{\beta_j} \phi_{kj} - r_{n+1}^* \right), & i = l \\ \frac{1}{2} \left(\frac{\Gamma_l}{\sqrt{\gamma_l}} \phi_{kl} + \sum_{j \neq l} \frac{\Gamma_j}{\beta_j} \phi_{kj} + r_{n+1}^* \right), & i = 0 \end{cases} \quad (26)$$

Observe that the participation factors for the l -th and 0-th modes are much larger than those for other modes as a consequence of the scaling mentioned above.

The preceding results for the case of perfect tuning to the l -th structure mode and gross detuning to all other structure modes exhibit a remarkable symmetry between the l -th and the 0-th modes of the combined system. Observe that their frequencies are equally spaced on each side of the tuning frequency and their damping coefficients and modal masses are equal. Furthermore, note that the expressions for their modal shapes and participation factors are symmetrical in form. It is also seen that the frequencies, damping coefficients, modal masses, and participation factors of the grossly detuned modes are unaffected by the addition of the light equipment.

RESPONSE OF EQUIPMENT-STRUCTURE SYSTEM

It is well known (e.g., see Ref. 5) that any response of a linear MDOF system to a prescribed input excitation can be obtained as a superposition of modal contributions. In particular, any response quantity of the equipment in the combined system under consideration can be written as

$$R(t) = \sum_{i=0}^n R_i(t) = \sum_{i=0}^n \Psi_i^* S_i(t) \quad (27)$$

where $R_i(t) = \Psi_i^* S_i(t)$ is the contribution of mode i to the response, Ψ_i^* is the i -th effective participation factor, and $S_i(t)$ is the i -th normal coordinate representing the response of an oscillator of frequency ω_i^* and damping coefficient ζ_i^* to the given excitation. The effective participation factor, Ψ_i^* , depends on the particular response sought and in general is given as a product of the i -th participation factor, Γ_i^* , and a linear combination of the components of the i -th modal vector. For example, for the displacement response of the equipment, it is simply

$\Gamma_i^* \phi_{n+1,i}^*$, and for its displacement relative to the attachment point, it is $\Gamma_i^*(\phi_{n+1,i}^* - \phi_{ki}^*)$.

It is interesting to examine the values of the effective participation factors for the equipment displacement in the case of perfect tuning. Using Eqs. 26, these become

$$\Psi_i^* \approx \begin{cases} -\frac{\Psi_i}{\beta_i}, & i \neq l, 0 \\ -\frac{1}{2} \left(\frac{\Psi_l}{\sqrt{\gamma_l}} - \sum_{j \neq l} \frac{\Psi_j}{\beta_j} - r_{n+1}^* \right), & i = l \\ +\frac{1}{2} \left(\frac{\Psi_l}{\sqrt{\gamma_l}} + \sum_{j \neq l} \frac{\Psi_j}{\beta_j} + r_{n+1}^* \right), & i = 0 \end{cases} \quad (28)$$

where $\Psi_i = \Gamma_i \phi_{ki}$, $i = 1, 2, \dots, n$, is the effective participation factor for the i -th mode for the displacement of the attachment point in the structure alone. It is important to observe the presence of $\sqrt{\gamma_l}$ in the denominators of the leading terms of Ψ_l^* and Ψ_0^* . Because of this term, the effective participation factors for these modes are two large numbers of opposite signs and almost equal magnitudes. This fact has an important bearing on the manner in which these two modes combine to produce the equipment response, as will be shown subsequently.

Depending on the type of input excitation, a deterministic or a probabilistic method is required to evaluate Eq. 27. For a deterministic input time history, any suitable time-integration method could be used with no problems anticipated. On the other hand, if the excitation is stochastic, particular care must be exercised to account for the correlation between normal coordinates, $S_i(t)$, which can be significant for modes with closely spaced frequencies, as would occur for the case of tuned or nearly tuned equipment. Because of this, and since the probabilistic approach is of such current interest (e.g., in earthquake engineering, offshore structures, flight structures), this problem is addressed in detail in the next two sections. Specifically, in the following section, the response of the equipment to a stationary excitation of the combined system described through an input power spectral density function is discussed. This provides the basis for the development of a response spectrum method, presented in the subsequent section, which properly accounts for closely spaced modes. These developments

make use of new results presented in Refs. 2 and 3 , which are especially pertinent to the problem of closely spaced modes.

RESPONSE TO STOCHASTIC INPUT: POWER SPECTRAL DENSITY APPROACH

Consider the stationary response of the combined equipment-structure system to a zero-mean stationary input excitation, $F(t)$, with a one-sided power spectral density $G_F(\omega)$. Using mode superposition, the power spectral density of the equipment response is

$$G_R(\omega) = \sum_{i=0}^n \sum_{j=0}^n \Psi_i^* \Psi_j^* G_F(\omega) H_i(\omega) \bar{H}_j(\omega) \quad (29)$$

where, for base input excitation, $H_i(\omega) = (\omega_i^{*2} - \omega^2 + 2i\zeta_i^* \omega_i^* \omega)^{-1}$ is the complex frequency-response function for mode i , and a superposed bar denotes the complex conjugate.

It has been shown (10) that most response quantities of interest can be expressed in terms of the first three moments, λ_0 , λ_1 , and λ_2 , of the response power spectral density

$$\lambda_m = \int_0^{\infty} \omega^m G_R(\omega) d\omega, \quad m=0,1,2 \quad (30)$$

Using the modal description of Eq. 29 and following Ref. 3, the spectral moments can be expressed as

$$\lambda_m = \sum_{i=0}^n \sum_{j=0}^n \Psi_i^* \Psi_j^* \lambda_{m,ij}^*, \quad m=0,1,2 \quad (31)$$

where

$$\lambda_{m,ij}^* = \text{Re} \left[\int_0^{\infty} \omega^m G_F(\omega) H_i(\omega) \bar{H}_j(\omega) d\omega \right], \quad m=0,1,2 \quad (32)$$

are defined as cross-spectral moments of normal coordinates for modes i and j of the combined system. As in Ref. 3, it is useful to introduce coefficients $\rho_{m,ij}$ as

$$\rho_{m,ij} = \frac{\lambda_{m,ij}^*}{\sqrt{\lambda_{m,ii}^* \lambda_{m,jj}^*}}, \quad m=0,1,2 \quad (33)$$

(It is noted that $\rho_{0,ij}$ and $\rho_{2,ij}$ are correlation coefficients between the normal coordinates for modes i and j of the combined system and between their time derivatives, respectively; see Ref. 3.) Introducing these relations in Eq. 31, spectral moments of response can be obtained in

Ref. 3.) Introducing these relations in Eq. 31, spectral moments of response can be obtained in terms of spectral moments of individual normal coordinates as

$$\lambda_m = \sum_i \sum_j \Psi_i^* \Psi_j^* \rho_{m,ij} \sqrt{\lambda_{m,ii}^* \lambda_{m,jj}^*}, \quad m=0,1,2 \quad (34)$$

Closed form expressions for $\lambda_{m,ij}^*$ and $\rho_{m,ij}$ for response to white-noise and filtered white-noise inputs have been given in Ref. 3. For completeness of this presentation, the results for white-noise input are given in Appendix I. In the case of wide-band excitations, $\rho_{m,ij}$ rapidly diminish as the spacing between the frequencies of the two modes i and j grows, especially when damping is small. As an example, $\rho_{m,ij}$ for response to white-noise input are illustrated in Fig. 4. Because of this behavior, cross-modal terms in Eqs. 31 and 34 need only be retained for modes with closely spaced frequencies. It should be noted from Fig. 4 that in the case of closely spaced modes the coefficients $\rho_{m,ij}$ critically depend on the modal frequencies and damping ratios. Therefore, in the case of tuned equipment when closely spaced modes occur in the combined system, the accurate estimation of modal frequencies and damping ratios is essential, as was alluded to before.

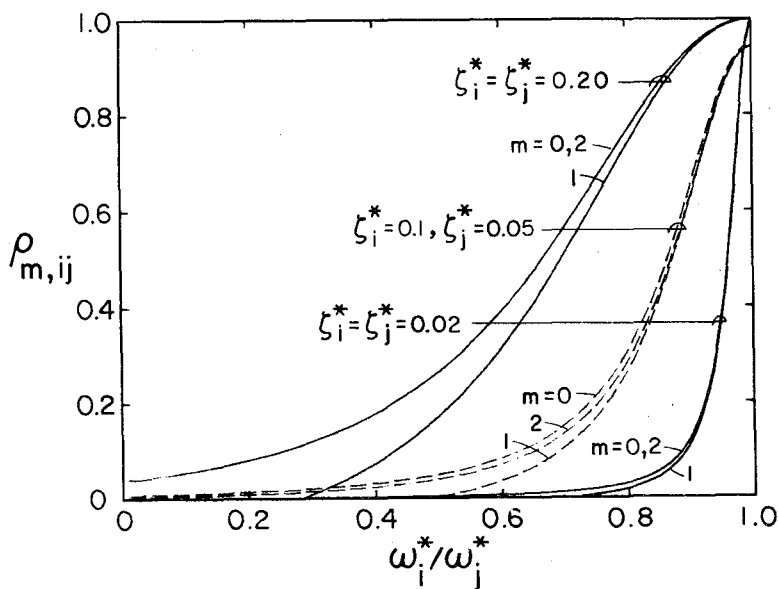


Figure 4. Modal Cross-Correlation Coefficients for Response to White-Noise Input

In terms of the spectral moments, the equipment response quantities are the root-mean-square response, $\sigma_R = \sqrt{\lambda_0}$, and the root-mean-square of response rate, $\sigma_{\dot{R}} = \sqrt{\lambda_2}$. In addition, when $F(t)$ and, hence, $R(t)$ are Gaussian, the mean zero-crossing rate of the response process is obtained as $\nu = \bar{\omega}/\pi = \sigma_{\dot{R}}/\pi\sigma_R = \sqrt{\lambda_2/\lambda_0}/\pi$, where $\bar{\omega}$ is the response mean frequency. Furthermore, in this case, the distribution as well as the mean and variance of the peak response over a specified duration can be expressed in terms of the spectral moments. Specifically, for the maximum absolute response over duration τ , denoted by $R_\tau = \max_{\tau} |R(t)|$, the cumulative distribution as given in Ref. 11 is

$$F_{R_\tau}(r) = \left[1 - \exp(-a^2/2) \right] \exp \left[-\nu\tau \frac{1 - \exp(-\sqrt{\pi/2} \delta_e a)}{\exp(a^2/2) - 1} \right], \quad r > 0 \quad (35)$$

where $a = r/\sigma_R$, $\delta_e = \delta^{1.2}$, and $\delta = \sqrt{1 - \lambda_1^2/\lambda_0\lambda_2}$. The parameter δ has a value between zero and one and has been shown to be a measure of dispersion of the shape of the power spectral density about its centroid (10). For this reason, it will be referred to herein as the shape factor. The mean and standard deviation associated with the above distribution can be obtained as $\bar{R}_\tau = p\sigma_R$ and $\sigma_{R_\tau} = q\sigma_R$, respectively, where p and q are peak factors. For $10 \leq \nu\tau \leq 1000$, which is the usual range of interest in earthquake engineering, approximate expressions for p and q based on results in Ref. 3 are

$$p = \sqrt{2 \ln \nu_e \tau} + \frac{0.5772}{\sqrt{2 \ln \nu_e \tau}} \quad (36)$$

$$q = \frac{1.2}{\sqrt{2 \ln \nu_e \tau}} - \frac{5.4}{13 + (2 \ln \nu_e \tau)^{3.2}} \quad (37)$$

in which

$$\nu_e \tau = \begin{cases} \max(2.1, 2\delta\nu\tau), & 0 < \delta \leq 0.1 \\ (1.63\delta^{0.45} - 0.38)\nu\tau, & 0.1 < \delta < 0.69 \\ \nu\tau, & 0.69 \leq \delta < 1 \end{cases} \quad (38)$$

is a reduced mean crossing rate signifying an effective portion of zero-crossings which may be considered as statistically independent.

It is interesting to examine the case where the equipment is perfectly tuned to the l -th structure mode with the other modes having frequencies well spaced from the tuning fre-

quency. In this case, the cross terms in Eq. 34 can all be neglected except those between modes l and 0. Using Eqs. 28, the spectral moments then become

$$\begin{aligned} \lambda_m \approx & \sum_{i \neq l} \frac{\Psi_i^2}{\beta_i} \lambda_{m,ii} + \left[-\frac{\Psi_l}{2\sqrt{\gamma_l}} + \frac{1}{2} \left(\sum_{i \neq l} \frac{\Psi_i}{\beta_i} + r_{n+1}^* \right) \right]^2 \lambda_{m,ll}^* \\ & + \left[+\frac{\Psi_l}{2\sqrt{\gamma_l}} + \frac{1}{2} \left(\sum_{i \neq l} \frac{\Psi_i}{\beta_i} + r_{n+1}^* \right) \right]^2 \lambda_{m,00}^* \\ & + 2\rho_{m,10} \left[-\frac{\Psi_l^2}{4\gamma_l} + \frac{1}{4} \left(\sum_{i \neq l} \frac{\Psi_i}{\beta_i} + r_{n+1}^* \right)^2 \right] \sqrt{\lambda_{m,ll}^* \lambda_{m,00}^*}, \quad m=0,1,2 \end{aligned} \quad (39)$$

where $\lambda_{m,ii}$ are the spectral moments for the structure alone. For wide-band inputs, on the basis of response to white noise (see Appendix I), it is reasonable to assume that $\lambda_{m,ii}$ is proportional to ω_i^{-m} . Thus, for small γ_l , noting that for perfect tuning $\zeta_l^* \approx \zeta_0^* \approx (\zeta_l + \zeta_e)/2$, and using the approximate expressions for frequencies given in Eqs. 21, the spectral moments for modes l and 0 become, after some algebraic manipulation

$$\lambda_{m,ll}^* \approx \left(1 - \frac{3-m}{2} \sqrt{\gamma_l} \right) \lambda_{m,aa}, \quad m=0,1,2 \quad (40)$$

$$\lambda_{m,00}^* \approx \left(1 + \frac{3-m}{2} \sqrt{\gamma_l} \right) \lambda_{m,aa}, \quad m=0,1,2 \quad (41)$$

where $\lambda_{m,aa}$ are spectral moments evaluated at frequency $\omega_a = (\omega_l + \omega_e)/2 = \omega_l = \omega_e$, and damping coefficient $\zeta_a = (\zeta_l + \zeta_e)/2$. In this case (i.e. perfect tuning and a wide-band input), approximate expressions for the coefficients $\rho_{m,10}$ can be developed from the results given in Appendix I as

$$\rho_{0,10} \approx \rho_{2,10} \approx \frac{4\zeta_a^2}{\gamma_l + 4\zeta_a^2} \quad (42)$$

$$\rho_{1,10} \approx \rho_{0,10} \left[1 - \frac{\gamma_l}{2\pi\zeta_a} \right] \quad (43)$$

Using Eqs. 40-43 in Eqs. 39, spectral moments of the response reduce to

$$\begin{aligned} \lambda_m \approx & \sum_{i \neq l} \left(\frac{\Psi_i}{\beta_i} \right)^2 \lambda_{m,ii} + \left\{ \left[\frac{1}{2(\gamma_l + 4\zeta_a^2)} + \frac{\epsilon_m}{4\pi\zeta_a} \right] \Psi_l^2 \right. \\ & + \left[1 - \frac{\gamma_l}{2(\gamma_l + 4\zeta_a^2)} - \frac{\epsilon_m \gamma_l}{4\pi\zeta_a} \right] \left[\sum_{i \neq l} \frac{\Psi_i}{\beta_i} + r_{n+1}^* \right]^2 \\ & \left. + \frac{3-m}{2} \left[\sum_{i \neq l} \frac{\Psi_i}{\beta_i} + r_{n+1}^* \right] \Psi_l \right\} \lambda_{m,aa}, \quad m=0,1,2 \end{aligned} \quad (44)$$

where $\epsilon_0 = \epsilon_2 = 0$, and $\epsilon_1 = 1$.

A careful examination of the preceding expression is helpful in understanding the relative significance of the various terms. Clearly, the first term inside the braces is the most important because of its small denominator. It should be sufficient in many cases to retain only this term as a simple first approximation. However, under certain circumstances, other terms in the expression may also become significant. For example, when the effective participation factor of the tuned structure mode, Ψ_i , is much smaller than those of some of the other structure modes (which will occur if the equipment is attached to the structure at or near a node of the tuned mode, or if it is tuned to a very high structure mode), then contributions from the other modes may dominate. Also, when the input excitation is poor in frequencies around the tuning mode but rich in frequencies around some of the nontuning modes, then $\lambda_{m,aa}$ may be sufficiently small compared to some of the $\lambda_{m,ii}$ such that the entire term it multiplies can be neglected. The second most important term is the first summation expression on the right-hand side. Note that in that expression contributions from modes much higher than the tuning mode can be neglected, because not only are Ψ_i small but also β_i are large for such modes. The last two terms inside the braces would generally be insignificant because of cancellations in the summations.

The result in Eq. 44 can be compared with an exact expression given in Ref. 1 for a single-degree-of-freedom structure for $m=0$. After retaining the dominant terms in the exact result, Eq. 44 differs only in the presence of $\zeta_g = \sqrt{\zeta_l \zeta_e}$, the geometric mean, in place of ζ_a , the arithmetic mean, in the denominator of the dominant term on the right-hand side. This difference is a consequence of assuming modal damping for the combined equipment-structure system. Note that for all but extreme cases, the difference between ζ_g and ζ_a is small.

Another special case of interest is that of gross detuning, i.e. when the structure frequencies are well spaced from the equipment frequency. If, in addition, the structure frequencies are also well spaced, then all cross terms in Eqs. 34 can be neglected. Using the first and last of Eqs. 20 together with the general expressions for the mode shapes specialized for the case of gross detuning, the formulae for spectral moments then simplify to

$$\lambda_m = \sum_i \left(\frac{\Psi_i}{\beta_i} \right)^2 \lambda_{m,ii} + \left(\sum_i \frac{\Psi_i}{\beta_i} + r_{n+1}^* \right)^2 \lambda_{m,ee}, \quad m=0,1,2 \quad (45)$$

where $\lambda_{m,ee}$ are spectral moments evaluated at the equipment frequency and damping. It is observed that the major contributions to the equipment response in this case come from low structure modes, which have larger Ψ_i values, and from structure modes with frequencies closest to the equipment frequency, which have smaller β_i values.

RESPONSE TO STOCHASTIC INPUT: RESPONSE SPECTRUM APPROACH

In many engineering applications, characterization of the input excitation through the power spectral density function is not convenient. A widely used alternative, which is of special interest in earthquake engineering, is a description in terms of a response spectrum. Under certain conditions, based on the developments in the preceding section, it is possible to construct a rational method whereby the peak response of a system over a specified duration can be obtained in terms of the response spectrum of the input excitation. These conditions are (2): (a) that the input excitation be wide-banded, i.e. it have a smoothly varying power spectral density over a wide range of frequencies covering the significant modes of vibration of the combined system, (b) that the input excitation be a stationary Gaussian process, and (c) that the response of the system over the duration of interest be stationary. In practice, the latter two requirements can be considerably relaxed with little loss of accuracy.

The response spectrum used in this study is defined as a function, $\bar{S}_\tau(\omega, \zeta)$, representing the mean peak response over a duration τ of an oscillator of frequency ω and damping coefficient ζ to a prescribed ensemble of excitations. This definition is consistent with that of the "smooth" response spectrum commonly employed in earthquake engineering, with an additional refinement here to account for the duration of excitation.

To compute the spectral moments of the response in this approach, it is necessary to interpret the spectral moments of the normal coordinates, $\lambda_{m,ii}^*$, in Eqs. 34 in terms of the response spectrum. For a wide-band input, it is reasonable to assume (see Ref. 2) that the

mean zero-crossing rate and the shape factor associated with the i -th normal coordinate, denoted as ν_i and δ_i , respectively, can be approximated by their values for the response of a single-degree-of-freedom oscillator to white-noise input, i.e., by

$$\nu_i = \frac{\omega_i^*}{\pi} \quad (46)$$

and

$$\delta_i = \left[1 - \frac{1}{1-\zeta_i^2} \left(1 - \frac{2}{\pi} \tan^{-1} \frac{\zeta_i^*}{\sqrt{1-\zeta_i^2}} \right) \right]^{2^{1/2}} \approx 2 \left(\frac{\zeta_i^*}{\pi} \right)^{1/2} \quad (47)$$

where the approximation in Eq. 47 is valid for small damping. Noting that $\sqrt{\lambda_{0,ii}^*}$ is the root-mean-square of the i -th normal coordinate, and using Eqs. 36 and 38 to compute the associated peak factor p_i in terms of τ , ν_i , and δ_i , $\lambda_{0,ii}^*$ can be obtained as

$$\lambda_{0,ii}^* = \left[\frac{1}{p_i} \bar{S}_\tau(\omega_i^*, \zeta_i^*) \right]^2 \quad (48)$$

Using the relation $\nu_i = \sqrt{\lambda_{2,ii}^* / \lambda_{0,ii}^*} / \pi$ together with Eq. 46, this gives

$$\lambda_{2,ii}^* = \omega_i^{*2} \lambda_{0,ii}^* = \omega_i^{*2} \left[\frac{1}{p_i} \bar{S}_\tau(\omega_i^*, \zeta_i^*) \right]^2 \quad (49)$$

Finally, using the relation $\delta_i = \sqrt{1 - \lambda_{1,ii}^{*2} / \lambda_{0,ii}^* \lambda_{2,ii}^*}$ and Eq. 47, $\lambda_{1,ii}^*$ becomes

$$\lambda_{1,ii}^* = \omega_i^* \left(1 - \frac{4\zeta_i^*}{\pi} \right)^{1/2} \lambda_{0,ii}^* = \omega_i^* \left(1 - \frac{4\zeta_i^*}{\pi} \right)^{1/2} \left[\frac{1}{p_i} \bar{S}_\tau(\omega_i^*, \zeta_i^*) \right]^2 \quad (50)$$

With the three spectral moments determined, the various quantities of response can be evaluated as previously demonstrated. Specifically, using Eqs. 34, the parameters $\nu = \sqrt{\lambda_2 / \lambda_0} / \pi$ and $\delta = \sqrt{1 - \lambda_1^2 / \lambda_0 \lambda_2}$ are first determined. For this calculation, since the input is wide band, it is reasonable to use expressions of $\rho_{m,ij}$ which are based on white-noise input (see Appendix I and Ref. 2). These parameters, together with τ and $\sigma_R = \sqrt{\lambda_0}$, are then used to evaluate the distribution of the peak response from Eq. 35 and the mean and standard deviation through the peak factors p and q given in Eqs. 36-38.

It is useful to further examine the expression for the mean of the peak response. From

the relation $\bar{R}_\tau = p\sqrt{\lambda_0}$, one obtains

$$\bar{R}_\tau = \left(\sum_{i=0}^n \sum_{j=0}^n \frac{p^2}{p_i p_j} \rho_{0,ij} \bar{R}_{i\tau} \bar{R}_{j\tau} \right)^{1/2} \quad (51)$$

where $\bar{R}_{i\tau} = \Psi_i^* \bar{S}_\tau(\omega_i, \zeta_i)$ is the mean value of the peak contribution from mode i . Note that because of the presence of p , the above relation implicitly depends on τ , λ_1 and λ_2 , in addition to λ_0 . It is easy to show (2) that the ratios p/p_i are usually close to unity and slowly decrease with increasing mode number. When the frequencies of modes making dominant contributions to the response are not very widely spaced, and when the response itself is not extremely narrow banded, for all practical purposes these ratios can be replaced by unity. In that case, Eq. 51 reduces to

$$\bar{R}_\tau = \left(\sum_{i=0}^n \sum_{j=0}^n \rho_{0,ij} \bar{R}_{i\tau} \bar{R}_{j\tau} \right)^{1/2} \quad (52)$$

In the special case of perfect tuning to the fundamental mode with very small mass ratio and light damping, the equipment response can be extremely narrow banded, e.g., $\delta < 0.1$. In that case the peak factor for the response process, p , can be considerably smaller than the peak factor for the fundamental mode, p_1 . Numerical tests have shown that for $\gamma_1 = 0.001$, $\zeta_1 = 0.05$, and $\zeta_e = 0.02$, the ratio p/p_1 can be as small as 0.7. It should be clear that in such cases Eq. 52 would tend to overestimate the equipment response. In spite of its possible conservatism, Eq. 52 can be very useful in practice since it gives the mean of the peak response directly in terms of the response spectrum ordinates through the modal contributions, $\bar{R}_{i\tau}$, without the necessity of computing the spectral moments, λ_m . Furthermore, this result is independent of the duration τ , except for the dependence implied through the specified input response spectrum. Finally, it is noted that cross terms in Eqs. 51 and 52 can be neglected if the frequencies of the combined system are well spaced, since $\rho_{0,ij}$ vanish for such modes. In this case, Eq. 52 reduces to the well known square-root-of-sum-of-squares (SRSS) method for modal combination.

It is important to realize that in most equipment-structure systems closely spaced modes do occur. Thus, the use of the SRSS method for equipment-structure systems is in general inappropriate and should be avoided. In particular, for tuned and nearly tuned equipment, the effective participation factors Ψ_j^* and Ψ_0^* , associated with the resulting two closely spaced modes, are large numbers of opposite sign and nearly equal magnitude. Because of this, the cross terms associated with these modes are negative and of the same magnitude as the squared terms associated with the individual modes. Hence, it is clear that the SRSS rule, which neglects the cross terms, has the potential for severely overestimating the equipment response. This has been demonstrated in several numerical studies which have indicated that the overestimation can be as much as a factor of 10 or more (4). For these reasons, general use of the SRSS method for equipment-structure systems is inappropriate.

In a manner similar to that used for determining the mean of the peak response, employing the relation $\sigma_{R_\tau} = q\sigma_R$, one obtains for the standard deviation of the peak response

$$\sigma_{R_\tau} = \frac{q}{p} \bar{R}_\tau = \left(\sum_{i=0}^n \sum_{j=0}^n \frac{q^2}{p_i p_j} \rho_{0,ij} \bar{R}_{i\tau} \bar{R}_{j\tau} \right)^{1/2} \quad (53)$$

The ratios q/p_i in this expression can not be set equal to unity and, hence, all three spectral moments must be computed to evaluate the standard deviation.

A useful parameter characterizing the response is the response mean frequency, $\bar{\omega} = \sqrt{\lambda_2/\lambda_0}$. Using Eqs. 48 and 49 in Eqs. 34,

$$\bar{\omega} = \left(\frac{\sum_{i=0}^n \sum_{j=0}^n \frac{p^2}{p_i p_j} \rho_{2,ij} \bar{R}_{i\tau} \bar{R}_{j\tau} \omega_i^* \omega_j^*}{\sum_{i=0}^n \sum_{j=0}^n \frac{p^2}{p_i p_j} \rho_{0,ij} \bar{R}_{i\tau} \bar{R}_{j\tau}} \right)^{1/2} \quad (54)$$

where, for convenience, the numerator and the denominator in the expression have been multiplied by p . As noted above, in many instances, the ratios p/p_i are near unity; then, the preceding simplifies to

$$\bar{\omega} = \left(\frac{\sum_{i=0}^n \sum_{j=0}^n \rho_{2,ij} \bar{R}_{i\tau} \bar{R}_{j\tau} \omega_i^* \omega_j^*}{\sum_{i=0}^n \sum_{j=0}^n \rho_{0,ij} \bar{R}_{i\tau} \bar{R}_{j\tau}} \right)^{1/2} \quad (55)$$

Observe that in this case the response mean frequency is the average of the modal frequencies of the equipment-structure system as weighted by the modal contributions to the response. This response parameter should be useful in studies concerned with equipment fatigue.

It is observed from Eqs. 51-55 that the equipment response quantities are obtained directly in terms of the ordinates of the response spectrum describing the input into the combined system. Note that the other quantities appearing in these equations are given in terms of the dynamic properties of the structure alone and of the equipment alone. It is important to point out that these results are computationally simple and far more comprehensive than any of the current response spectrum methods for equipment response. Furthermore, these results are superior to those currently available because they are based on a more realistic model of the combined system, since they include the equipment-structure interaction, and because they properly account for the correlation between modal responses. Some of these aspects have invariably been neglected in the literature on the equipment-structure problem.

It is useful to examine the mean of the peak response for the special cases discussed in the previous section in terms of the response spectrum formulation. For simplicity, the ratios p/p_i are discarded in this study. In the special case of perfect tuning to structure mode l , using Eqs. 44 for $m = 0$, the mean peak response is

$$\begin{aligned} \bar{R}_\tau = & \left\{ \sum_{i \neq l} \left(\frac{\Psi_i}{\beta_i} \right)^2 \bar{S}_\tau^2(\omega_i, \zeta_i) + \left[\frac{\Psi_l}{\sqrt{2(\gamma_l + 4\zeta_a^2)}} \right]^2 \right. \\ & \left. + \left[1 - \frac{\gamma_l}{2(\gamma_l + 4\zeta_a^2)} \right] \left(\sum_{i \neq l} \frac{\Psi_i}{\beta_i} + r_{n+1}^* \right)^2 + \frac{\Psi_l}{2/3} \left(\sum_{i \neq l} \frac{\Psi_i}{\beta_i} + r_{n+1}^* \right) \right\} \bar{S}_\tau^2(\omega_a, \zeta_a)^{1/2} \quad (56) \end{aligned}$$

where it is recalled that $\omega_a = (\omega_l + \omega_e)/2 = \omega_l = \omega_e$ and $\zeta_a = (\zeta_l + \zeta_e)/2$. Based on the comparison of Eq. 44 with the exact results for a single-degree-of-freedom structure (1), it is conjectured that the more appropriate expression for perfect tuning that would account for nonmo-

dal damping of the combined equipment-structure system would be obtained by replacing ζ_a by $\zeta_g = \sqrt{\zeta_l \zeta_e}$ in the denominators on the right-hand side of Eq. 56.

For the case of gross detuning, the corresponding result is

$$\bar{R}_\tau = \left\{ \sum_i \left(\frac{\Psi_i}{\beta_i} \right)^2 \bar{S}_\tau^2(\omega_i, \zeta_i) + \left(\sum_i \frac{\Psi_i}{\beta_i} + r_{n+1}^* \right)^2 \bar{S}_\tau^2(\omega_e, \zeta_e) \right\}^{1/2} \quad (57)$$

The preceding relation for the case of gross detuning is equivalent to results obtained in Refs. 6 and 7. However, to the writers' knowledge, the result in Eq. 56 for the case of perfect tuning has no equivalent in the literature.

It should be pointed out that although the results developed here correspond to the expected value of the responses to an ensemble of input excitations, the simplified expression for the mean peak response in Eq. 52 should also provide a good approximation for the peak response to an individual deterministic input because the dispersion in the peak response is generally small (i.e. 0.1-0.3 coefficient of variation, depending on the response frequency). Expected errors associated with such an approximation are anticipated to range at most between 10-40 percent.

As indicated in the introduction, the conventional floor spectrum associated with the k -th degree of freedom (i.e., the k -th floor), which is equivalent to the peak equipment response ignoring interaction, can easily be generated with the present method. For this purpose, it is sufficient to set all $\gamma_i = 0$ in Eq. 51 or 52 to obtain the ordinate of the mean floor spectrum for selected values of ω_e and ζ_e . At tuning, when $\omega_e = \omega_l$, Eqs 51 and 52 assume an indefinite form because in that case $\Psi_0^*/\Psi_l = -\Psi_l^*/\Psi_l = \infty$. The proper limit for this special case is obtained from Eq. 56 by letting $\gamma_l = 0$

$$\bar{R}_\tau = \left\{ \sum_{i \neq l} \left(\frac{\Psi_i}{\beta_i} \right)^2 \bar{S}_\tau^2(\omega_i, \zeta_i) + \left[\left(\frac{\Psi_l}{2\sqrt{2}\zeta_a} \right)^2 \right] \right\}$$

$$+ \left\{ \left(\sum_{i \neq l} \frac{\Psi_i}{\beta_i} + r_{n+1}^* \right)^2 + \frac{\Psi_l}{2/3} \left(\sum_{i \neq l} \frac{\Psi_i}{\beta_i} + r_{n+1}^* \right) \right\} \bar{S}_r^2(\omega_a, \zeta_a) \}^{1/2} \quad (58)$$

As was indicated after Eq. 56, it would be appropriate to replace ζ_a appearing in the denominator of the right-hand side of the above equation by ζ_g to account for the effect of nonmodal damping. In the case of gross detuning, Eq. 57 is the appropriate expression for the floor spectrum under all conditions. This equation is equivalent to floor spectrum results for gross detuning in Refs. 6 and 7. However, Eq. 58 for perfect tuning is new.

It should be observed that with the present method the conventional floor spectrum is given directly in terms of the modal properties of the structure and the mean response spectrum associated with an ensemble of input excitations. This clearly is far more efficient than the traditional method of generating floor spectra which would involve a multitude of time history computations for any selected values of the frequency and the damping coefficient of the floor spectrum.

In light of the results developed in this study, there is in principle no point in using the conventional floor spectrum method for a single-degree-of-freedom equipment since it ignores equipment-structure interaction. Nevertheless, in certain practical applications where it is desired to develop equipment design criteria which is independent of an explicit description of the structure and the input excitation (e.g., when two different groups perform the structural design and the equipment evaluation), the floor spectrum method may still be useful. For this purpose, using the present method, floor spectra including the effect of interaction can be generated. Thus, for a typical floor, a family of spectra corresponding to a sequence of equipment masses can be developed. The equipment designer would then employ that spectrum which corresponds to the mass of the equipment under consideration. The floor spectrum might also be useful for a light multi-degree-of-freedom equipment item that is attached to a single floor. When the interaction can be neglected, the equipment response can be obtained by modal superposition using the conventional floor spectrum as the base input in the usual way. When interaction is significant, it might also be possible to determine the equipment response using

the family of spectra previously described. However, at this time it is not apparent as to how to assign to each mode of the equipment the appropriate spectrum from the family. Furthermore, it is not clear how accurate the response spectrum method will be for a multi-degree-of-freedom equipment, since the floor response is generally not a wide-band process, as was assumed in the development of the response spectrum method. These matters require further research.

NUMERICAL EXAMPLES

As an example structure, a 10-story uniform shear building is considered. Mass, stiffness, and modal damping coefficients are shown in Fig. 5. Two alternative attachment positions of the equipment are studied, i.e., one on the fifth floor and one on the tenth floor. The modal frequencies and the elements of the modal vectors, normalized to give unit modal masses, for the two attachment points are shown in Table 1. These values are used to determine the

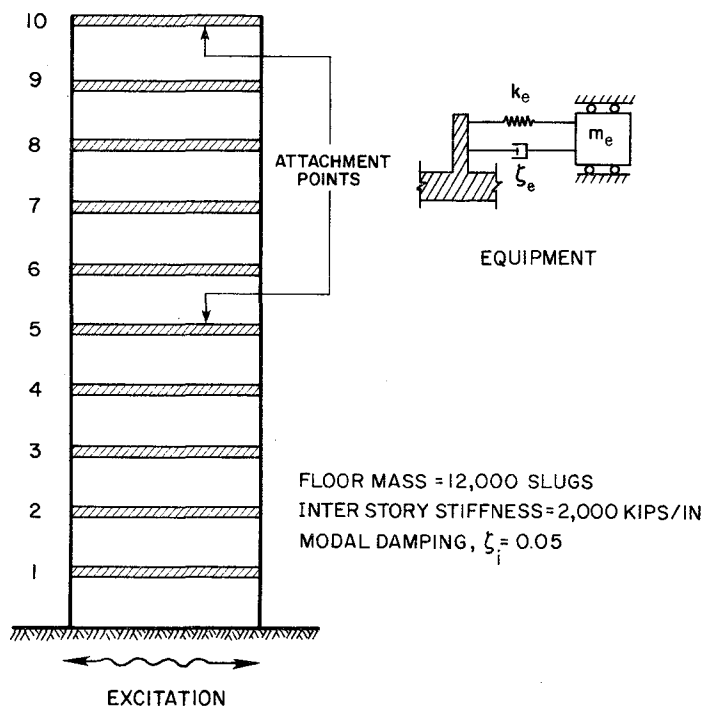


Figure 5. Example Structure and Equipment

effective mass ratios, γ_i , for each of the structure modes for any given equipment mass. In this study, three values of equipment mass were chosen so as to produce an effective mass ratio based on the first mode of $\gamma_1 = 0.001, 0.01, \text{ and } 0.05$. These correspond to equipment masses of $m_e = 63.4, 634, 3170$ slugs, respectively, and to equipment to floor mass ratios of $0.005, 0.053, \text{ and } 0.264$, respectively.

Table 1. Structure Natural Frequencies and Components of Mode Shapes for Attachment Degrees of Freedom

Mode i	Freq., rad/sec ω_i	Elements of Mode Shapes	
		ϕ_{5i}	ϕ_{10i}
1	6.684	0.297	0.435
2	19.903	-0.341	0.425
3	32.677	0.246	-0.406
4	44.721	-0.378	-0.378
5	55.767	-0.189	-0.341
6	65.566	-0.406	0.297
7	73.901	0.128	-0.246
8	80.585	0.425	0.189
9	85.469	-0.065	-0.129
10	88.444	-0.435	0.065

In order to examine the accuracy of the natural frequencies and mode shapes of the equipment-structure system as given by Eqs. 7, 8, 9, 13, 14, and 16, comparisons are made in Tables 2, 3, and 4 with exact results for the cases of perfect tuning to structure modes 1, 2, and 3, respectively. The exact results were obtained by a direct solution of the eigenvalue problem for the combined (11-degree-of-freedom) equipment-structure system. To describe the error in the i -th mode shape, the difference vector $\Delta_i = \Phi_i^{*E} - \Phi_i^*$ was considered where Φ_i^{*E} is the exact mode shape. Rather than listing the elements of this vector, for brevity, only the ratio of its length to that of the modal vector as normalized by the mass matrix, i.e. the ratio $(\Delta_i^T \mathbf{M}^* \Delta_i / \Phi_i^{*T} \mathbf{M}^* \Phi_i^*)^{1/2}$, is listed in Tables 2 to 4 for each of the modes.

A perusal of Tables 2 to 4 demonstrates that the perturbation procedure generates very accurate estimates of eigenproperties of the combined equipment-structure system for light equipment. Even for heavy equipment, e.g., $\gamma_1 = 0.05$ when the equipment mass is more than

a quarter of the floor mass, acceptable accuracy is obtained. Note that errors in both frequencies and mode shapes tend to increase as the equipment of fixed mass is tuned to higher structure modes.

Table 2. Comparison of Natural Frequencies and Mode Shapes of Equipment-Structure System with Exact Values: Equipment Tuned to 1st Structure Mode

Mass Ratio γ_1	Mode	Equipment Attached to 5th Floor			Equipment Attached to 10th Floor		
		Exact Frequency rad/sec	Error in Frequency %	Error in Mode Shape %	Exact Frequency rad/sec	Error in Frequency %	Error in Mode Shape %
0.001	0	6.579	-0.00	0.23	6.580	-0.00	0.16
	1	6.790	0.00	0.23	6.790	0.01	0.16
	2	19.905	0.00	0.02	19.904	0.00	0.02
	3	32.678	0.00	0.00	32.678	0.00	0.00
	4	44.722	0.00	0.00	44.722	0.00	0.00
	5	55.767	0.00	0.00	55.767	0.00	0.00
	6	65.567	0.00	0.00	65.566	0.00	0.00
	7	73.901	0.00	0.00	73.901	0.00	0.00
	8	80.586	0.00	0.00	80.585	0.00	0.00
	9	85.469	0.00	0.00	85.469	0.00	0.00
	10	88.444	0.00	0.00	88.444	0.00	0.00
0.01	0	6.354	-0.09	0.72	6.355	-0.06	0.49
	1	7.021	0.08	0.74	7.023	0.05	0.50
	2	19.920	0.00	0.19	19.915	0.00	0.16
	3	32.682	0.00	0.10	32.683	0.00	0.09
	4	44.730	0.00	0.09	44.725	0.00	0.06
	5	55.768	0.00	0.05	55.769	0.00	0.04
	6	65.573	0.00	0.08	65.568	0.00	0.03
	7	73.902	0.00	0.04	73.902	0.00	0.02
	8	80.591	0.00	0.09	80.586	0.00	0.02
	9	85.469	0.00	0.03	85.469	0.00	0.01
	10	88.449	0.00	0.08	88.444	0.00	0.00
0.05	0	5.959	-0.59	1.65	5.966	-0.40	1.15
	1	7.443	0.40	1.73	7.453	0.27	1.22
	2	19.986	0.00	0.93	19.963	0.00	0.79
	3	32.702	0.00	0.48	32.708	0.00	0.44
	4	44.763	0.00	0.45	44.741	0.00	0.30
	5	55.775	0.00	0.27	55.779	0.00	0.22
	6	65.598	0.00	0.38	65.574	0.00	0.16
	7	73.904	0.00	0.20	73.906	0.00	0.12
	8	80.614	0.00	0.46	80.588	0.00	0.08
	9	85.470	0.00	0.17	85.470	0.00	0.05
	10	88.471	0.00	0.42	88.444	0.00	0.03

Table 3. Comparison of Natural Frequencies and Mode Shapes of Equipment-Structure System with Exact Values: Equipment Tuned to 2nd Structure Mode

Mass Ratio γ_1	Mode	Equipment Attached to 5th Floor			Equipment Attached to 10th Floor		
		Exact Frequency rad/sec	Error in Frequency %	Error in Mode Shape %	Exact Frequency rad/sec	Error in Frequency %	Error in Mode Shape %
0.001	1	6.680	0.00	0.02	6.680	0.00	0.01
	0	19.544	-0.01	0.11	19.599	0.01	0.25
	2	20.267	0.01	0.19	20.214	-0.00	0.17
	3	32.684	0.00	0.13	32.685	0.00	0.12
	4	44.730	0.00	0.10	44.726	0.00	0.06
	5	55.768	0.00	0.05	55.769	0.00	0.04
	6	65.572	0.00	0.07	65.568	0.00	0.03
	7	73.902	0.00	0.04	73.902	0.00	0.02
	8	80.590	0.00	0.09	80.586	0.00	0.02
	9	85.469	0.00	0.03	85.469	0.00	0.00
	10	88.449	0.00	0.08	88.444	0.00	0.00
0.01	1	6.647	0.00	0.18	6.647	0.00	0.15
	0	18.792	-0.11	0.66	18.972	0.04	1.19
	2	21.059	0.10	1.12	20.904	-0.02	0.78
	3	32.743	0.00	1.28	32.761	0.00	1.20
	4	44.811	0.00	0.98	44.763	0.00	0.65
	5	55.783	0.00	0.53	55.792	0.00	0.43
	6	65.628	0.00	0.74	65.582	0.00	0.31
	7	73.906	0.00	0.37	73.910	0.00	0.22
	8	80.639	0.00	0.87	80.590	0.00	0.16
	9	85.470	0.00	0.31	85.471	0.00	0.10
	10	88.495	0.00	0.79	88.444	0.00	0.05
0.05	1	6.503	0.02	1.67	6.503	0.03	0.73
	0	17.585	-1.00	4.49	17.964	0.15	4.17
	2	22.475	0.67	1.64	22.168	0.13	4.05
	3	32.999	0.01	6.17	33.100	-0.04	5.92
	4	45.164	0.00	4.77	44.934	-0.01	3.28
	5	55.849	0.00	2.66	55.895	-0.01	2.20
	6	65.877	0.00	3.70	65.646	0.00	1.57
	7	73.928	0.00	1.86	73.948	0.00	1.13
	8	80.855	-0.00	4.41	80.611	0.00	0.79
	9	85.475	-0.00	1.56	85.480	0.00	0.51
	10	88.713	-0.02	4.03	88.446	0.00	0.25

Table 4. Comparison of Natural Frequencies and Mode Shapes of Equipment-Structure System with Exact Values: Equipment Tuned to 3rd Structure Mode

Mass Ratio γ_1	Mode	Equipment Attached to 5th Floor			Equipment Attached to 10th Floor		
		Exact Frequency rad/sec	Error in Frequency %	Error in Mode Shape %	Exact Frequency rad/sec	Error in Frequency %	Error in Mode Shape %
0.001	1	6.681	0.00	0.02	6.681	0.00	0.01
	2	19.882	0.00	0.23	19.888	0.00	0.01
	0	32.251	-0.01	0.24	32.208	0.02	1.05
	3	33.103	0.01	0.54	33.171	-0.02	0.82
	4	44.763	0.00	0.45	44.741	0.00	0.30
	5	55.773	0.00	0.19	55.776	0.00	0.15
	6	65.586	0.00	0.24	65.571	0.00	0.10
	7	73.903	0.00	0.12	73.904	0.00	0.07
	8	80.601	0.00	0.26	80.586	0.00	0.05
	9	85.469	0.00	0.09	85.470	0.00	0.03
	10	88.459	0.00	0.23	88.444	0.00	0.01
0.01	1	6.650	0.00	0.16	6.650	0.00	0.14
	2	19.699	0.00	2.28	19.755	-0.00	1.94
	0	31.370	-0.21	1.70	31.296	0.26	4.25
	3	33.981	0.23	3.39	34.296	-0.17	2.66
	4	45.126	0.01	4.35	44.917	-0.01	3.01
	5	55.826	0.00	1.91	55.858	-0.00	1.57
	6	65.769	-0.00	2.41	65.617	-0.00	1.03
	7	73.918	0.00	1.16	73.930	-0.00	0.70
	8	80.748	-0.00	2.65	80.600	-0.00	0.48
	9	85.473	0.00	0.93	85.476	-0.00	0.30
	10	88.600	-0.01	2.37	88.445	-0.00	0.15
0.05	1	6.514	0.04	0.82	6.515	0.02	0.68
	2	18.978	0.01	10.19	19.229	-0.08	8.76
	0	30.075	-1.32	7.60	30.038	1.32	12.21
	3	35.231	1.70	12.88	36.310	-0.12	11.25
	4	46.521	0.14	18.58	45.734	-0.21	15.03
	5	56.056	0.01	9.29	56.252	-0.08	8.26
	6	66.557	0.01	11.80	65.839	-0.03	5.40
	7	73.985	0.00	5.74	74.056	-0.02	3.72
	8	81.395	-0.00	13.69	80.666	-0.01	2.54
	9	85.486	0.00	4.64	85.504	-0.00	1.60
	10	89.344	-0.17	12.46	88.452	-0.00	0.79

To examine the accuracy of the response spectrum method for equipment response developed herein, an ensemble of 20 simulated ground acceleration time histories with a mean peak ground acceleration of 0.5g and a strong motion duration of 11 seconds were employed. The bases for generation of these time histories and the computation of their associated mean ground response spectra are discussed in Ref. 2. A typical member of the ensemble and the mean ground response spectra are shown in Figs. 6 and 7, respectively. Using these spectra,

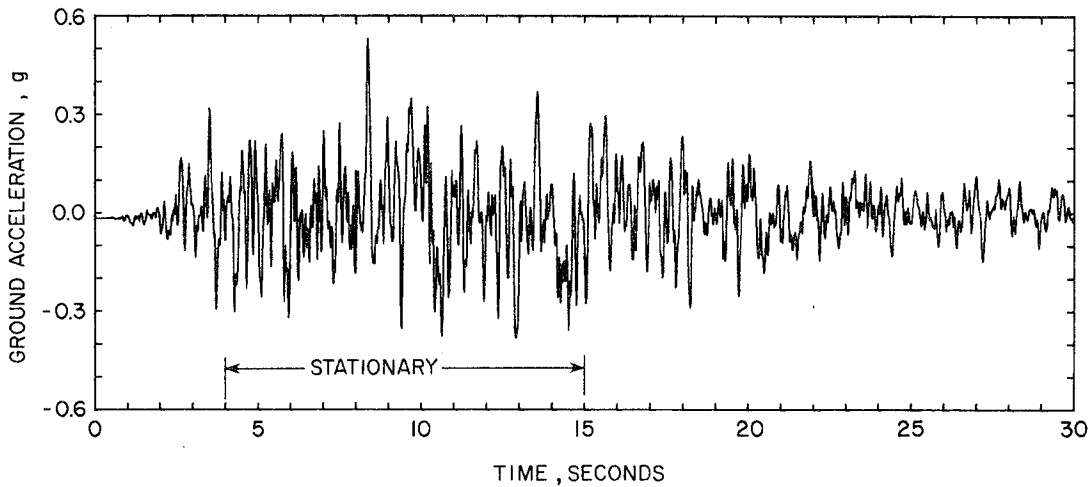


Figure 6. Sample of Simulated Earthquake Ground Motion

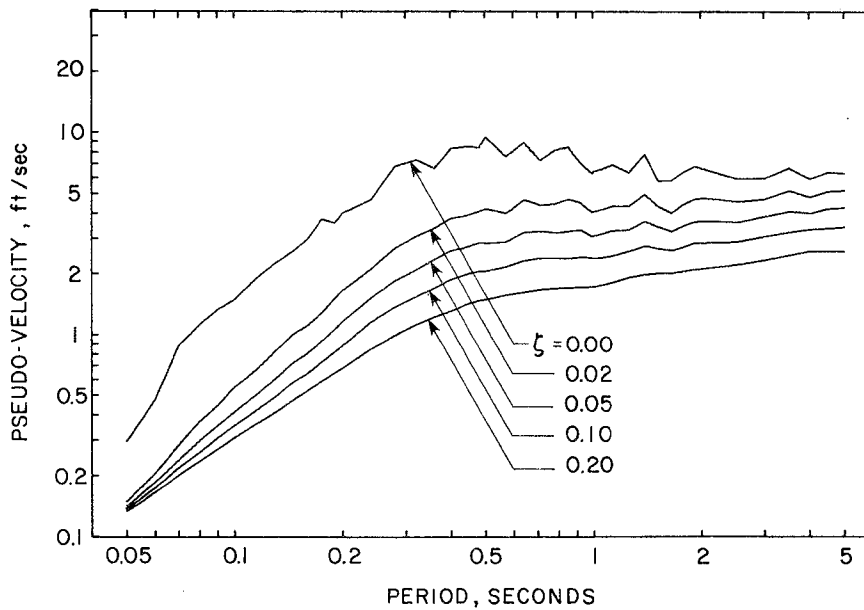


Figure 7. Mean Pseudo-Velocity Spectra for 20 Simulated Earthquakes

statistical measures of the equipment acceleration response (i.e., the root-mean-square, the mean of the peak, and the standard deviation of the peak) were computed for a continuous range of equipment frequencies, for the three values of γ_1 , and for two values of equipment damping, $\zeta_e = 0.02$ (Fig. 8) and $\zeta_e = 0.05$ (Fig. 9). For brevity, the results are presented only

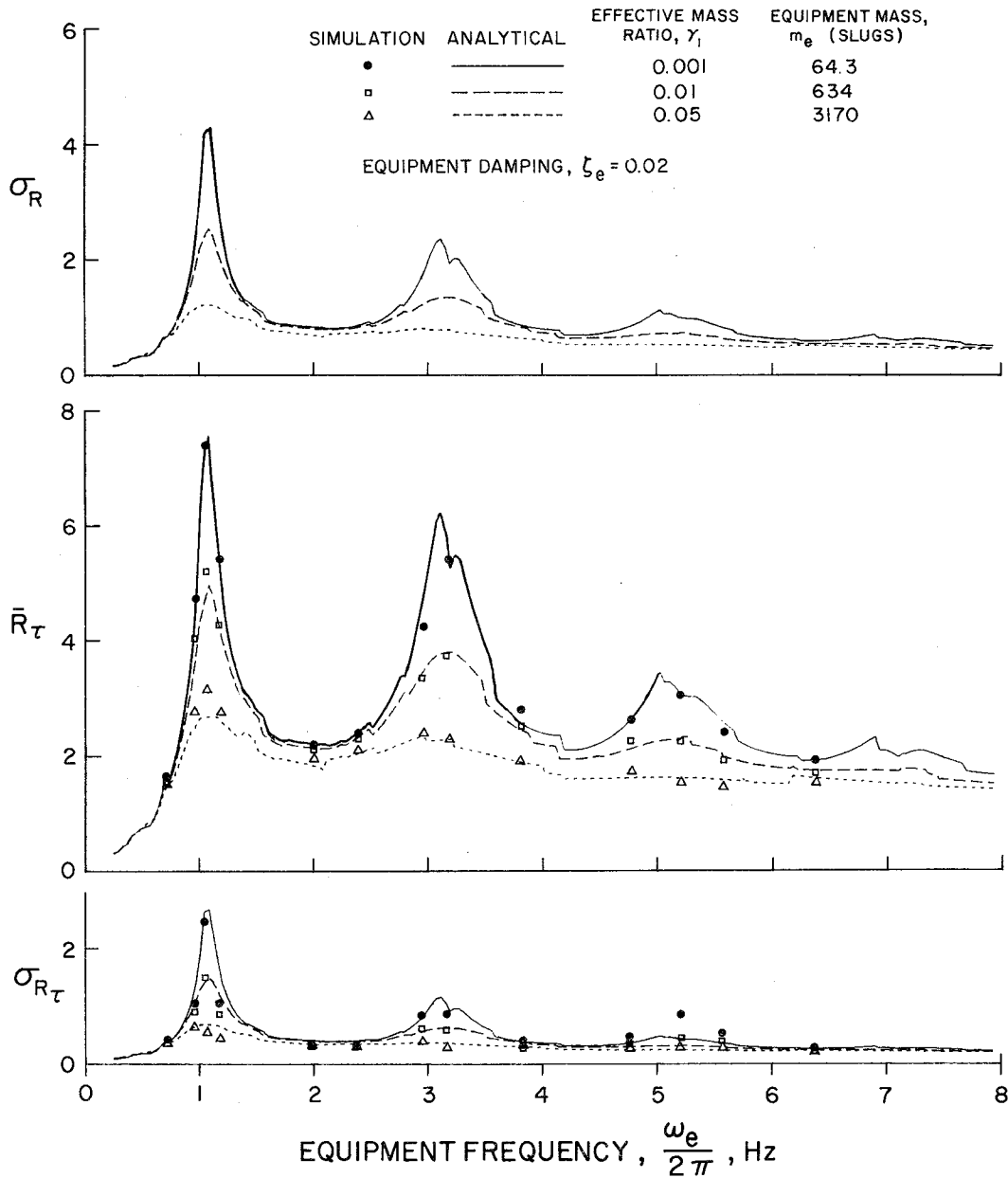


Figure 8. Acceleration Response Quantities for Equipment Attached to 10-th Floor with $\zeta_e = 0.02$: σ_R = root-mean-square; \bar{R}_τ = mean of peak; σ_{R_τ} = standard deviation of peak

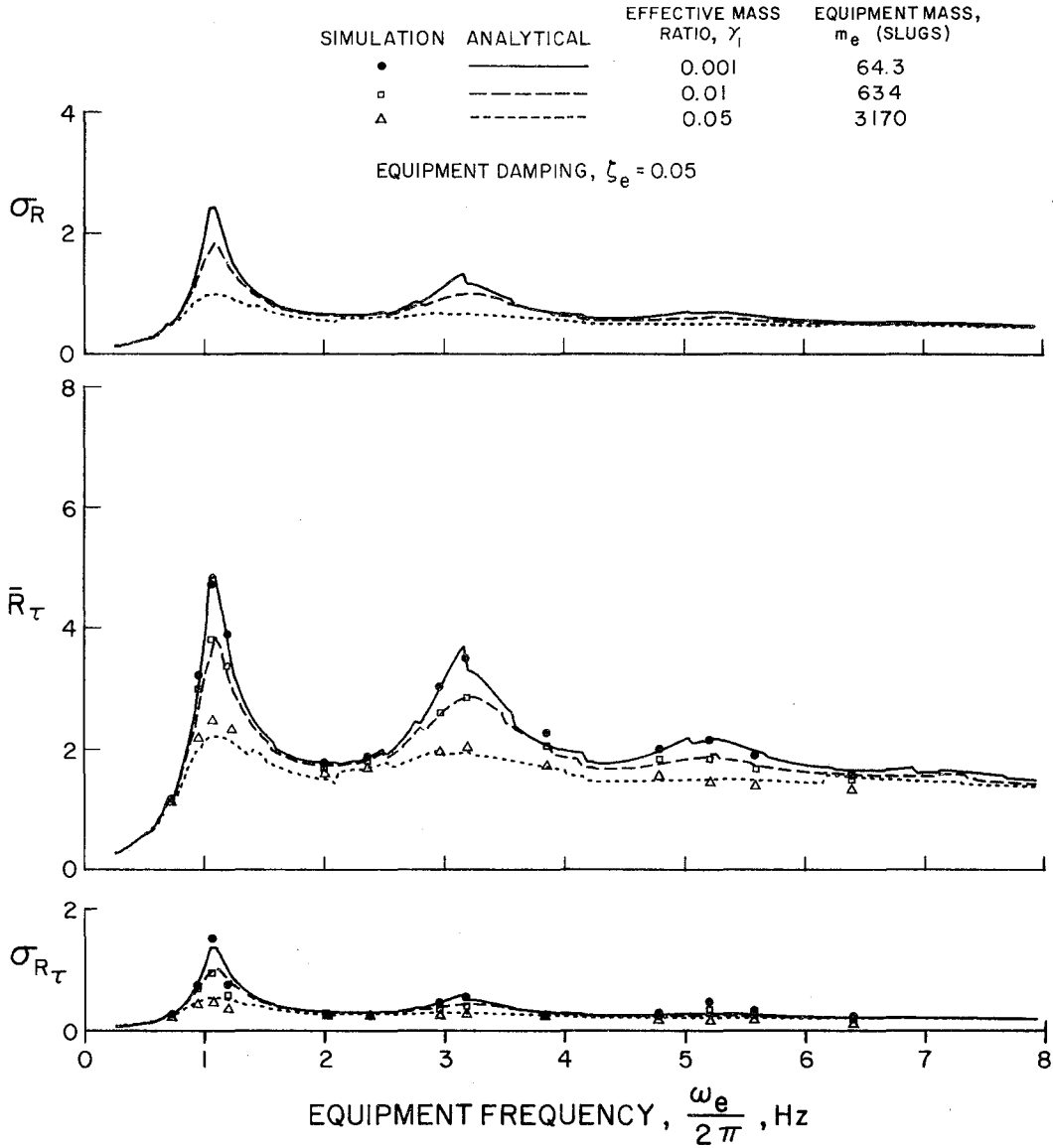


Figure 9. Acceleration Response Quantities for Equipment Attached to 10-th Floor with $\zeta_e = 0.05$: σ_R = root-mean-square; \bar{R}_τ = mean of peak; σ_{R_τ} = standard deviation of peak

for the case of the equipment attached to the 10-th floor of the structure. Results for attachment to the 5-th floor are of the same character. The computation involved determining the modal spectral moments from Eqs. 48-50 which were then used in Eq. 34 to give the spectral moments of the response. The response measures were then computed in terms of these spectral moments as described in the text following Eq. 34. In particular, the mean and standard deviation of the peak response were computed from Eqs. 51 and 53, respectively. It is noted

that the simplified expression for the mean, Eq. 52, gave more or less the same results as Eq. 51 except at tuning to the first mode of the structure when up to 30 percent overestimation was observed for the lightest equipment mass. As indicated before, this is a consequence of the narrow-bandedness of the equipment response process for that case.

To provide a basis for comparison, ensembles of "exact" peak responses were computed for selected values of the equipment frequency by means of time-history analyses of the combined equipment-structure system for the simulated ground motions. A matrix time marching algorithm rather than a modal approach was used in this computation so as not to introduce any error due to nonmodal damping. Means and standard deviations of these ensembles were computed and are shown in Figs. 8 and 9 by various symbols for the three values of γ_1 . The agreement between the analytical and the simulated results is remarkable.

From Figs. 8 and 9 the effect of equipment-structure interaction on the equipment response is seen to be significant. For example, with the equipment damping equal to 0.02, there is about 30 percent reduction in the mean peak equipment response at tuning when γ_1 is increased from 0.001 to 0.01. For the damping values considered, further reduction in the mass ratio below 0.001 did not cause any significant increase in the mean peak response. Thus, the mean peak response curves in Figs. 8 and 9 associated with $\gamma_1=0.001$ ($m_e=63.4$ slugs) essentially represent the conventional floor spectra. The other curves, i.e. those associated with $\gamma_1=0.01$ ($m_e=634$ slugs) and $\gamma_1=0.05$ ($m_e=3170$ slugs) may be considered as floor spectra which include the effect of equipment-structure interaction. These spectra could be used directly by the equipment analyst to design any equipment item having a prescribed mass, frequency, and damping.

Another quantity of the equipment response that would be of interest to the analyst is the response mean frequency, $\bar{\omega}$, which provides an estimate of the number of cycles of oscillation. As indicated before, this quantity would be useful in studies concerning equipment fatigue. Using Eq. 54, this quantity for the acceleration response has been computed in terms of the input response spectrum for the equipment attached to the 10-th floor of the structure and is

plotted in Fig. 10 for two damping values of the equipment. It is seen that the response mean frequency, $\bar{\omega}$, is slightly higher than the equipment frequency, ω_e , when ω_e is less than the fundamental frequency of the structure and tends to be less than ω_e when ω_e is greater than the fundamental frequency of the structure. An interesting observation is that $\bar{\omega}$ tends to

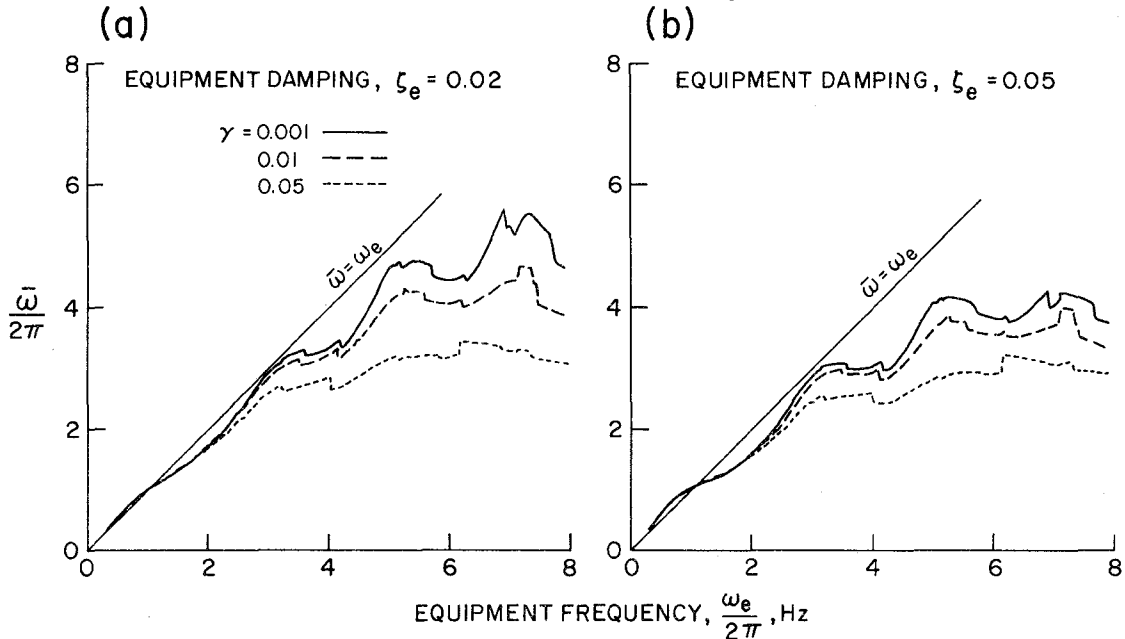


Figure 10. Mean Frequency of Acceleration Response for Equipment Attached to 10-th Floor

approach closer to ω_e when tuning occurs. This is a consequence of the fact that $\bar{\omega}$ is an average of the modal frequencies as weighted by the modal contributions to the response, and at tuning the tuning modes, which have frequencies close to ω_e , make dominant contributions.

SUMMARY AND CONCLUSIONS

A simple and accurate method is developed to estimate the response of light equipment in structures subjected to random excitations, such as earthquake ground motions. The method employs perturbation techniques to determine the dynamic properties of the combined equipment-structure system in terms of those of the structure alone and of the equipment alone. These derived properties are used to determine the equipment response by a modal superposition technique which accounts for cross-correlation between modal responses. The results include the effect of equipment-structure interaction, which is particularly significant

when the equipment is tuned to a natural frequency of the structure. Two characterizations of the input excitation are employed; namely, the power spectral density approach for a stationary input and the response spectrum approach for earthquake-type excitations. Various statistical measures of the equipment response including its root-mean-square and that of its time derivative, and the mean, the standard deviation and the cumulative distribution function of the peak response over a specified duration, are obtained in terms of the dynamic properties of the two subsystems separately and the description of the input excitation. In the case of the response spectrum approach, which is of particular interest in practical applications in earthquake engineering, the equipment response is given in terms of the response spectrum associated with the input ground motion.

The method developed generates as a special case the conventional floor spectrum by simply setting the equipment mass equal to zero, which is equivalent to neglecting equipment-structure interaction. More generally, the method produces floor spectra which include the effect of interaction and are as easily applicable in practice as is the conventional spectra.

A comprehensive numerical study employing a 10-degree-of-freedom example structure is presented. The results demonstrate the accuracy of the methodology in generating the eigen-properties of the combined equipment-structure system and in estimating the response of the equipment. It is found that in a variety of situations, the equipment-structure interaction is significant and must be taken into account.

APPENDIX I.-CROSS-SPECTRAL MOMENTS FOR RESPONSE TO WHITE NOISE

Cross-spectral moments of normal coordinates corresponding to response to a stationary excitation can be expressed as

$$\lambda_{m,ij} = \rho_{m,ij} \sqrt{\lambda_{m,ii} \lambda_{m,jj}}, \quad m = 0,1,2 \quad (59)$$

For the case of response to white noise input, from Ref. 3 or 10

$$\lambda_{0,ii} = \frac{\pi G_0}{4\zeta_i \omega_i^3} \quad (60)$$

$$\lambda_{1,ii} = \frac{\pi G_0}{4\zeta_i \omega_i^2} \frac{1 - \frac{2}{\pi} \tan^{-1}(\zeta_i / \sqrt{1 - \zeta_i^2})}{\sqrt{1 - \zeta_i^2}} \quad (61)$$

$$\lambda_{2,ii} = \frac{\pi G_0}{4\zeta_i \omega_i} \quad (62)$$

where G_0 is the constant amplitude of the power spectral density. Exact expressions for $\rho_{m,ij}$ for this case were given in Ref. 3, which for small damping were shown to reduce to

$$\rho_{0,ij} = \frac{2\sqrt{\zeta_i \zeta_j} \left[(\omega_i + \omega_j)^2 (\zeta_i + \zeta_j) + (\omega_i^2 - \omega_j^2) (\zeta_i - \zeta_j) \right]}{4(\omega_i - \omega_j)^2 + (\zeta_i + \zeta_j)^2 (\omega_i + \omega_j)^2} \quad (63)$$

$$\rho_{1,ij} = \frac{2\sqrt{\zeta_i \zeta_j} \left[(\omega_i + \omega_j)^2 (\zeta_i + \zeta_j) - \frac{4}{\pi} (\omega_i - \omega_j)^2 \right]}{4(\omega_i - \omega_j)^2 + (\zeta_i + \zeta_j)^2 (\omega_i + \omega_j)^2} \quad (64)$$

$$\rho_{2,ij} = \frac{2\sqrt{\zeta_i \zeta_j} \left[(\omega_i + \omega_j)^2 (\zeta_i + \zeta_j) - (\omega_i^2 - \omega_j^2) (\zeta_i - \zeta_j) \right]}{4(\omega_i - \omega_j)^2 + (\zeta_i + \zeta_j)^2 (\omega_i + \omega_j)^2} \quad (65)$$

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APPENDIX III.- NOTATION

The following symbols are used in this report:

$a = r/\sigma_R$	normalized response level;
\mathbf{C}	damping matrix of structure alone;
\mathbf{C}^*	damping matrix of combined system;
c_{ij}	element of \mathbf{C} ;
\mathbf{D}	diagonal matrix with elements M_i ;
$F(t)$	zero-mean stationary input process;
$F_{R_\tau}(r)$	cumulative probability distribution function of R_τ ;
$G_F(\omega)$	one-sided power spectral density of input process;
$G_R(\omega)$	one-sided power spectral density of equipment response;
$H_i(\omega)$	complex frequency-response function for mode i ;
\mathbf{K}	stiffness matrix of structure alone;
\mathbf{K}^*	stiffness matrix of combined system;
k_{ij}	element of \mathbf{K} ;
l	mode number of structure to which equipment is tuned;
\mathbf{M}	mass matrix of structure alone;
\mathbf{M}^*	mass matrix of combined system;
M_i	i -th modal mass of structure alone;
M_i^*	i -th modal mass of combined system;
m_{ij}	element of \mathbf{M} ;
m_e	mass of equipment;
n	number of degrees of freedom of structure;
p, q	peak factors for equipment response;
p_i, q_i	peak factors for response in mode i ;
$R(t)$	equipment response;
$R_i(t)$	contribution of mode i to equipment response;
\overline{R}_τ	peak of $R(t)$ over τ ;
\underline{R}_τ	mean of R_τ ;
$\overline{R}_{i\tau}$	mean of peak contribution of mode i to equipment response;
\mathbf{R}	influence vector for structure alone;
\mathbf{R}^*	influence vector for combined system;
r	measure of modification of frequencies and mode shapes;
r_i	element of \mathbf{R} and \mathbf{R}^* ;
r_{n+1}^*	element of \mathbf{R}^* associated with equipment degree of freedom;
$S_i(t)$	i -th normal coordinate of combined system;
$\overline{S}_\tau(\omega, \zeta)$	response spectrum ordinate at frequency ω and damping ζ ;
α_i	modal amplification factor relative to attachment point for mode i ;
β_i	detuning parameter associated with mode i ;
Γ_i	participation factor associated with i -th mode of structure;
Γ_i^*	participation factor associated with i -th mode of combined system;
γ_i	effective mass ratio associated with mode i ;
Δ_i	difference vector associated with mode i ;
δ	power spectral density shape factor;
δ_e	an effective value of δ ;
δ_i	shape factor for response in mode i ;
ϵ_m	constant associated with λ_m ;
ζ_a	arithmetic mean of ζ_e and ζ_i ;
ζ_e	damping coefficient of equipment;
ζ_g	geometric mean of ζ_e and ζ_i ;
ζ_i	i -th modal damping coefficient of structure alone;

ζ_i^*	i -th modal damping coefficient of combined system;
λ_m	m -th spectral moment of equipment response;
$\lambda_{m,aa}$	m -th spectral moment associated with ω_a and ζ_a ;
$\lambda_{m,ee}$	m -th spectral moment associated with ω_e and ζ_e ;
$\lambda_{m,i,j}$	m -th cross spectral moment associated with modes i and j of structure alone;
$\lambda_{m,i,j}^*$	m -th cross spectral moment associated with modes i and j of combined system;
ν	mean zero-crossing rate of $R(t)$;
ν_e	reduced mean zero-crossing rate;
ν_i	mean zero-crossing rate associated with mode i ;
$\rho_{m,i,j}$	cross-correlation coefficient between modal responses of combined system;
σ_R	root-mean-square of $R(t)$;
$\sigma_{\dot{R}}$	root-mean-square of time-derivative of $R(t)$;
σ_{R_τ}	standard deviation of R_τ ;
τ	duration of stationary excitation;
Φ	modal matrix of structure alone;
Φ^*	modal matrix of combined system;
Φ_i, Φ_i^*	i -th columns of Φ and Φ^* , respectively;
Φ_i^{*E}	exact value of Φ_i^* ;
ϕ_{mi}, ϕ_{mi}^*	m -th elements of Φ_i and Φ_i^* , respectively;
Ψ_i	effective participation factor for attachment point for i -th mode of structure alone;
Ψ_i^*	effective participation factor for i -th mode of combined system;
ω	frequency;
$\bar{\omega}$	response mean frequency;
ω_a	arithmetic mean of ω_e and ω_l ;
ω_e	equipment frequency;
ω_i	i -th modal frequency of structure alone;
ω_i^*	i -th modal frequency of combined system;

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