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REPORT WR 71-1

INVESTIGATION OF THE POSSIBLE STRUCTURAL OVER-TEST
DUE TO THE QUALIFICATION OF SPACECRAFT STRUCTURES
IN A REVERBERANT ROOM

FOR

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By

V. Mason

Wyle Laboratories
128 Maryland Street
El Segundo, California

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SUMMARY

This report examines the possible magnitude of the structural over-test received by a spacecraft structure which is qualified in a reverberant room. It is assumed that the broadband excitation levels in the reverberant room are the same as those in the actual in-service environment. The results from one critical analysis have indicated the over-test might be larger. This report examines this problem assuming that it results from the fact that a reverberant sound field is composed of a multitude of resonant acoustic standing waves whereas the actual environment will be essentially non-resonant with a smooth excitation spectrum. The problem was approached by using a simple model of a structure in a reverberant room. This showed that when the coupling between the reverberant sound field and the structure was small, the structural response could be obtained, to a first approximation, by the product of the magnitude of the acoustic excitation spectrum and the magnitude of the structural response function without reference to phase angle. From this, a basis for an over-test was established. The magnitude of the over-test was found to depend on the band width used to analyze the structural response. Expressions for the magnitude of the over-test were obtained. It was shown that an over-test could be produced in a reverberant acoustic test but that its magnitude was not sufficient to account for the reported large discrepancy between vibro-acoustic response for reverberant tests and in-service environments.

TABLE OF CONTENTS

1.0 INTRODUCTION

A very convenient method of testing aerospace structures under a high level acoustic environment is to utilize a reverberant room. The advantages of such an approach are numerous as long as the detailed implications of replacing the actual acoustic or other fluctuating pressure environment with a reverberant acoustic environment are fully understood. One potential consequence of this change in environments is that the structure could receive an over-test. This could be due to one of several reasons, although in this paper only one will be examined.

The sound pressure level which is specified in order to test a structure in a reverberant room is usually the same as the level measured or estimated for that structure in its in-service environment. This is known to be an inadequate way to specify the reverberant test levels because of the basic difference in the detailed structure of the reverberant sound field compared with that of the actual acoustic environment it is intended to simulate. One way to determine the degree to which these differences affect the response of the structure is to compare the structural response obtained in a reverberant acoustic environment with that obtained in the actual in-service environment. Such an evaluation has been made by Kennedy and Saint (Reference 1) who compared the response of a Saturn IV-B Forward Skirt, Instrument Unit (IU) and Spacecraft Lunar Module Adaptor (SLA), as measured in the 100,000 cubic foot reverberation room at Wyle Laboratories, Huntsville, Alabama, with the response measured during an actual flight of the Saturn V rocket. To make this comparison, they computed the vibro-acoustic transfer function, T, defined as

$$T^2 = \frac{PSD_v}{PSD_a}$$

where PSD_v is the measured structural acceleration power spectral density and PSD_a the acoustic pressure power spectral density. Thus the transfer function, T, is proportional to the ratio of the root mean square acceleration to the root mean square acoustic pressure. When this transfer function was calculated, a consistent difference was evident between the transfer function derived from the reverberant room tests and those from the actual flight. Figure 1 shows one specific set of results presented by Kennedy and Saint. On the basis of this figure plus a similar plot obtained for another measurement position, they obtained a curve showing the average decibel difference between the reverberant room transfer function and the in-flight transfer function. This curve is shown in Figure 2 and shows an increasing discrepancy with decreasing frequencies.

Several mechanisms have been postulated to explain the above discrepancy. They all relate to the differences between the detailed characteristics of the reverberant sound field and the actual in-service acoustic environment. The first is the difference in the propagation velocity between the actual and reverberant sound fields. Thus the actual in-service sound field will be propagating either at the speed of sound (for purely acoustic excitation) or at the boundary layer convection velocity (for turbulent boundary layer excitation). A reverberant sound field, on the other hand, involves an infinity of propagating waves with convection velocities along the structure ranging from sonic to essentially infinite.

A second major difference is in the spatial correlations for each type of excitation. Thus the in-service acoustic sound field (as produced, for example, by a rocket exhaust) will have a spatial correlation which can be approximated by a decaying cosine wave in the direction of propagation and a unity or slowly decaying correlation normal to the direction of propagation. For boundary layer excitation, the correlation functions are similar although the decay rates both in the direction of propagation and perpendicular to it are more rapid than for acoustic excitation. For a reverberant field, however, the spatial correlation is radially symmetric to a first order approximation, having the form of the function $(\sin x)/x$. These two factors are potentially capable of producing marked differences between the response of a structure to a reverberant field compared to that produced by its in-service environment. White and Bozich (References 2 and 3) have investigated these effects theoretically and shown they can be significant under certain circumstances.

A third major difference is that the reverberant sound field is essentially composed of a series of acoustic standing waves corresponding to acoustic modes of the room which generally have a high Q factor, whereas the in-service acoustic excitation has, in general, a uniform distribution of energy across the frequency range. This discrepancy has several potential consequences. The first of these is that at low frequencies, the frequency spacing of the acoustic resonances in a reverberant room may not be sufficiently close to provide uniform random excitation of structural modes. This situation can be accentuated if certain of the lower frequency acoustic modes do not couple well with the horn used to excite the room modes. There is therefore a possibility of a considerable variation in structural response, depending on whether a structural resonance coincides with a strong acoustic resonance or not. There is also a possibility that the situation can be further aggravated by an effect similar to that noted by Eldred, et al (Reference 4) when examining the response to a reverberant environment of a small resonant structure (such as an electrical "black-box") mounted on the main structure of an aerospace vehicle. They showed that under certain conditions the effective Q factor for the small structure could equal the product of the Q factor for the main structure with the Q factor for the small structure in isolation from the main structure. In this case, the response of the smaller structure can be several orders of magnitude greater than would be expected. This result follows from the examination of the response of two simple mass spring systems coupled in series. Such an examination has also been made by Crandall and Mark (Reference 5) and by Curtis and Boykin (Reference 6). The dynamic vibration absorber also falls into this category and den Hartog (Reference 7) has discussed the practical aspects of such a phenomenon.

The response of a structure excited by a reverberant field can also be considered to fall into this category because the acoustic resonant response is coupled to a resonant structure. The main difference between this reverberant situation and the analyses reported in the literature lies in the number of interacting resonances. Thus in the literature, the interaction of only two modes is examined whereas in reverberation room testing, a multitude of acoustic modes interact with a similar number of structural modes. Fortunately, for the reverberant response, the problem can be simplified by examining the response of a single structural mode, determining the effect of the acoustic-structural coupling and hence deducing whether the resulting response differs significantly from that caused by the actual in-service environment. In the following sections, this aspect of the response of a structure to an acoustic field will be examined to determine whether this has a bearing on the discrepancy discussed by Kennedy and Saint.

2.0 A SIMPLE COUPLED ACOUSTIC-STRUCTURAL RESONANT SYSTEM

As a preliminary step in examining this problem, we consider the simple system shown in Figure 3. This shows a tube with cross-sectional area A. At one end of the tube (at x = o) is a piston forced to move sinusoidally with velocity amplitude U_o. At a distance x = L down the tube there is a mass spring system, which has an acceleration transfer function

$$H(\omega) = \frac{\ddot{y}}{P} = \frac{-(A/m)\,\omega^2}{\omega_n^2 - \omega^2 - 2i\omega\omega_n\xi} \tag{1}$$

Here, P is the amplitude of the pressure waves exciting the mass-spring system, the mass (m) has a natural frequency ω_n on the spring and the viscous damper has critical damping ratio $\xi = 1/2Q$.

The motion of the piston at x = o causes acoustic waves to travel with velocity c down the tube, which is assumed to be filled with air of density ρ. These waves first reflect off the face of the mass at x = L, the complex ratio of the acoustic pressure after reflection to the acoustic pressure before reflection being b_2. The reflection coefficient for the waves reflecting off the piston at x = o is assumed to be wholly real and equal to b_1. If the mass-spring system were held rigid, we choose $b_1 = b_2$ and then the magnitude of b_1 is determined by choosing a reverberation time for the acoustic waves traveling within the tube. If the reverberation time (T) is defined as the time required for the sound pressure level to decay 60 dB, then the reflection coefficient is given by

$$\log(b_1) = -(3L/cT) \tag{2}$$

where the value (cT/L) is the number of times the sound is reflected during time T.

When the mass-spring system is free to vibrate, the reflection coefficient b_2 can be determined by considering the reflection of plane waves off the face of the mass. This yields

$$b_2(\omega) = b_1 \left\{ \frac{1 - i\,(\rho c/\omega)\,H(\omega)}{1 + i\,(\rho c/\omega)\,H(\omega)} \right\} \tag{3}$$

The acoustic pressure at any point in the tube can be obtained by summing the contribution from the plane waves produced by the piston at x = o as they are reflected successively from each end of the tube. This implies summation of an infinite series. However, for $b_1 < 1.0$, the series is finite and yields a pressure

$$P(x,\omega) = \frac{P_o\left(e^{ikx} + b_2(\omega)e^{2ikL}\,e^{-ikx}\right)e^{-i\omega t}}{1 - b_1\,b_2(\omega)\,e^{2ikL}} \tag{4}$$

where $P_o = \rho c\, U_o$ is the pressure amplitude of the plane wave produced by the motion of the piston at $x = o$. This expression is in agreement with the similar expression obtained by Heckl and Seifert (Reference 8). From this expression, the acoustic pressure exciting the mass-spring system is

$$P(L,\omega) = \frac{P_o e^{ikL}\,(1 + b_2(\omega)\,e^{-i\omega t}}{\left(1 - b_1 b_2(\omega)e^{2ikL}\right)} \tag{5}$$

Consequently, the response of the mass-spring system to the reverberant sound field in the length $o \leqslant x \leqslant L$ is

$$\ddot{y}(\omega) = P(L,\omega)\;H(\omega) \tag{6}$$

This equation gives the response of a simple structural system which is driven via a resonant acoustic system. It should be noted that the product of the term (A/m) from Equation (1) and $(\rho c/\omega)$ in Equation (3) controls the degree of coupling between the acoustic and structural systems. The non-dimensional quantity $(A\rho c/m\omega_a)$ is therefore termed the coupling factor. If it is zero, the acoustic and mass-spring systems vibrate independently of each other. The quantity ω_a is the fundamental resonant frequency of the acoustic system with rigid ends.

It should be realized that the simple system of Figure 3 and the assumptions made in deriving the above equations are not completely representative of the situation that exists in a reverberant test chamber. Nevertheless, the similarities are sufficiently great for the above equations to give valuable insight into the response of a single structural mode to a reverberant sound field. Equations (5) and (6) were therefore evaluated for a series of conditions with varying degrees of coupling. In all cases, the damping of the mass-spring system was chosen to give $Q = 1/2\xi = 15$ and $b_1 = 0.995$. This value of b_1 corresponds to a reverberation decay time of 24 seconds when the damping provided by the mass-spring system is ignored and the tube is approximately 20 feet long. Figure 4 shows the acoustic pressure developed at the surface of the mass, m, when the resonance of the mass-spring system is equal to the fundamental acoustic resonant frequency for the tube with rigid ends. The reference level for the acoustic pressures is $P_o = \rho c\, U_o$, which is the acoustic pressure which would be produced by the piston if it radiated into an infinite tube. The figure shows that when the coupling is large (i.e., $A\rho c/m\omega_a = 0.5$), the two separate resonances which result from the coupled motion in the frequency range near $(\omega/\omega_a) = 1.0$ can be clearly identified in the acoustic response. At the uncoupled resonant frequency $\omega/\omega_a = 1.0$, the mass-spring system is acting as a dynamic absorber and greatly reduces the acoustic pressures. As the coupling factor is reduced, the frequency separation between the coupled resonant peaks decreases while the acoustic level at the uncoupled resonant frequency increases. The higher acoustic resonances ($\omega/\omega_a = 2, 3, 4$ ---) are affected much less by the mass-spring resonance. There is some decrease in the level of the higher acoustic resonances because of the energy absorbed by the off-resonant excitation of the mass-spring system, but it only amounts to a few decibels, even with large coupling factors.

Figure 5 shows the acceleration response of the mass-spring system produced by the acoustic pressures shown in Figure 4. The reference level for the response is the mass law response at high frequency. This figure shows also the basic acceleration response function H(ω). The acceleration response shows many of the same characteristics as the acoustic excitation curves. However, the most important result from Figure 5 is that the response of the mass-spring system in the region of its (structural) natural frequency increases as the coupling is decreased. In the limiting case where the coupling is very small, the response of the mass-spring system can be seen to equal the product of the magnitude of the acceleration response function H(ω) and the magnitude of the uncoupled acoustic response curve of Figure 4 without regard to phase (i.e., the individual decibel response levels can be added directly). Thus, with very low coupling, the acceleration response at the mass-spring resonant frequency is given by the product of the Q factors for the mass-spring system and acoustic system. This is directly analogous to the "resonance-on-resonance" effect discussed in Reference 4.

Figures 6 and 7 show the acceleration response of the mass-spring system when the relationship between the mass-spring resonance and acoustic resonance is altered. In Figure 6, the structural resonant frequency was chosen to be one and a half times greater than the fundamental acoustic resonance, while in Figure 7 it is fifteen times greater. Both figures show that significant changes in the response occur when the structure is closely coupled to the acoustics, but that with low structural-acoustic coupling, the structural response can again be approximated by the addition, in decibel form, of the individual acoustic and structural response curves.

3.0 APPLICATION TO A REVERBERATION ROOM

The response of any structural mode to acoustic excitation can be written in a form very similar to the mass-spring system response (Equation 1) by using the concepts of generalized force, mass, stiffness and damping, the acoustic environment will have the form of a series of discrete peaks at resonant frequencies, although the frequency spacing of the peaks will not be constant as it was in the preceding simple example. Similarly, the coupling factor between each acoustic mode and one of the structural modes will not be constant. Consequently, some of the structural modes will respond to a greater extent than others. However, when we consider the acoustic excitation of a cylindrical structure in a reverberant room, the coupling for all structural modes will be low so that the structural response can be obtained by multiplication of the acoustic and structural response curves.

If it were possible to examine the acceleration response of just a single structural mode, it would have the characteristics shown in Figure 8. (This curve is only a conceptual sketch.) It should be emphasized that this is the response of a single structural mode. The peaks in this curve are the result of the acoustic excitation. The peak response in the structural mode may not occur at or near the structural resonant frequency, particularly if the acoustic modal density is low. However, it should be noted that the acceleration response of the mode above its resonant frequency may be of the same order of magnitude as the response at resonance. Similarly, the displacement response of the mode below the resonance may be of the same order of magnitude as the displacement response near the resonant frequency. The complete response of a structure in a reverberant environment is the sum of all structural modes such as that depicted in Figure 8. It will be realized that the peaks in the response function for each structural mode will occur at the same frequency, since they depend primarily on the acoustic excitation.

4.0 RANDOM EXCITATION

Up to this point, the response of the structures has been considered from the viewpoint of discrete frequency sinusoidal excitation. The results presented above can also be applied to random excitation. Considering the simple system of Figure 3, if the motion of the piston at x = o is assumed to be white noise with mean square spectral density U_o (feet per second)^2 per Hertz, then Figures 4 through 8 can be interpreted as spectral density response curves when the appropriate change is made to the scale of the vertical axis. (The scale values should be multiplied by a factor of two.) The response at other band widths such as one-third octave or full octave can then be obtained by integration of the spectral response curves. Thus it is now possible to examine Figures 5 through 8 to see what light they shed on the over-test problem discussed by Kennedy and Saint. From this point onward, the discussion will assume that the excitation and response are random in nature.

The starting point for the discussion will be the assumption that the actual environment which the reverberant excitation is trying to simulate has a nearly flat, smoothly varying spectral content. We will consider a portion of that spectrum covering the frequency band width Δf. For the sake of clarity in the subsequent argument, we will assume this is an octave band width, although it could in practice be any convenient measurement band width. We will further assume that the center frequency of this octave band is low (i.e., near the lowest acoustic modes in the reverberant chamber) so that there are only a few acoustic modes within this octave band. The actual (in-service) acoustic spectrum in this octave band will be nearly flat, so that the octave band level will be 10 log (Δf) decibels greater than the spectral level (see Figure 9). This octave band level is the level that is specified in the reverberation room test. Within this octave band, the reverberant spectrum levels will show marked variations, as shown in Figure 9, due to the acoustic resonances. If there are n such resonances within the octave band, each of which responds to approximately the same level, then the resonant peaks in the reverberant spectrum will be approximately 10 log (n) decibels below the octave band level. (The acoustic resonances which are not strongly excited are not included in the determination of n.) Thus in Figure 9, there are four such resonances, so the peaks in the reverberant spectrum will be 6 dB below the octave band level. If the octave band has a band width greater than 4 Hz, then for this particular example, the peaks in the reverberant spectrum will have a higher level than the broadband spectrum they are attempting to simulate. In more general terms, this excess will be approximately equal to

$$\Delta dB_e = 10 \log (\Delta f/n) \qquad (7)$$

Thus from the excitation viewpoint, there is seen to be a potentially significant difference in level between the reverberant sound pressure level and the actual environmental level to be simulated.

However, we can only determine if an over-test situation exists by examining the structural response. For the purpose of this argument, we will assume that only one structural mode responds to any significant extent in the octave band Δf. The generalized acceleration response of this mode to both the reverberant and broadband excitation is shown in Figure 10. The response spectrum resulting from the broadband excitation will show the classical single-degree-of-freedom response shape (as in Equation 1). The reverberant response will have the same basic spectrum as the reverberant excitation but modified by the gain of the structural mode. Thus the peaks in the structural response to the reverberant field will exceed the response to the broadband excitation by the same amount as the reverberant acoustic levels exceeded the broadband excitation spectrum levels, that is by 10 log ($\Delta f/n$) decibels for our simple model. Thus the potential for a structural over-test is seen to exist when the acoustic environment is controlled in broad frequency bands (e.g., octave bands) and the response is measured as spectral levels.

In practice, the structural response is also measured in band levels. It will be assumed that the response is measured in the same frequency bands as those used to specify the excitation. The octave band level resulting from the structural response to broadband excitation will equal the peak spectrum level, multiplied by an effective band width Δf_m. If $\Delta f_m < \Delta f$ and is wholly contained within Δf, then from Reference 4

$$\Delta f_m = \frac{\pi f_m}{2Q} \qquad (8)$$

where f_m is the resonant frequency of the structural mode. Thus the octave band (Δf) level for the broadband structural response will be 10 log (Δf_m) decibels above the peak in the response spectrum.

The amount by which the octave band level for the reverberant structural response exceeds the peak in the reverberant response spectrum will depend upon the number of significant peaks (n_m) in that octave band. This will vary with the spacing of the significant acoustic modes in the reverberant field and their place on the frequency scale relative to the structural resonant frequency. This number cannot be accurately predicted, but will lie in the range $n \geqslant n_m \geqslant 1$.

Thus the final difference between the octave band response level due to a reverberant excitation compared to the octave band response level due to broadband excitation is

$$\Delta dB_r = 10 \log (\Delta f/n) - 10 \log (\Delta f_m/n_m)$$

$$= 10 \log \left\{ \frac{\Delta f}{\Delta f_m} \quad \frac{n_m}{n} \right\} \qquad (9)$$

This is essentially a measure of the irregularity of the frequency spacing of the acoustic resonances in the reverberant field. When the acoustic modal spacing is regular and the modal density is reasonably high, then Equation 9 will equal zero. This means that in the working frequency range of a reverberant room there should be no significant difference between the octave band structural response level as measured in a reverberant room and the octave band level measured in the actual environment.

5.0 THE MAGNITUDE OF THE OVER-TEST PROBLEM

Having obtained two formulas (Equations 7 and 9) predicting the extent of any structural over-test, we will now determine if the actual magnitude of the predicted over-test is significant. The spectral response over-test (Equation 7) is directly proportional to the actual modal density in the reverberant room. To take a very pessimistic view, it might be possible for the actual modal density to be as low as one mode per 10 Hz at the low frequency end for a poorly designed reverberant room. This would give a spectral over-test of 10 dB, which must be considered a practical upper bound. Obviously, as the modal density approaches one mode per Hertz, this effect approaches zero and could become negative if the modal density increased beyond this amount. Turning to a case relevant to the results published by Kennedy and Saint, an examination of a sine-sweep trace taken in the Wyle Laboratories 100,000 cubic foot reverberant facility shows average modal densities of between 0.5 and 0.25 modes per Hertz for the one-third octave bands in the range 20 Hz to 80 Hz. Thus for the Wyle Laboratories facility, the potential spectral over-test lies in the range 3 dB to 6 dB.

The broadband response over-test (Equation 9) is proportional to the ratio of the modal density in the measurement band width to the modal density in the structural band width. It will therefore probably be smaller than the spectral response over-test. It is also more difficult to obtain representative values to substitute into Equation 9. However, consider a one-third octave band, centered on 60 Hz (Δf = 14 Hz) which contains only one acoustic room mode (i.e., the value of spectral over-test is 11.5 dB) and assume that the acoustic mode falls within the effective band width of a structural mode having a Q of 15 (i.e., Δf_m = 6 Hz from Equation 8). With these values, Equation 9 gives a broadband response over-test value of 3.7 dB. Of course, if that one acoustic mode did not fall within the structural mode band width, then the structure would be undertested. It will be noted that the broadband over-test factor is actually dependent on the structural Q factor. There are indications from model structural tests that Q factors much greater than 15 can be found. If the over-test factor is reexamined in this light and we again assume the extreme case where there is only one acoustic mode in the measurement band width and that one mode also falls within the structural band width, then the over-test will increase because the ratio of measurement to structural band width has increased. However, this tendency will be balanced by an increased chance that the acoustic mode will <u>not</u> lie within the structural band width, and hence reduce the chance of an over-test occurring.

In practice there will probably be more than one structural mode which responds within a certain measurement band width. If one of these modes has much greater response than the other, then obviously this mode will determine the magnitude of the broadband over-test factor. However, if the structural modes respond at nearly equal level, then the broadband response is given by the sum of the individual modal responses. The broadband over-test factor can, in this case, be determined by applying Equation 9 to each mode in turn before doing the summation over the measurement band width. In which case, the mode which has the greatest over-test factor on the basis of Equation 9 will determine the degree of apparent over-test in that measurement band width.

Finally, the analysis of Kennedy and Saint (Reference 1) must be considered in the light of the above results. Unfortunately, full details of their analysis are not at present available, but there is one major difference between their analysis and the assumptions used so far in this paper. Thus the reverberant sound pressure levels used in the test on which Kennedy and Saint based their results were controlled in broad frequency bands whereas they made their analysis in 5 Hz band widths. This does not alter the potential magnitude of the spectral over-test given by Equation 7 but it could have an important effect on the apparent over-test produced by their 5 Hz band width analysis. In particular, it should be noted that Equation 9 depends upon the measurement band width Δf being greater than the structural band width Δf_m. This, in general, will not be the case with the results of Kennedy and Saint. Thus even for a structural mode with a relatively low resonant frequency of 60 Hz and a Q of 15, the structural band width is 6 Hz. This means that the structural band width Δf_m in Equation 9 should be replaced by the measurement band width (i.e., 5 Hz) which will have the effect of increasing the magnitude of the apparent over-test below the frequency at which the modal density in the 5 Hz measurement band width equals the modal density in the Δf control band width. However, the effect is not very large, giving an apparent increase in over-test of 0.8 dB for a structural mode at 60 Hz (Q = 15) or 3 dB for a structural mode at 100 Hz.

The above results therefore demonstrate that it is not possible for the effects discussed and quantified above to produce the discrepancies of the magnitude of those found by Kennedy and Saint. Another important result is that the discrepancies that would be expected due to irregularities in the acoustic modal frequency spacing can be seriously affected by the band width used in the structural analysis.

Finally, before closing this discussion, it should be pointed out that, while there is no doubt that it is possible for discrepancies of the type reported by Kennedy and Saint to exist, there are several points which are not satisfactorily answered in their paper and which could significantly change the magnitude of the discrepancy shown in Figure 2. These are:

- The positions of the microphones used to measure the acoustic levels are not specified. The acoustic pressure which should be used to calculate the transfer function, T, is the pressure at the surface of the structure. The pressure at other points in a reverberant sound field will vary considerably from the surface pressure level. The same may also be true of the in-flight acoustic pressure measurements.

- The scatter on the in-flight results as shown in Figure 1 is very large indeed - much larger than would be expected.
- Figure 2 indicates an asymptotic difference between the two transfer functions of approximately 5 dB at high frequencies (i.e., 800 Hz and above). This is unexpected and could be a result of not using the acoustic pressure actually measured at the surface of the structure, as discussed above.

6.0 CONCLUSIONS

This paper has discussed the potential consequences that result from the fact that a reverberant acoustic field is composed of a multitude of acoustic resonances. The response of a structure to such a field was considered and compared with the structural response to a broadband (non-resonant) acoustic environment which is more typical of the in-service environment.

A preliminary analysis was undertaken to determine the effects of a structural resonance on the acoustic resonances composing the acoustic field. It was shown that when the coupling between the acoustic and structural modes was small, the structural response could be obtained as the product of the magnitude of the reverberant sound pressure spectral level and the magnitude of the structural response function without regard to phase effects. In general, the absolute level of coupling between a structure and a reverberant sound field is known to be low enough for this effect to apply to a first order approximation.

Utilizing this fact, the problem of the apparent over-test provided by reverberant excitation was examined and compared with the magnitude of the over-test observed by Kennedy and Saint (Reference 1). It was shown that two basic types of apparent over-test exist. Both rely on the fact that the levels in the reverberant sound field are controlled in broad-frequency bands to equal the broadband levels in the actual acoustic field being simulated.

The first type of over-test was found when the spectral (per Hertz) response of the structure to a reverberant field was examined. This is directly proportional to the acoustic modal density in the reverberant room and will therefore be negligible when the actual modal density in the reverberant room is high. This type of over-test will therefore only be significant below or at the lower end of the working frequency range, but even here it is unlikely to exceed 10 dB.

When the structural response was examined in broad-frequency band (e.g., one-third or full octave bands), a second potential over-test situation was noted. This had a smaller magnitude than the first effect and rather than being directly dependent on modal density, was found to depend on irregularities in the modal frequency distribution within the measurement band width. When the modal density is high, this second apparent over-test will be negligible and is unlikely to exceed 4 dB even at the lower end of the working frequency range for the reverberant room.

It has therefore not been found possible to explain the large discrepancies found by Kennedy and Saint on the basis of the above analysis. However, the results presented by Kennedy and Saint are preliminary in nature and it is felt that when the results of their more extended analysis are available, they may well be reduced to a level compatible with those deduced in this paper.

REFERENCES

1. Kennedy, D.R. and Saint, L.D., "Reverberation Room Testing of Space Vehicle Structures." Paper presented at 75th Meeting of the Acoustical Society of America, Ottawa, Ontario, Canada, May 1968.

2. White, R.W., "Predicted Vibration Responses of Apollo Structure and Effects of Pressure Correlation Lengths on Response," Wyle Laboratories Research Staff Report WR 67-4, Revised March 1968.

3. Bozich, D.J. and White, R.W., "A Study of the Vibration Responses of Shells and Plates to Fluctuating Pressure Environments," NASA CR-1515, 1970.

4. Eldred, K., Roberts, W., and White, R., "Structural Vibrations in Space Vehicles," Wright-Patterson Air Force Base, WADD Technical Report, 61-62, March 1961.

5. Crandall, S.H. and Mark, W.D., "Random Vibrations in Mechanical Systems," Academic Press, 1963.

6. Curtis, A.J. and Boykin, T.R., "Response of a Two Degree of Freedom System to White Noise Base Excitation," J. Acoustical Society America, Vol. 33, p. 655, 1961.

7. den Hartog, J.P., "Mechanical Vibrations," McGraw Hill, 1956.

8. Heckl, M. and Seifert, K., "Investigations of the Influence of Eigen-resonances of Rooms on the Results of Sound Insulation Measurements." Acustica, Vol. 8, p. 212, 1958, (in German).

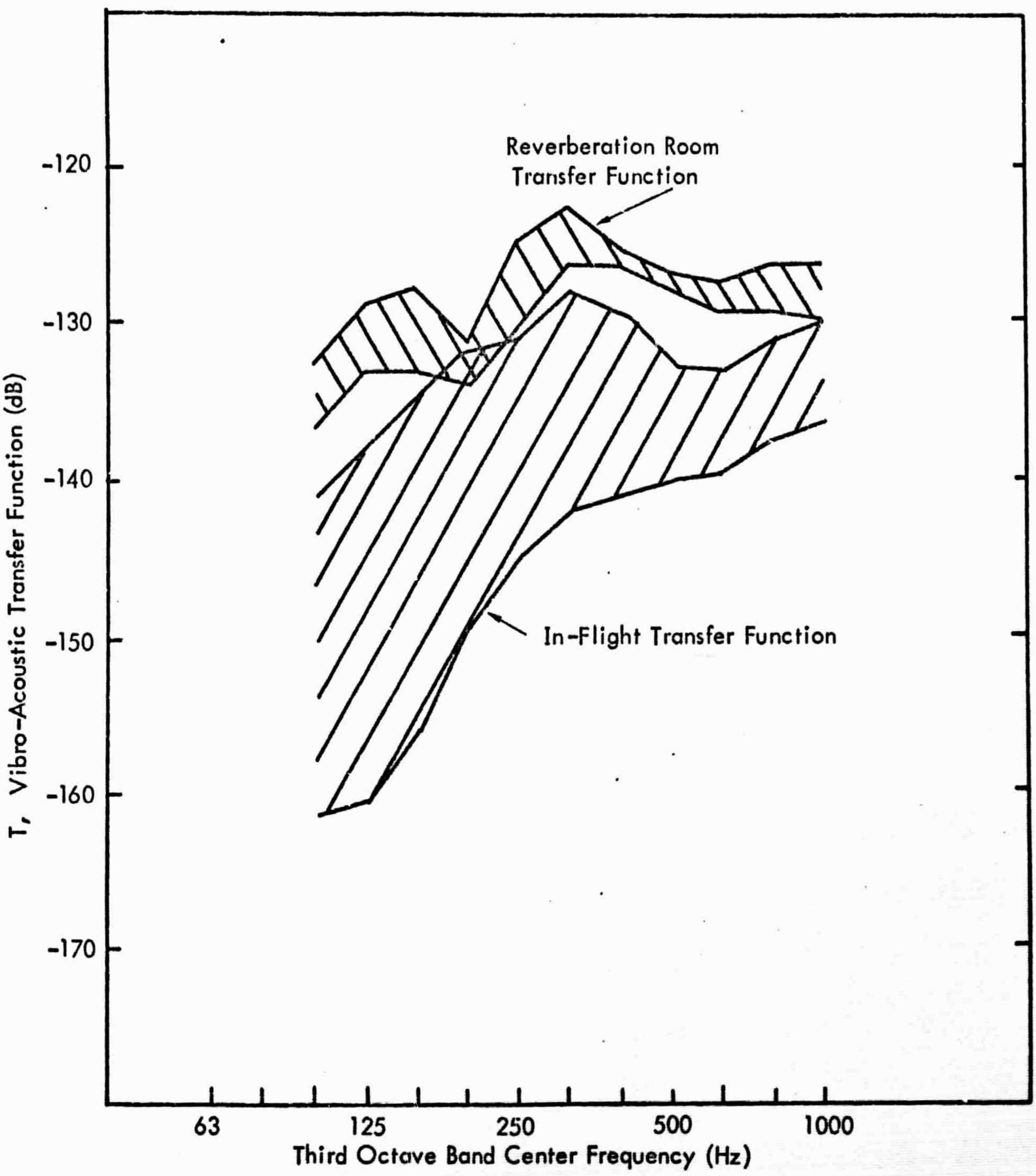

Figure 1. Comparison of In-Flight and Reverberation Room Transfer Functions (from Reference 1).

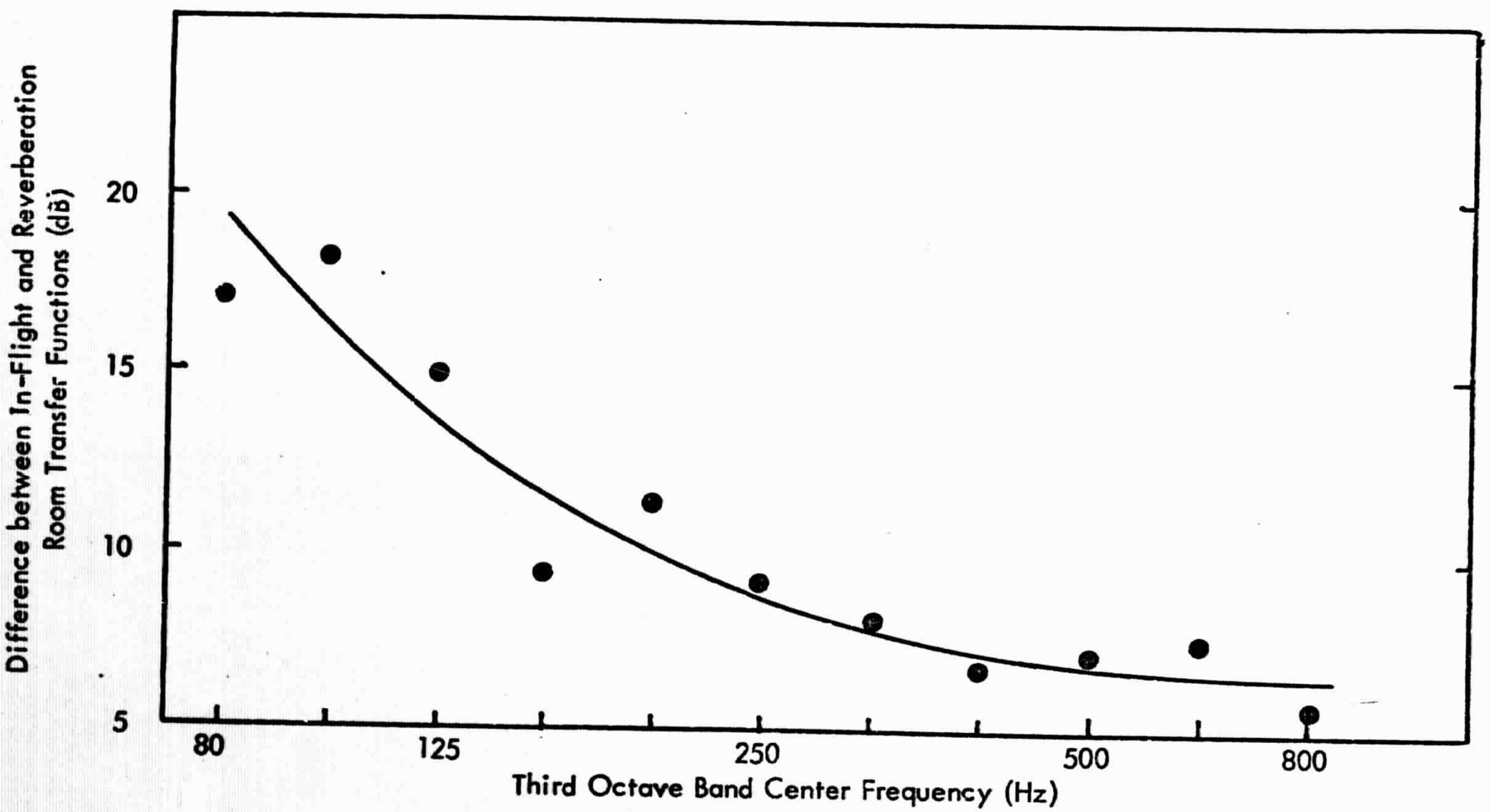

Figure 2. Difference between the Calculated In-Flight and Reverberation Room Transfer Functions (from Reference 1).

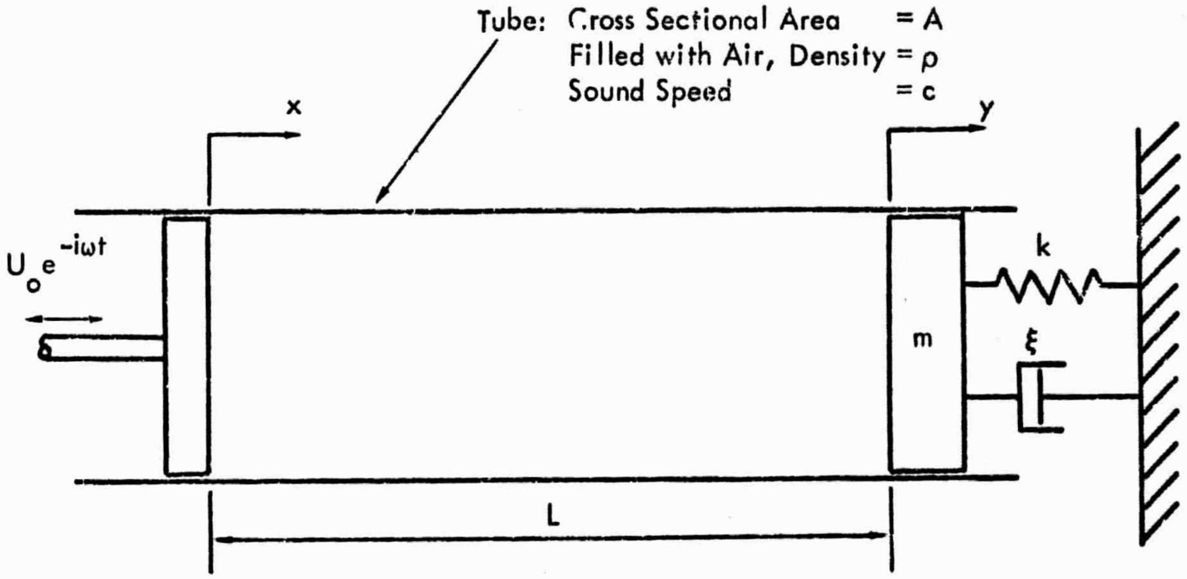

Figure 3. A Simple Coupled Acoustic and Structural System

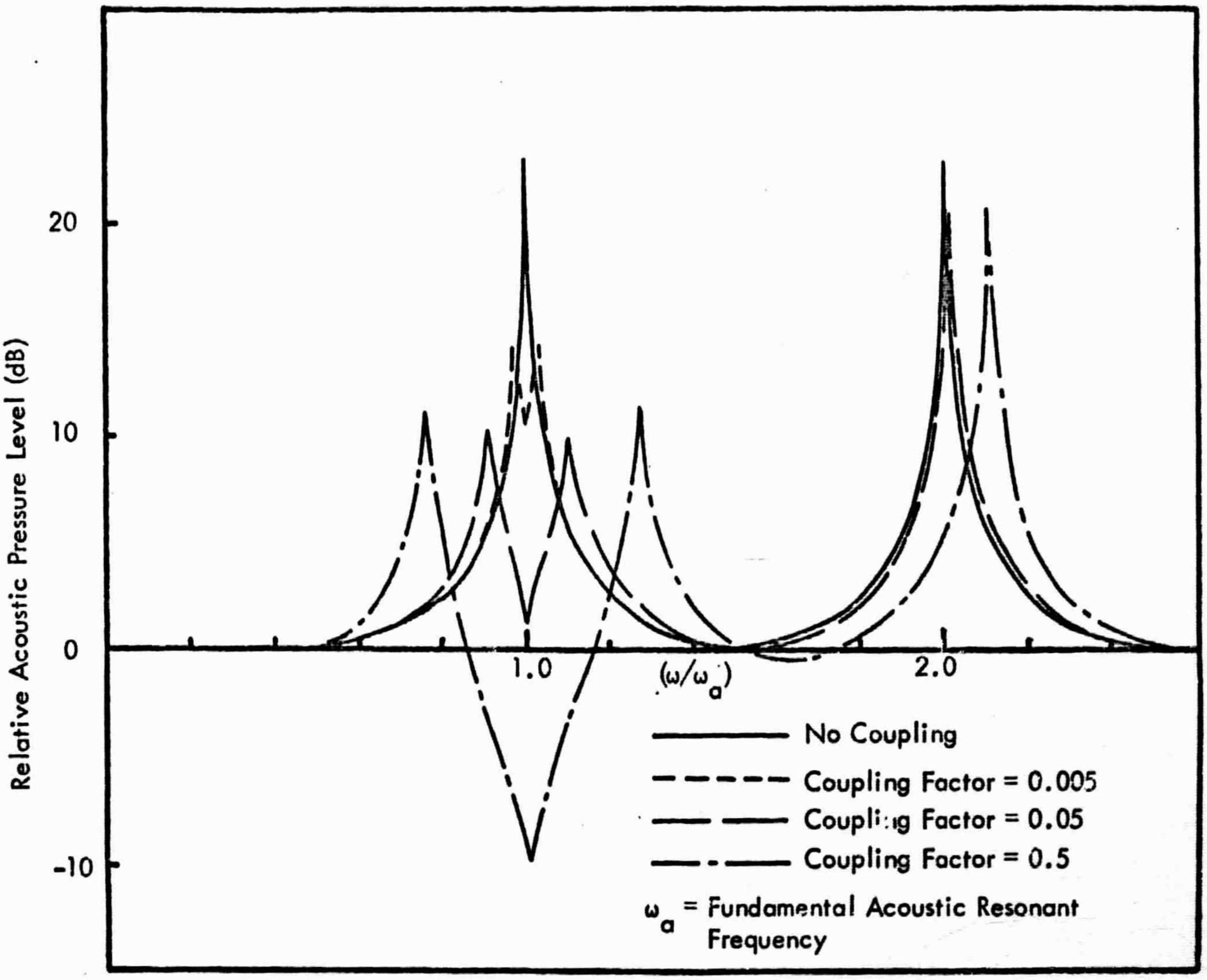

Figure 4. Acoustic Pressures Forcing the Mass-Spring System with Resonant Frequency Equal to Fundamental Acoustic Resonant Frequency.

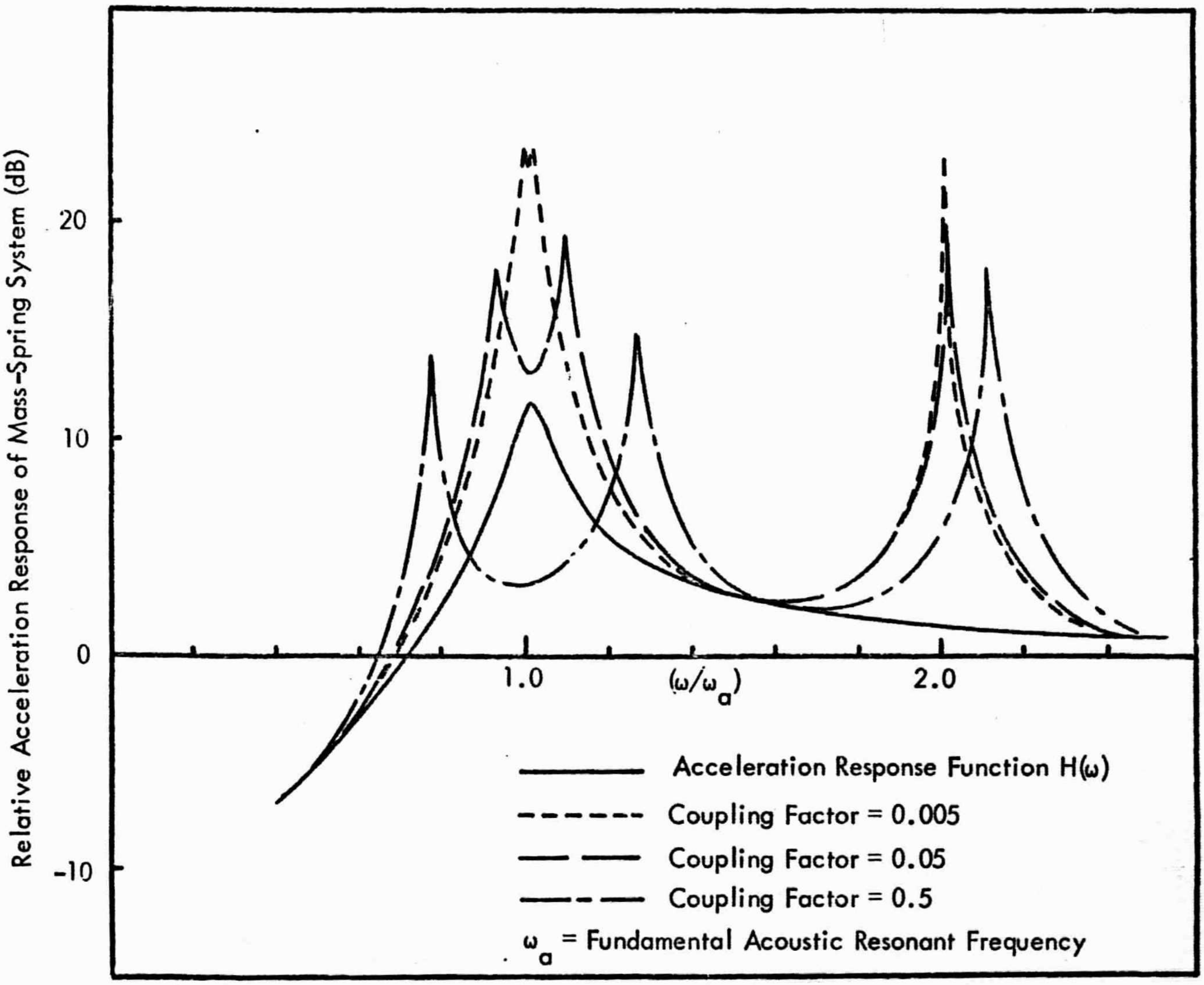

Figure 5. Response of the Mass-Spring System with Resonant Frequency Equal to Fundamental Acoustic Resonant Frequency.

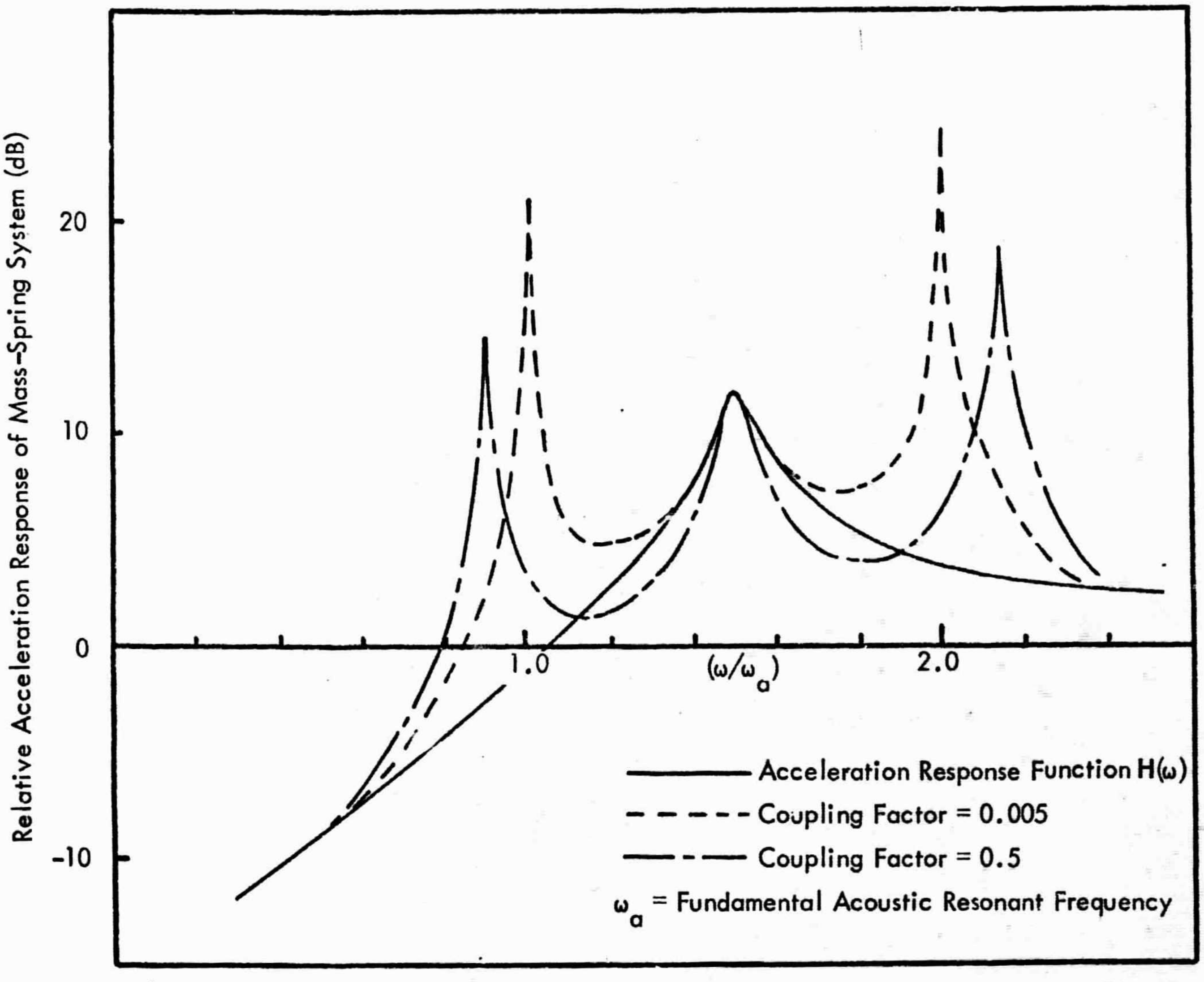

Figure 6. Response of Mass-Spring System with Resonant Frequency at 1.5 times the Fundamental Acoustic Resonant Frequency.

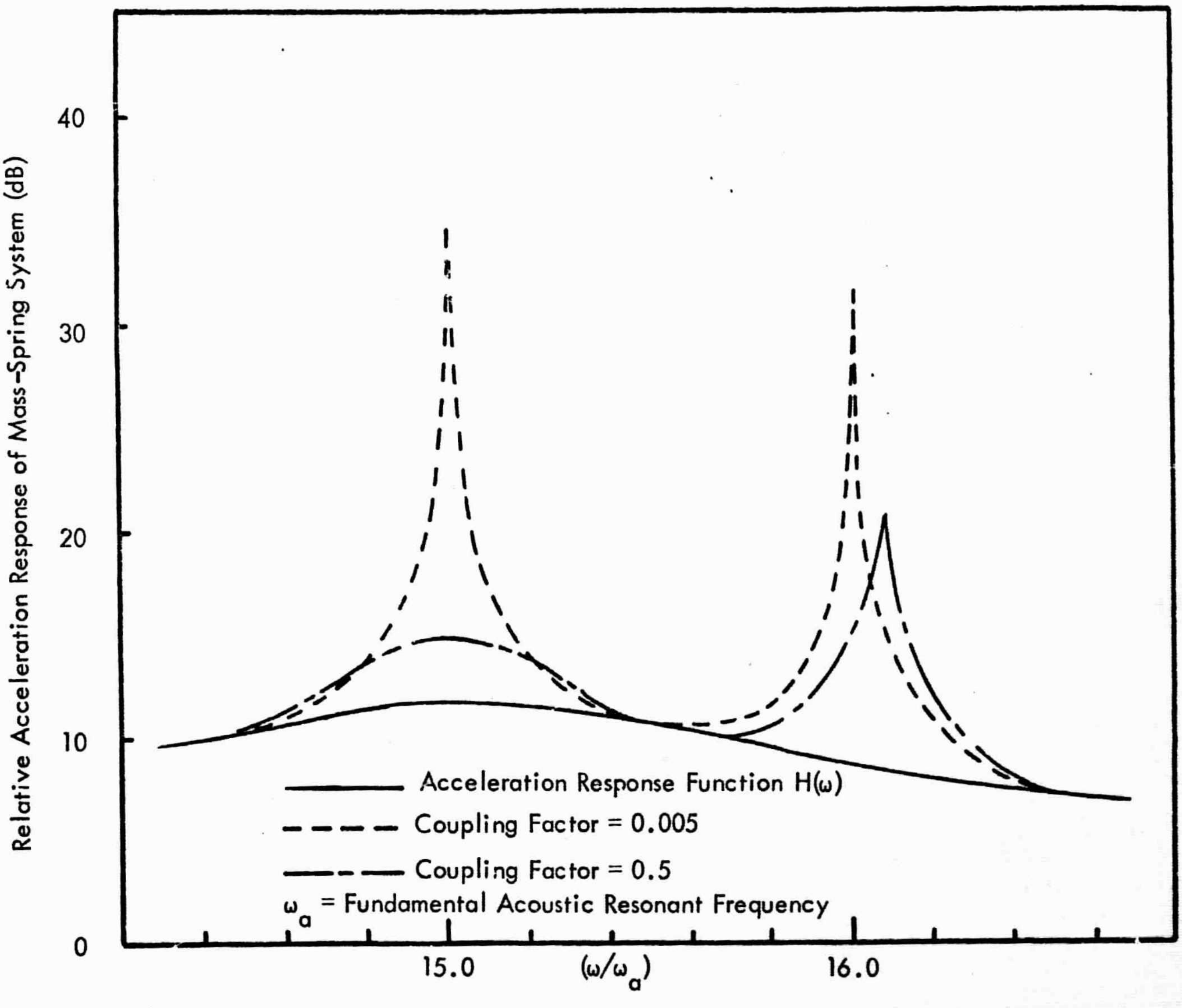

Figure 7. Response of the Mass-Spring System with Resonant Frequency at Fifteen times the Fundamental Acoustic Resonant Frequency.

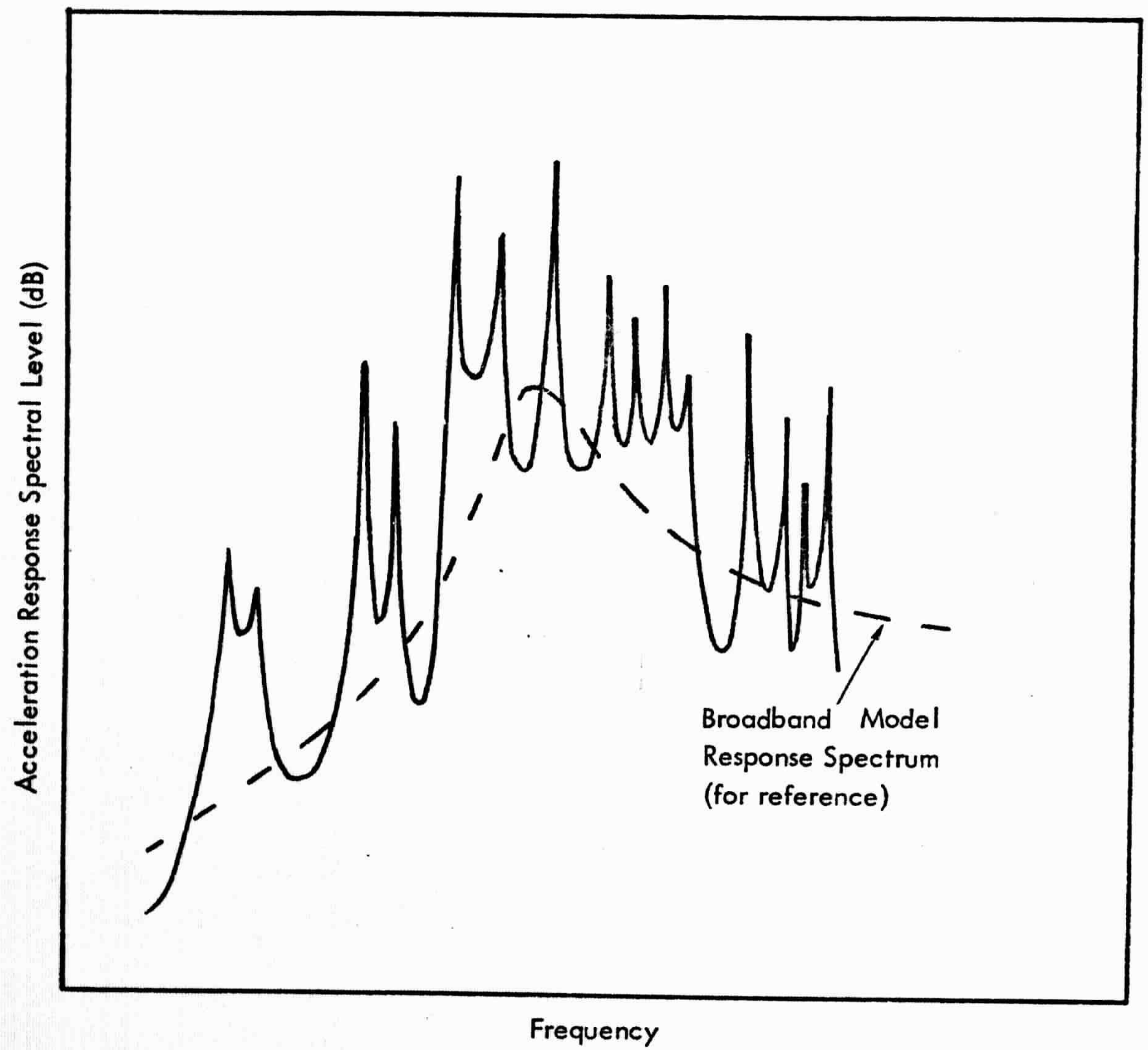

Figure 8. Sketch of Typical Response Spectrum for a Single Structural Mode.

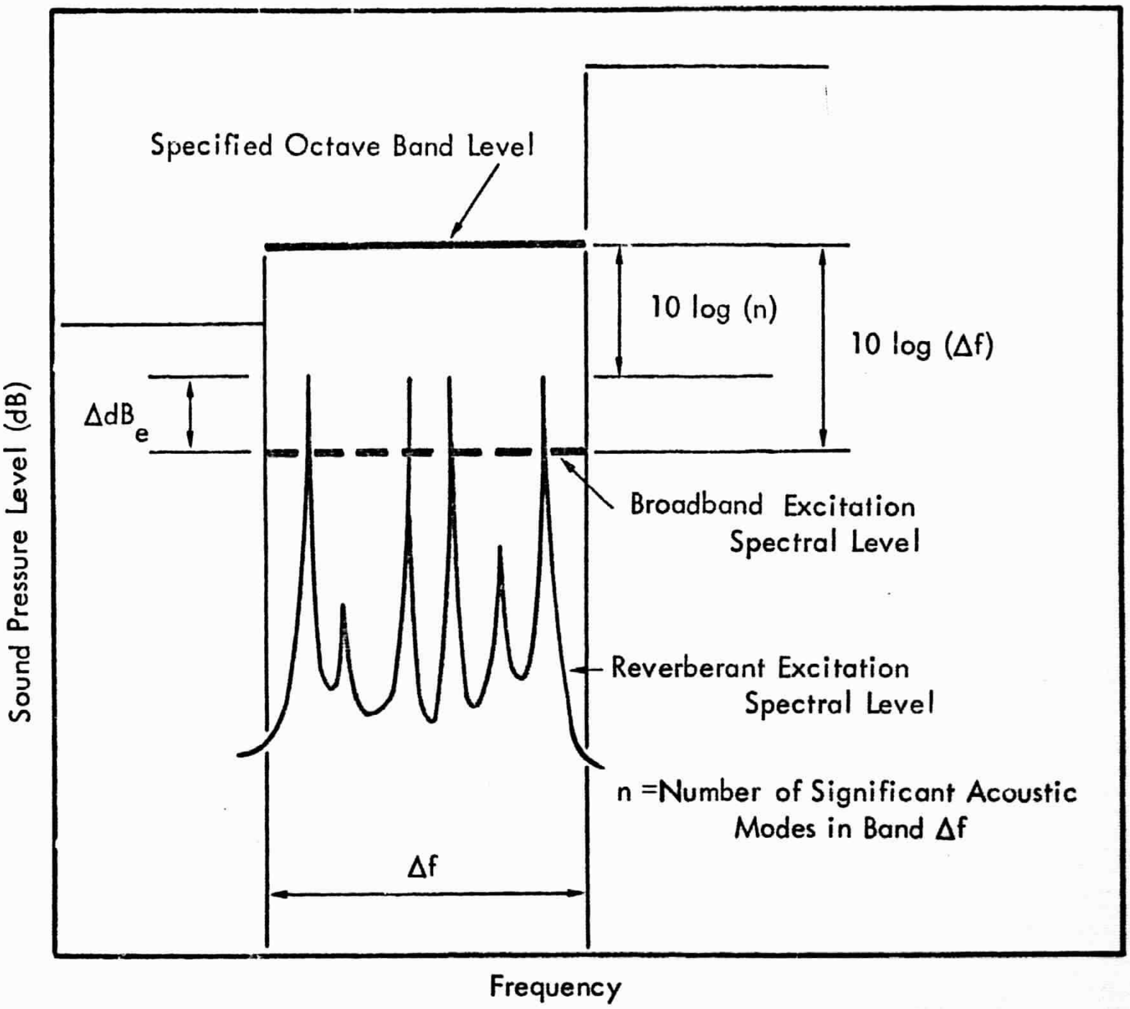

Figure 9. The Relationship between the Reverberant Excitation Spectra and Broadband Excitation Spectra.

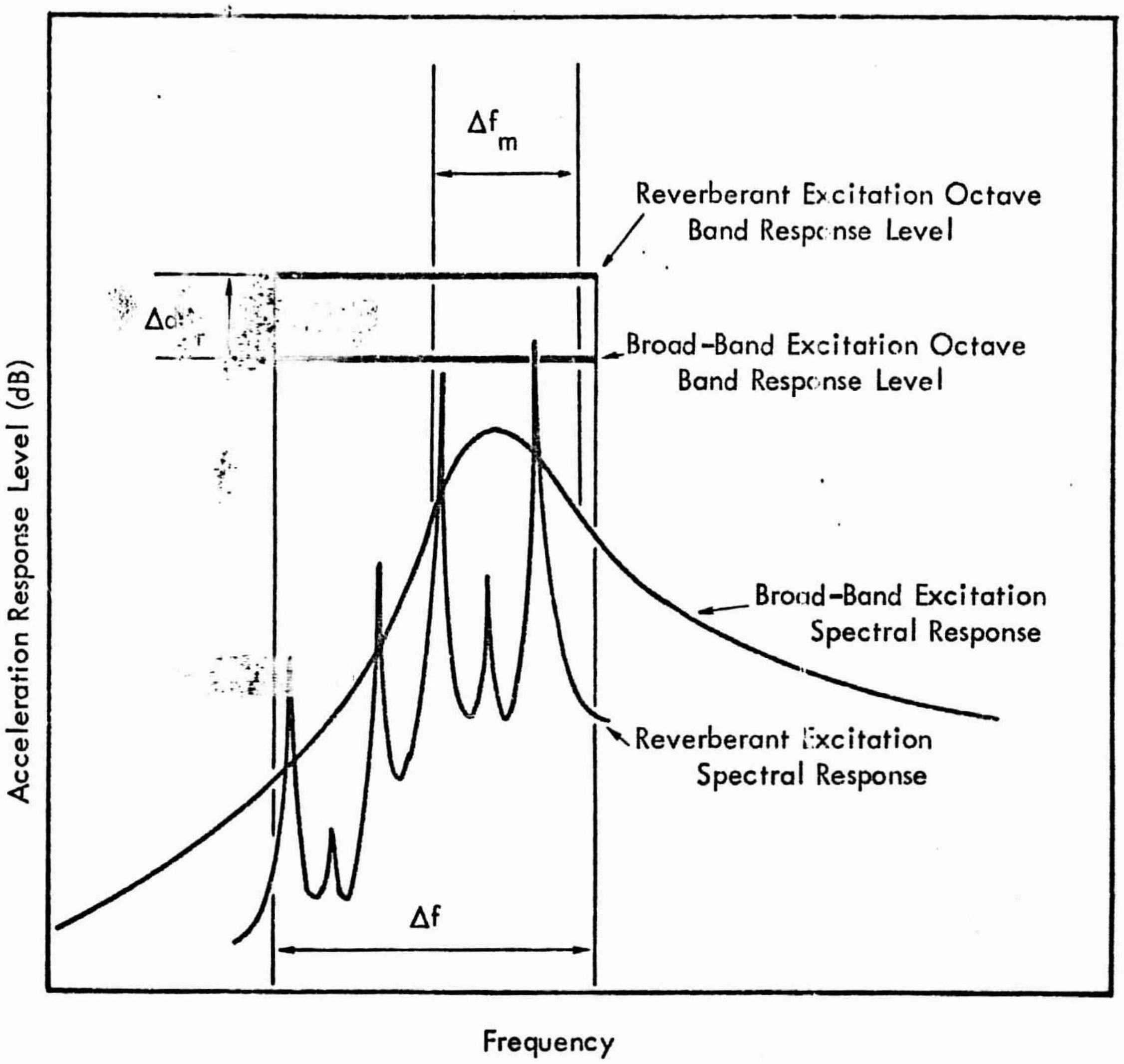

Figure 10. The Variation in Response of a Structural Mode between Reverberant and Broadband Excitation.