PASSIVE CONTROL OF A FLEXIBLE PLANAR TRUSS USING A REACTION MASS ACTUATOR

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ABSTRACT

Damping is an important aspect in controlling large flexible structures in space. This paper deals with the passive control of a flexible twenty–bay truss using a reaction mass actuator. The actuator was electronically tuned to the second mode of the truss and attached to the tip of the truss. The truss was then subjected to random vibrations. The results were impressive, with an 86 percent reduction in the peak amplitude of vibration of the second mode of the truss when compared to the structure's uncontrolled response. In addition, the peak amplitudes of vibrations of the higher modes of the truss were reduced with the actuator passively tuned to the second mode; there was a small change in the peak amplitude of vibration of the first mode. Further, the settling time for the structure with the passively tuned actuator was less when compared to that of the truss without the actuator attached. With no passive damping it took 19.5 seconds for the second mode of the truss to settle, while the addition of the passive actuator reduced the settling time to four seconds. The passively tuned reaction mass actuator is, therefore, an effective mechanism to reduce the vibrations of a flexible truss.

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INTRODUCTION

It is envisioned that, in the near future, space structures will become larger and more complex. Due to the high cost of transporting materials into orbit, these future space structures will be made from lightweight and flexible components. While the components will reduce the number of launches required to place large space structures in orbit, their flexibility will become an important control issue.

A flexible structure's vibrations can be reduced, or eliminated, either by active control, passive damping, or a combination of both. Active control incorporates feedback control systems to sense the structure's vibrations and calculates control inputs to dampen unwanted oscillations. Passive damping deals with the structure damping its own oscillations as a result of its structural design, materials properties, or the addition of devices to the structure.[1] This paper deals only with examining the effects of passive damping on a structure.

Specifically, the purpose of this paper is to examine, both analytically and experimentally, the interactions between a passively tuned actuator and a flexible truss structure to which it is attached. A study similar to this has already been conducted,[2] but this work utilizes a truss structure with characteristics mirroring those anticipated on large flexible structures. In addition, while reference 2 dealt with reducing the magnitude of vibration of a structure's first resonant mode, this research effort has concentrated on reducing the vibrations of a structure's higher modes. A 20–bay planar truss has been built and a reaction mass actuator has been attached to its tip. In addition, a finite element model of the structure has been developed and validated. With these experimental and analytical tools, a study has been conducted to examine the interactions between the truss and the actuator. This paper summarizes the results of this study and addresses the experimental and analytical set–ups, as well as the results obtained when the actuator was used to passively dampen the vibrations on the truss.

PLANAR TRUSS SET–UP

This study was conducted using the 20–bay, 7.07–meter long planar truss located in the Engineering Mechanics Laboratory at the US Air Force Academy (see Figure 1). The diagonal and side dimensions of each square bay were 0.500 m and 0.354 m, respectively. The chordwise members were fastened to steel bars which provided extra mass to scale down the structure's natural frequencies. The chordwise truss members were, then, effectively rigid when compared to the longeron and diagonal members. The 239 pound truss was oriented in a horizontal plane and supported by 3/4 inch diameter steel balls on table tops. The steel balls rolled with very little friction on small steel plates which rested on the table surfaces. The root of the truss was attached to a heavy steel table that was bolted to a concrete floor, and the truss tip was free to move. Air jet thrusters were attached to the tip and mid–span of the truss. A complete description of the planar truss can be found in another study.[3]

Finite element models of the truss were generated using both MSC/NASTRAN and SDRC IDEAS, and these models were modified until they produced resonant frequencies close to those obtained from the experimental truss. These frequencies, shown in Table 1, validate the analytical models.

TABLE 1

Experimental and Analytical Natural
Frequencies of the 20–Bay Truss

MODE	EXPERIMENTAL (Hz)	ANALYTICAL (Hz)	PERCENT ERROR FROM ANALYTICAL
1	1.837	1.585	–16.046
2	9.438	9.414	–0.255
3	24.250	24.850	2.415
4	42.297	44.042	3.985

Note the rather large discrepancy between the experimental and analytical frequencies for the first mode of the truss. The truss actually experiences Coulomb damping in the first mode, and this was not accounted for in the analytical model. Even so, it was determined that this discrepancy did not adversely affect the results of this study. Note also that Table 1 shows only the first four resonant frequencies of the truss. Higher modes are not shown since, for the purposes of determining the actuator's optimum tuning, only the first four modes were used and the analytical models were hence reduced to fourth–order systems.[2]

ANALYTICAL RESULTS

To conduct a study into the interactions between the truss and a passively tuned reaction mass actuator (RMA), the actuator's frequency and damping ratio had to be determined. Another paper[4] described a method to calculate the optimum tuning of an actuator attached to a multi–degree of freedom system. With the fourth–order mass and stiffness matrices from the finite element models, the RMA was optimally tuned to the analytical structure's second resonant frequency using the following equations:[4]

$$\bar{\omega}_{\mathrm{opt}} = k_{\omega} \left[\frac{\mu}{1 + \mu} \right] \tag{1}$$

$$\zeta_{\mathrm{opt}} = k_{\zeta} \left[\frac{3\mu}{8\ (1+\mu)^3} \right]^{1/2} \tag{2}$$

where μ is the ratio of the actuator mass to the modal mass of the second structural frequency, $\bar{\omega}_{\mathrm{opt}} = \omega_{\mathrm{opt}}/\omega_2$ is the optimum frequency ratio to tune the actuator (ω_{opt} is the optimum natural frequency and ω_2 is the frequency of the second structural mode), ζ_{opt} is the optimum damping ratio to tune the actuator, and k_{ω} and k_{ζ} are correction factors to obtain the optimum tuning.[2]

Figure 2 compares the frequency response functions of the baseline and optimally tuned structure from 6 to 12 Hertz, where baseline is defined to be the truss without the RMA. In this figure the effects of adding passive damping to the truss can readily be seen. The actuator, which is attached to the tip of the truss, introduces a new resonant mode to the structural characteristics in addition to altering the second mode of the truss. Specifically, the frequency of the second mode is shifted higher and the peak amplitude ratio is reduced. In fact, for the second mode, both peak amplitude ratios of the system response with passive damping are less than 14 percent of the structure's undamped peak amplitude ratio.

It is interesting to observe the actuator's effect on some of the other resonant modes of the 20–bay truss. Figure 3 compares the baseline and passively damped responses of the structure between 0 and 6 Hertz while Figure 4 shows the responses between 24 and 30 Hertz. Note that there are slight reductions in the peak amplitude ratios of the structure's first and third modes when the actuator, passively tuned to the structure's second resonant frequency, is attached to the 20–bay truss. In addition, with passive damping, the first resonant frequency is shifted lower while the third resonant frequency is shifted higher. Finally, for the structure's fourth resonant mode there is a slight reduction in the peak amplitude ratio and a shift to a slightly higher frequency with the passive actuator attached. However, the change in this mode's response is so small it can be neglected. Clearly, the analytical results show that attaching a RMA that is passively tuned to the second mode of the truss reduces not only the magnitude of vibration of the structure's second mode but also those of the first and third modes as well.

Figure 5 illustrates the sensitivity of the truss' structural characteristics if the RMA is not tuned to the optimum passive actuator parameters. Just a five percent mistuning of the actuator's optimum frequency causes noticeable shifts in the frequencies and peak amplitude ratios of the structure's second mode, as well as the new mode introduced by the actuator. However, the other resonant modes of the structure are not altered. Finally, it was observed that mis–tuning the actuator's optimum damping ratio by five percent produced almost no change in the structural characteristics of the 20–bay truss. Much larger errors in the tuning of the actuator's damping ratio introduce, of course, small differences in the peak amplitude ratios of the structure's resonant modes, but these differences are slight.

EXPERIMENTAL RESULTS

To verify the results obtained analytically, a reaction mass actuator assembly was attached to the tip of the 20–bay truss and the system was subjected to random vibrations from the air jet thrusters. The RMA assembly used for this study is shown in more detail in Figure 6. This assembly consisted of a moving mass core made up of a series of rare earth magnets encircling a stationary coil. A magnetic shaft, connected to this core, passed through the center of a stationary linear velocity transducer which sensed the velocity of the reaction mass relative to the truss. The RMA assembly also included a non–contacting transducer to sense the relative position of the reaction mass. Each of the resulting analog signals from the two transducers was multiplied by an appropriate gain constant and fed back to the RMA via a power amplifier to generate electromagnetic stiffness and damping on the RMA. In this manner the RMA's natural frequency and viscous damping factor could be varied by adjusting a pair of feedback gain potentiometers to their desired values. There was a small amount of uncontrollable rolling friction in the motion of the RMA shaft through two sets of linear ball bearings; the total damping of the RMA was, therefore, a combination of this small, but uncontrollable, friction plus the controllable and much larger linear viscous damping.[2]

With the RMA attached to the truss and tuned to the optimum values of the frequency and damping ratio specified analytically, a transfer function of the structure's resonant frequencies was obtained with the aid of a Tektronix model 2630 Fourier Analyzer. Comparisons between the baseline and optimally tuned structural responses are shown in Figures 7–9. Figure 7 shows the structure's frequency responses from 0 to 50 Hertz; Figure 8 shows the responses from 0 to 10 Hertz; and Figure 9 shows the responses from 20 to 30 Hertz. Overall, it can be seen from the figures that the addition of the passive actuator to the truss does indeed reduce the magnitude of the structural vibrations.

In the optimally tuned response of Figure 8 there is a noticeable absence of the two peaks associated with the structure's second mode and the mode introduced by the actuator. Although these peaks are visible in the analytical response of Figure 2, they are significantly less than the baseline peak amplitude ratio. Experimentally, these peaks exist but are much less than the baseline response and, consequently, do not appear in Figure 8. Thus, the passively tuned actuator eliminates, for the most part, any vibrations associated with the second mode. The structural responses in Figure 8 mirror the analytical responses in Figure 3 in that the peak amplitude ratio of the first structural mode is slightly less when the passively tuned actuator is attached to the truss. However, experimentally the optimally tuned response shows the first frequency to be slightly higher than the baseline response. This is the opposite of the analytical result shown in Figure 3, but considering how close the baseline and passively damped frequencies are for the first mode the differences are not significant. Numerical and experimental inaccuracies can account for the difference in the results shown in Figures 3 and 8.

Figure 9 shows that the passively–tuned actuator does reduce the structure's vibrations at higher frequencies, as the peak amplitude ratio of the third mode is reduced by 59 percent. In addition, the resonant frequency is shifted higher, validating the analytical results of Figure 4. Finally, as shown in Figure 7, there is very little difference in the structural responses of the baseline and passively damped structures for the higher modes.

Based on the above observations, both the experimental and analytical results show that a RMA, passively tuned to the second resonant frequency of the 20–bay truss, reduces the flexible structure's vibrations. Specifically, adding a passive actuator to a flexible structure reduces the magnitude of the structure's vibrations, not only at the resonant mode to which the actuator is tuned, but at higher and lower modes as well. Selective tuning of the actuator produces the largest decreases in the magnitude of the structure's vibrations, but the beneficial effects of passive damping are not limited to the frequency of interest.

To experimentally observe the effects a mis–tuned actuator has on the structure's modal responses, the RMA was tuned to 95 percent and 105 percent of the optimal frequency. The results, shown in Figures 10 and 11, show that even a slight mis–tuning of the actuator's frequency causes noticeable shifts in the structure's first three resonant modes as well as the mode of the actuator. Notice, though, that higher resonant modes are not altered much. These results are similar to the analytical results summarized in Figure 5.

THE EFFECTS ON THE SETTLING TIME

One important aspect in reducing or eliminating a structure's vibrations is the time it takes to eliminate those vibrations due to an impulse disturbance. The settling time for the motion of the 20–bay truss can be determined experimentally by exciting the first three structural modes. A signal generator was used to produce a sine voltage to the air jet thrusters at one of the first three experimental natural frequencies shown in Table 1. The pair of thrusters located at the tip excited modal motion in the uncontrolled resonant mode of vibration corresponding to the desired excitation frequency. The signal amplitude was arbitrary, and when the motion reached a suitably high level the air jets were turned off, allowing the modal motion to gradually decay under the influence of inherent damping only.[3]

Time history decay traces for the first three modes of the truss in the uncontrolled configuration are displayed in the left–hand column of Figure 12. Decay traces of the truss with the RMA passively tuned to the structure's second resonant frequency are shown in the right–hand column of Figure 12, with the approximate time of when the thrusters are turned off marked by an arrow.

By comparing the baseline and passively damped decay times, it is readily apparent that the passively–tuned RMA does indeed reduce the settling time of the truss. There is a slight reduction in the settling time for the first mode; it takes about 19.5 seconds for the truss to settle without any passive damping while the addition of the RMA reduces this time to 14.5 seconds. The second mode shows a significant reduction in the settling time with the RMA attached to the truss. Without the passively tuned RMA the settling time for the second mode of the truss is again about 19.5 seconds. With the RMA attached to the truss this settling time is only 4 seconds. The settling time for the third mode is reduced from 9.5 seconds to 3.2 seconds when the RMA is added to the truss. Thus, the addition of the passively tuned RMA is quite effective in reducing the time it takes for the truss to settle after being excited in the first three resonant modes. It should be noted that the structure's higher resonant frequencies were not excited due to limitations in the electronics associated with the air jet thrusters.

CONCLUSIONS

It has been demonstrated that a passively tuned reaction mass actuator can be used to successfully reduce the vibrations of a flexible truss. The 20–bay truss was subjected to both random and sinusoidal excitations, and the RMA, when tuned to the structure's second resonant frequency and attached to the tip of the truss, reduced the magnitude of vibration and the settling time of the second resonant mode. In addition, the magnitudes of vibration and settling times of the first and third modes were also reduced when the actuator, still passively tuned to the structure's second resonant frequency, was attached to the tip of the truss. Selective tuning, then, produces the best results, yet the beneficial effects of passive damping is not limited to the frequency of interest. Finally, mis–tuning of the actuator's frequency does alter the structure's modal responses, particularly around the frequency to which the actuator is passively tuned.

The next step in this research effort is to utilize both active control and passive damping on the 20–bay truss. Since passive damping is an effective means of vibration suppression, it is hoped that a combination of active control and passive damping will produce a beneficial control system for flexible structures.

ACKNOWLEDGEMENTS

This work was sponsored by the US Air Force Academy and the US Office of Scientific Research. The authors are grateful for the contributions of Captain Jeff Turcotte and Mr Jim Smith.

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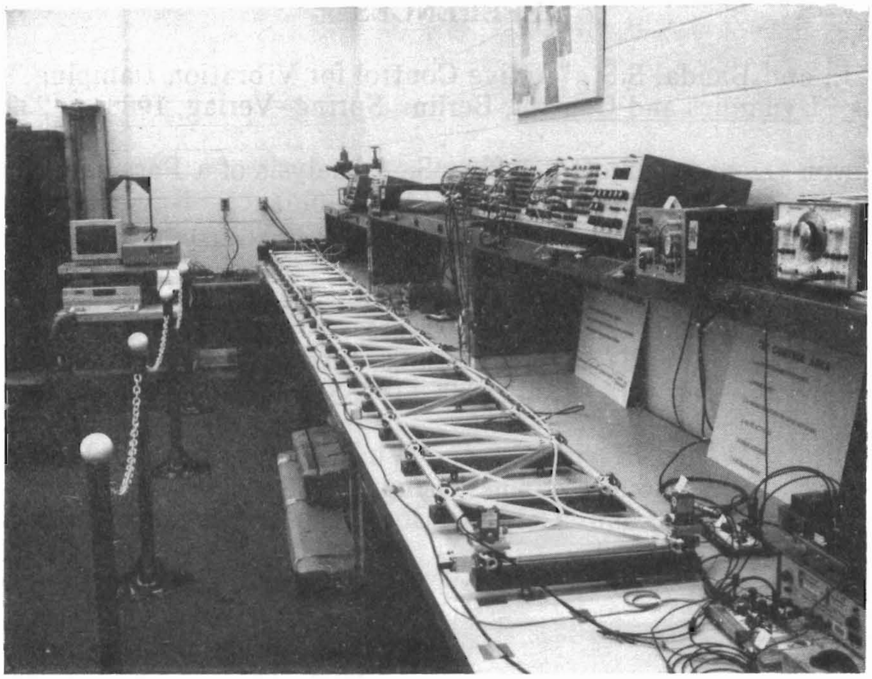

Figure 1. The Planar Truss

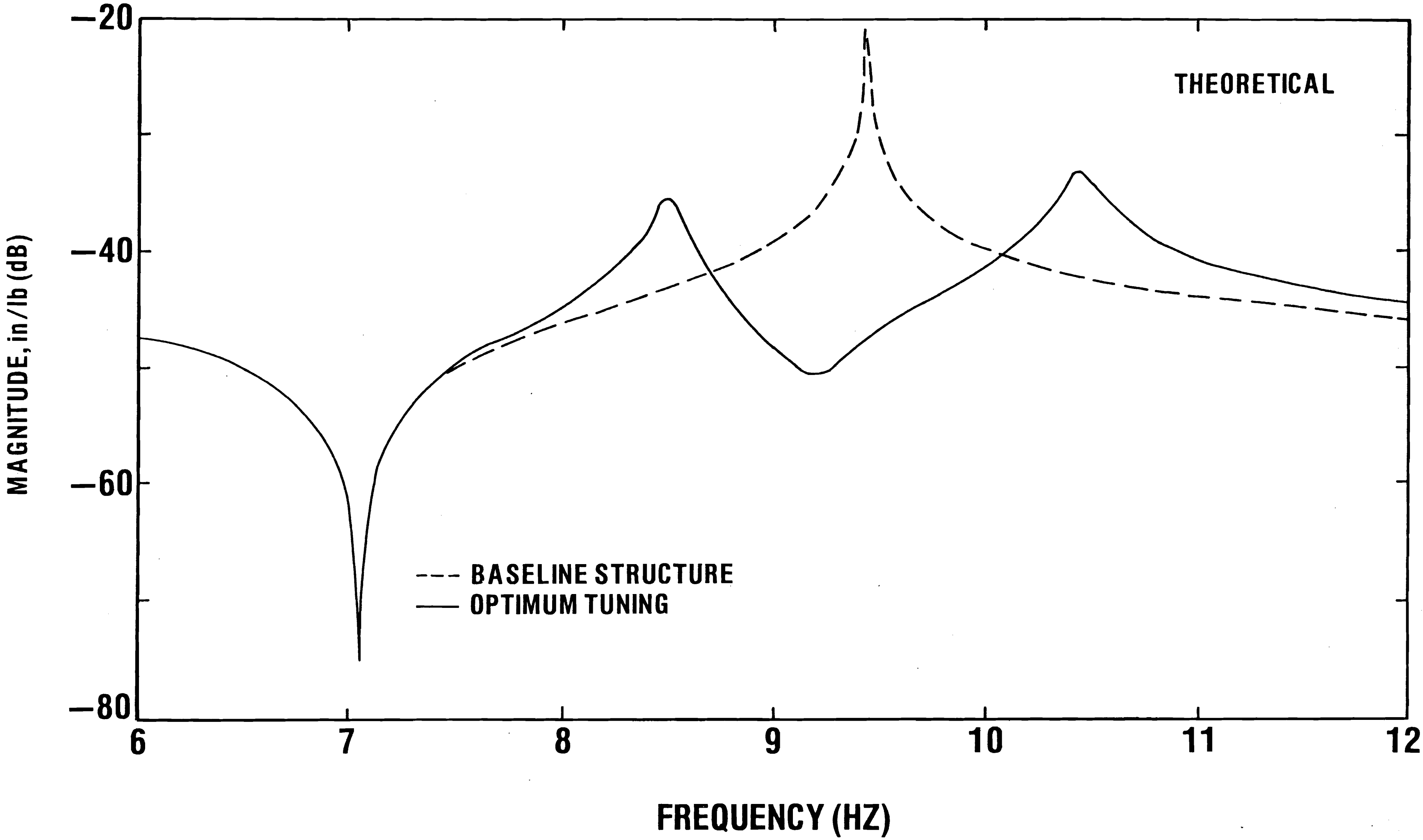

Figure 2. Analytical Baseline and Optimally Tuned Structural Responses from 6 to 12 Hertz

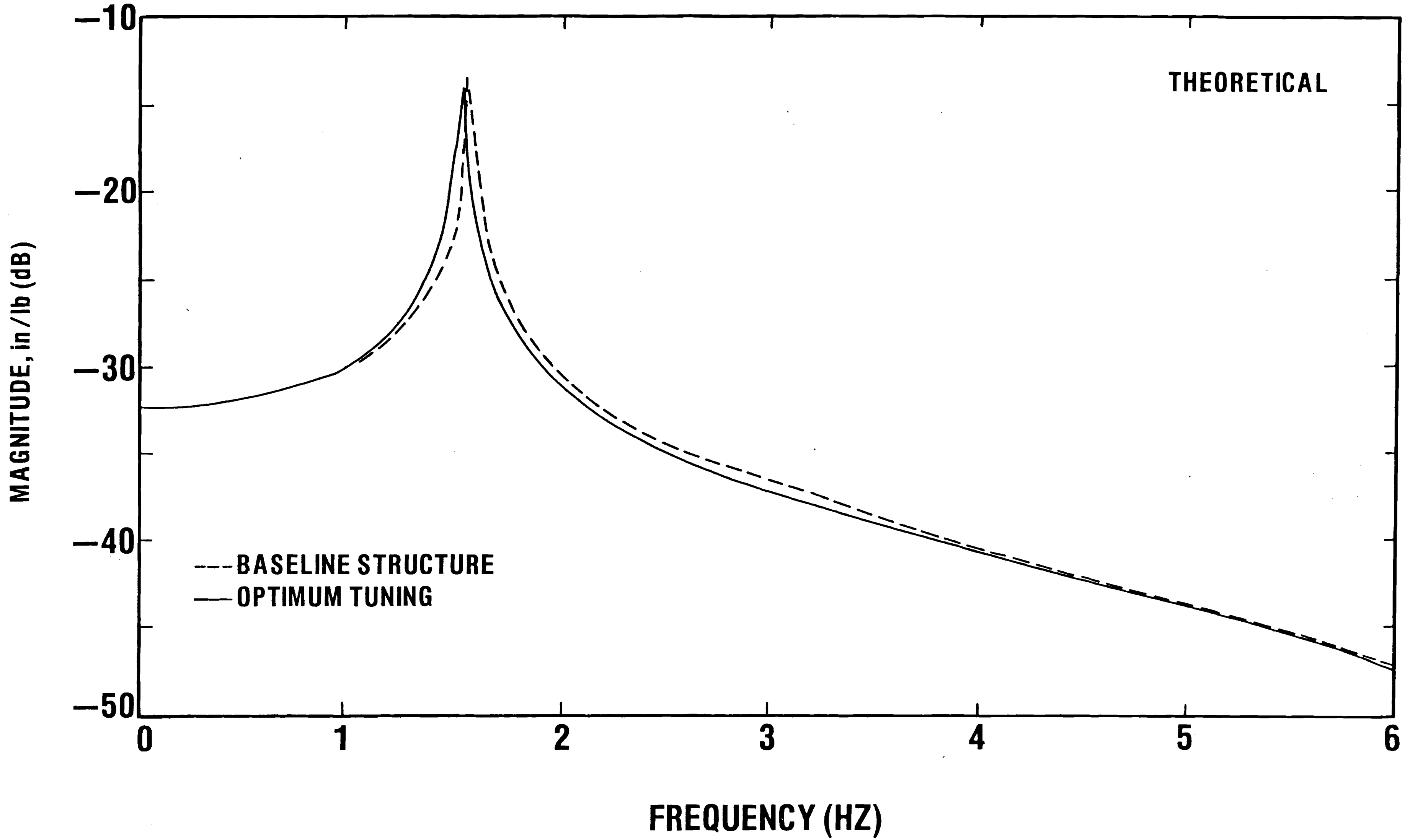

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Figure 3. Analytical Baseline and Optimally Tuned Structural Responses from 0 to 6 Hertz

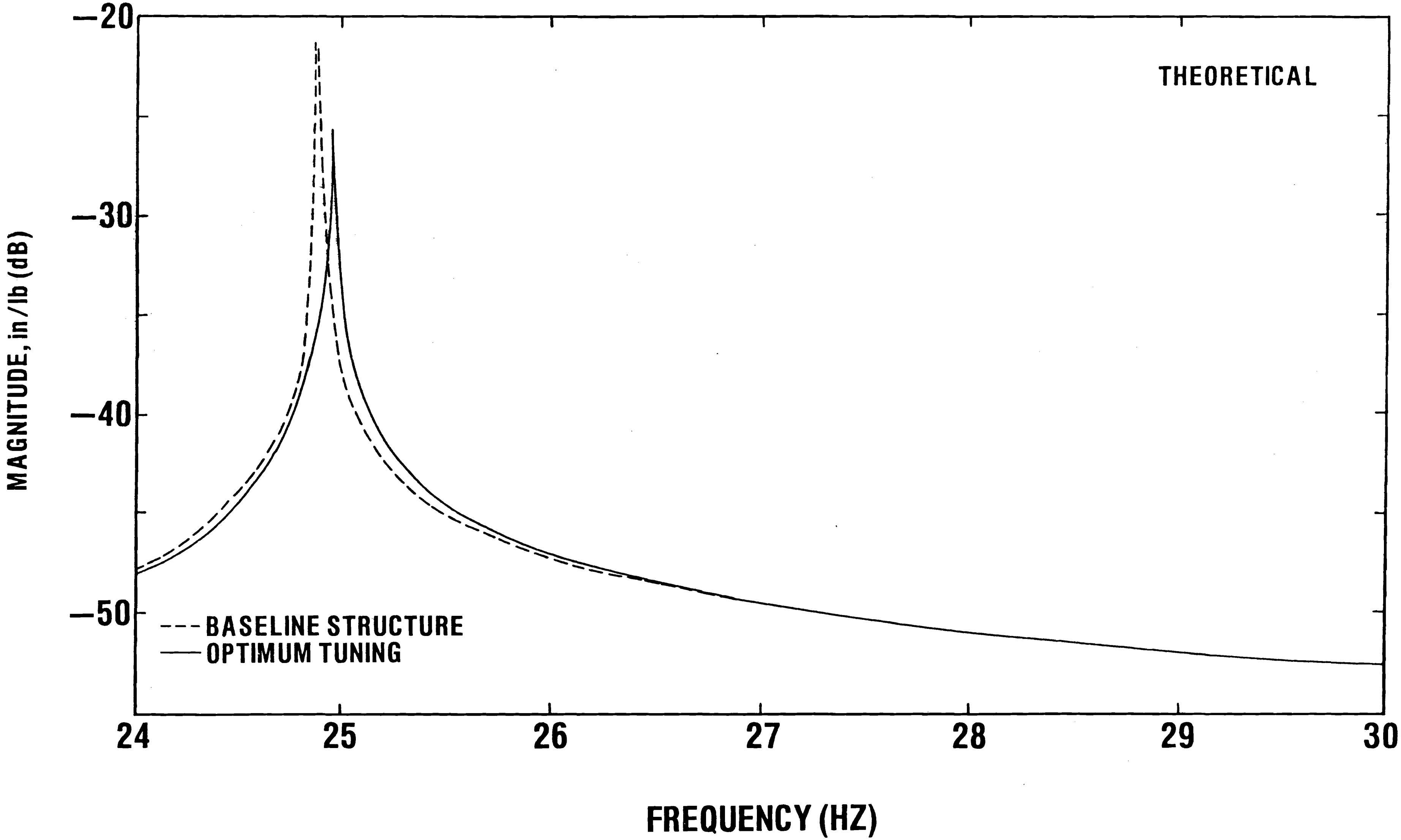

Figure 4. Analytical Baseline and Optimally Tuned Structural Responses from 24 to 30 Hertz

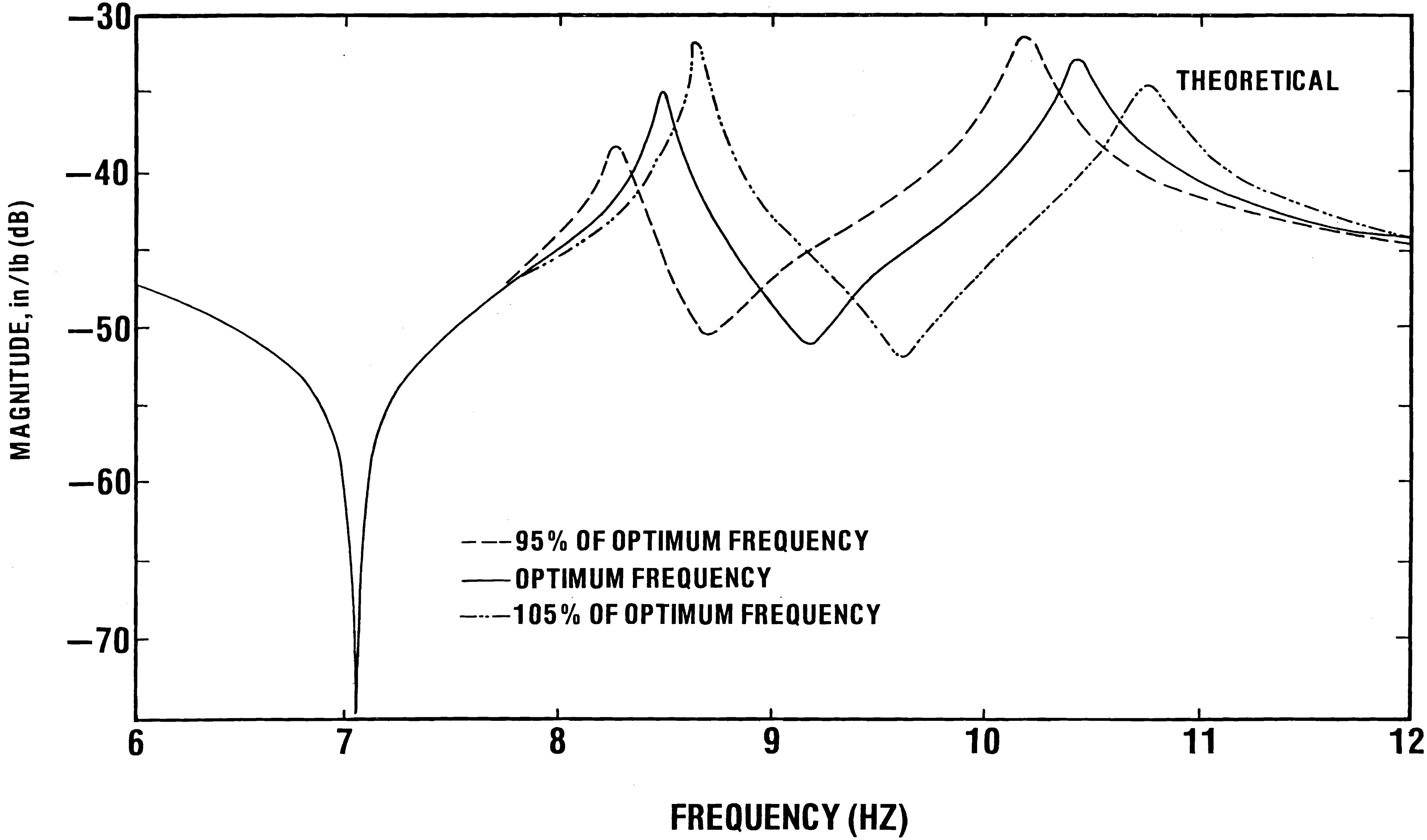

Figure 5. Analytical Optimally Tuned and Mis–Tuned Structural Responses

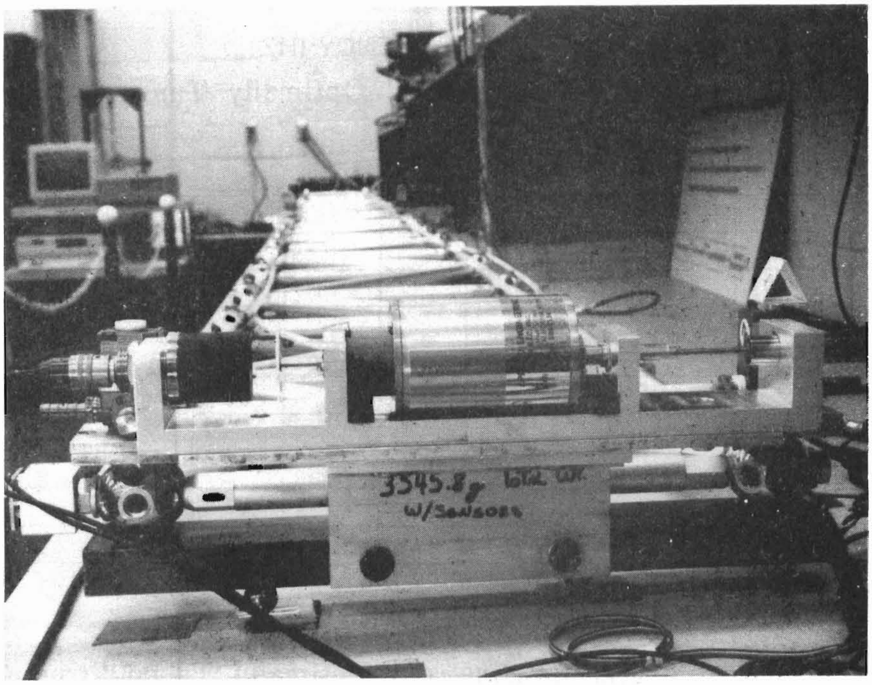

Figure 6. Reaction Mass Actuator Assembly

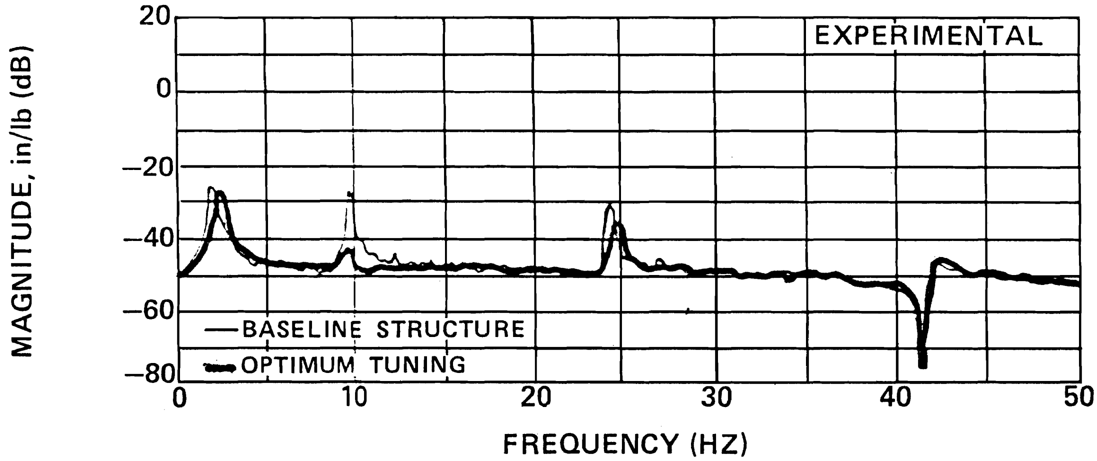

Figure 7. Experimental Baseline and Optimally Tuned Structural Responses from 0 to 50 Hertz

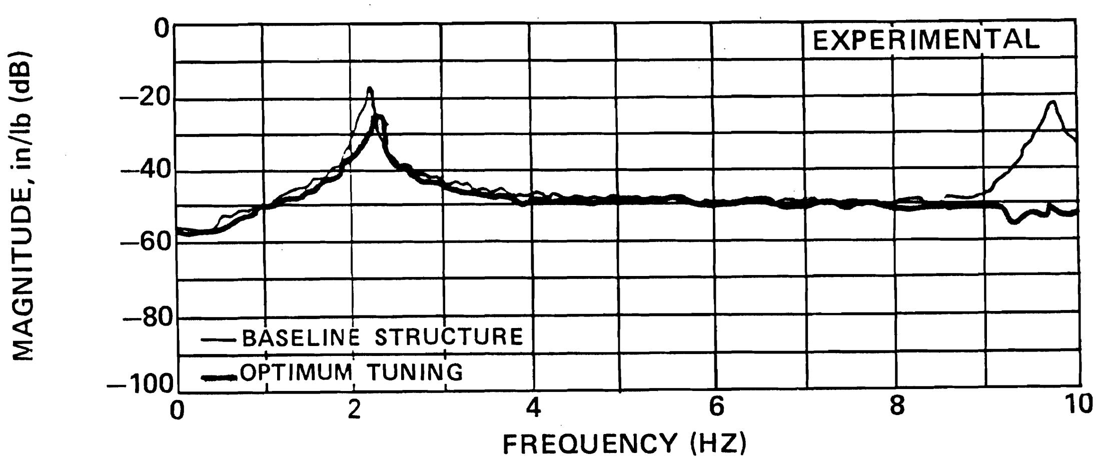

Figure 8. Experimental Baseline and Optimally Tuned Structural Responses from 0 to 10 Hertz

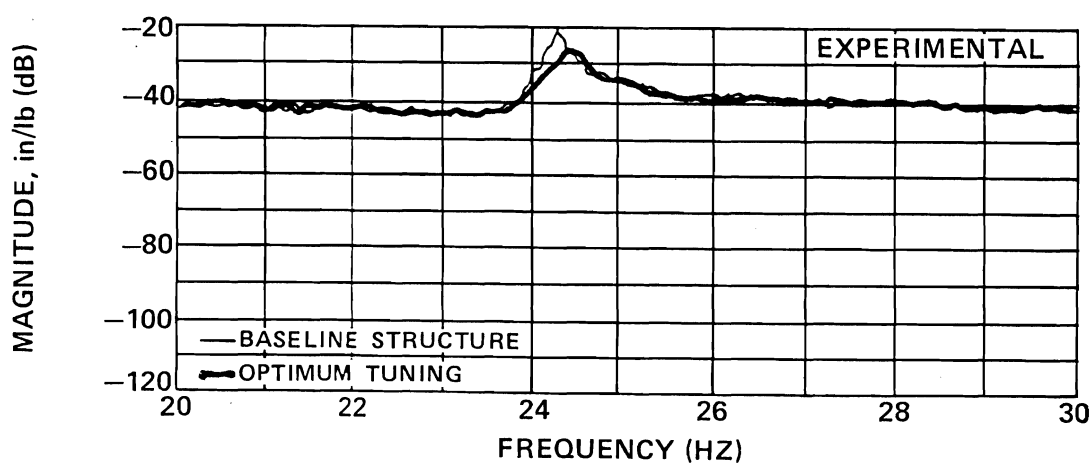

Figure 9. Experimental Baseline and Optimally Tuned Structural Responses from 20 to 30 Hertz

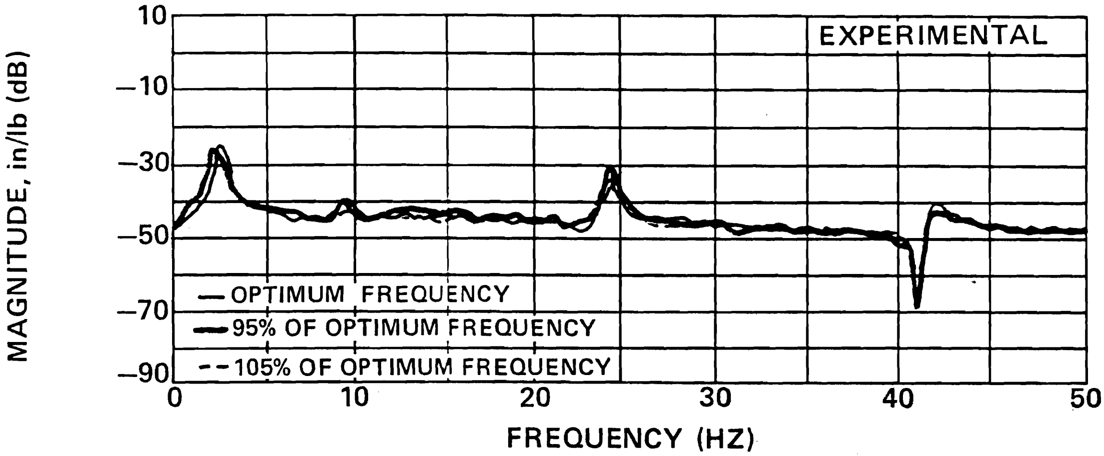

Figure 10. Experimental Optimally Tuned and Mis–Tuned Structural Responses from 0 to 50 Hertz

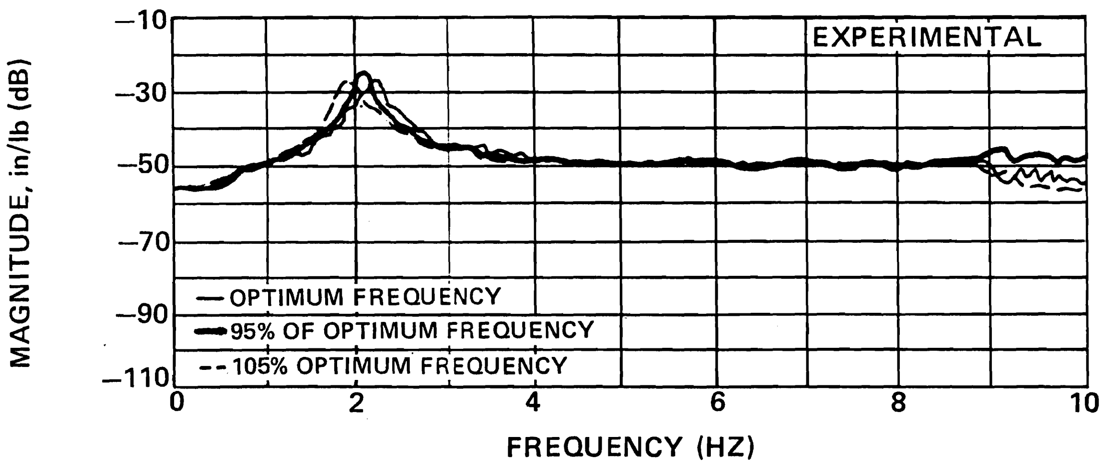

Figure 11. Experimental Optimally Tuned and Mis–Tuned Structural Response from 0 to 10 Hertz

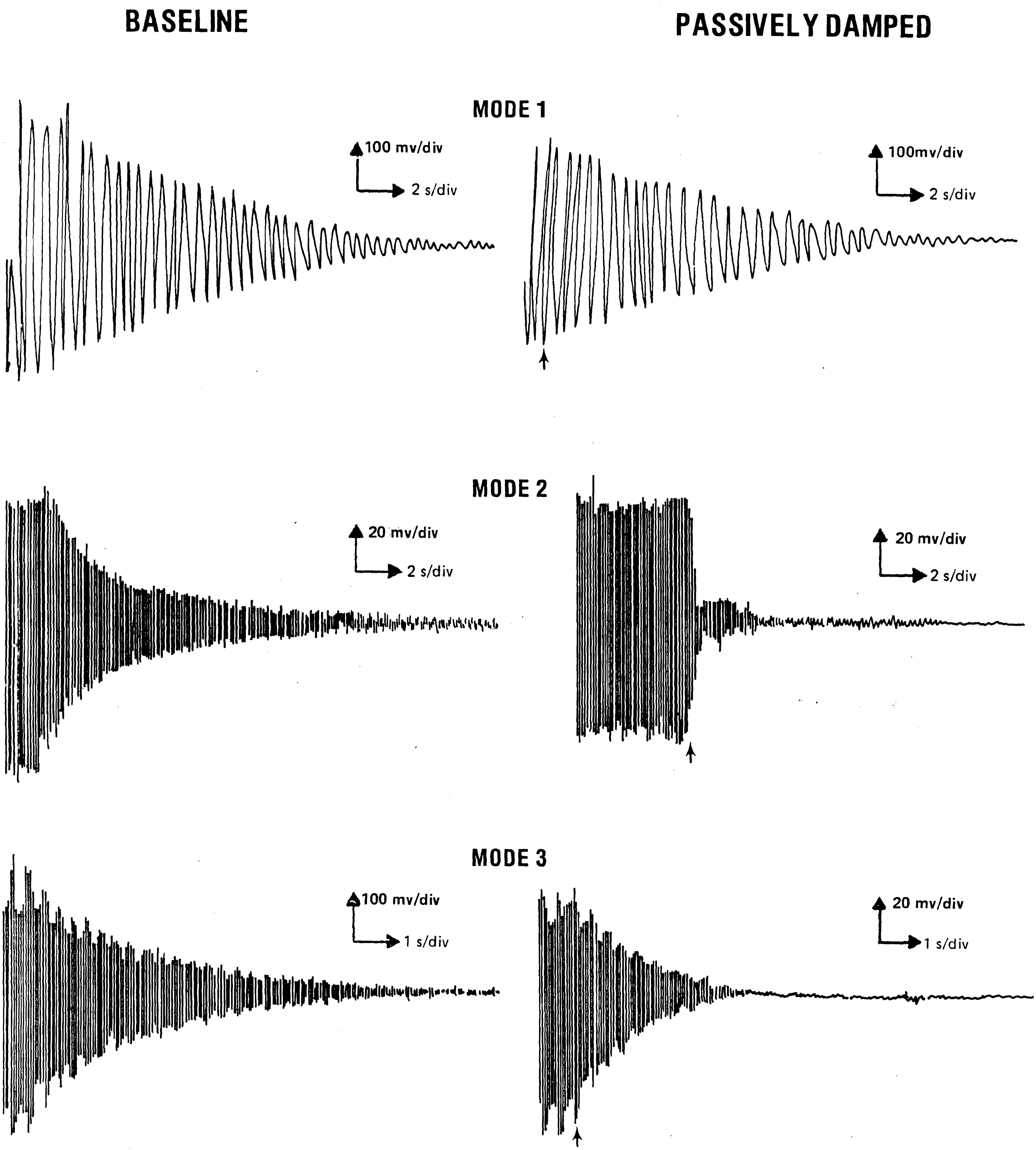

Figure 12. Experimental Baseline and Passively Damped Modal Decay Plots of Truss Tip Transverse Velocity

*U.S. GOVERNMENT PRINTING OFFICE: 1991/548-076/44406