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AN EXPLORATORY INVESTIGATION OF THE TUNED DYNAMIC ABSORBER

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ABSTRACT

An experimental investigation of applying tuned dynamic absorbers to reducing the mechanical vibration amplitudes of both low and high "Q" resonant systems was undertaken. An approach to establishing the relation of underwater radiated sound to damped panels was also initiated.

INTRODUCTION

GENERAL NOISE-REDUCTION METHODS

The three principal methods of reducing noise originating in systems are vibration isolation, sound absorption, and vibration damping. Vibration isolation is utilized in various forms (e.g., mounts and couplings) each designed to limit the vibrational energy from the vibrating source which reaches the structure or enclosure. Sound absorption is obtained with materials capable of reducing the sound-pressure level in a compartment by transforming airborne sound into heat energy. Vibration-damping treatments are utilized on solid structures to transform the mechanical vibrations of the system to heat energy.

In submarine structures all three noise-reduction methods are indispensable: vibration isolation, to reduce direct excitation of the communicating hull by the vibrational energy of the source; sound absorption, to provide more comfortable living areas for the crew and provide for some reduction of hull excitation; and vibration damping, to reduce the amplitude of vibrations present in the hull or communicating structural components thus reducing the subsequent hull radiation.

VIBRATION DAMPING

The vibration-damping properties inherent in structural materials such as steel are very poor indeed. Thus some other material must be applied to the steel member to compensate for its inherently low hysteresis losses. Some common types of vibration-damping materials include mastic, asphalted felt, fibrous glass blankets, etc., all of which reduce the energy of the vibrating structural member by mechanisms such as frictional action at the surface of application, flexural motion of the fibers, inertial property of the treatment, and frictional and viscous dissipation within the treatment.

In the fabrication of such an item as a hull, sufficient changes in the impedance characteristics of a continuous section of the hull, plus the environmental impedance variations at the interface with the surrounding media, form many sources of reflection for mechanical vibrations reaching the hull. At resonant frequencies these reflections are periodic and exhibit a standing-wave pattern of nodes and antinodes. The common vibration-damping treatment using the above noted types of materials consists of a "blanket" or total coverage of the vibrating area. Some treatments are aimed at covering only the antinodal areas. These methods of treatment become increasingly important as the vibrating area becomes either more complex in configuration, so that application of the above treatments is unfeasible, or surrounded by media in which the common treatment materials deteriorate and lose their effectiveness.

THE TUNED DYNAMIC ABSORBER

Under such circumstances, a tuned dynamic absorber may offer a solution, provided that it reduces vibration at resonance, and demonstrates a considerable weight advantage over conventional surface-coverage treatments. Basically, the tuned dynamic absorber is designed to execute a large vibrational motion, thus dissipating the energy of the vibrating system it is applied to, and thus reducing its amplitude. A comparison of the typical effect of an overall treatment and that of a tuned dynamic absorber is shown in Fig. 1. In practice, the tuned vibration absorber is so constructed that the stiffness-to-mass ratio is chosen to correspond to the square of the natural frequency of the main mass. For instance, the simply constructed absorber shown in Fig. 2 has a total mass formed by both the mass of the metallic and nonmetallic elements of the absorber, while the stiffness of the natural rubber stopper practically governs the stiffness of the absorber. By selection of the proper mass, the square root of the stiffness-to-mass ratio can be adjusted to the natural frequency of the main mass. Now as the plate vibrates, the absorber also vibrates, but out of phase with the main mass. In addition, the viscoelastic properties of the rubber serve to transform the vibrational energy into heat energy. Both of these factors tend to reduce the plate's vibrational energy. The rate of energy transformation, as well as the rate at which vibrational energy is supplied to the main mass, will determine the equilibrium of the system. Thus this equilibrium point is extremely important at the natural frequency of the main mass, and indeed controls its maximum amplitude of vibration.

DYNAMIC ABSORBER APPLICATIONS

Some probable areas of application of tuned dynamic absorbers include nonstructural bulkheads, including those bulkheads comprising fuel and water tanks, radar-antenna supports, and sonar domes. For the noise reduction of resonating submerged plating, the tuned dynamic absorber is ideal, since it can readily be sealed and thus be made impervious both to the surrounding medium and to the pressure fluctuations of the changing fluid depth—factors which hamper the effectiveness of common extended coverage materials. For resonating radar-antenna supports, which are relatively massive, a tuned dynamic absorber of greater mass than one normally visualized for application to nonstructural plating could be utilized. Again the advantage of the absorber over the extended treatment is that it can be sealed and that it therefore is not subject to destruction by any shear forces encountered. The tuned dynamic absorber could possibly be utilized advantageously for noise reduction on resonating sonar domes. In addition to its fluid-tight integrity, its application entails a smaller addition of total mass to the dome than would a conventional extended treatment. Thus it is possible that the dynamic absorber would offer less attenuation to the transmitted and received signals. Furthermore, once the configuration of a sonar dome has been selected for certain ideal characteristics, the spot applications of relatively light-weight dynamic absorbers might not alter these characteristics as much as the application of an extended coverage treatment.

In addition, a survey of the value of structural and nonstructural damping aboard submarines to reduce both self noise and far-field noise brings to light the controversial issue concerning the applicability of damping to massive hull structures exhibiting low "Q" resonances to reduce the vibration amplitude and consequent radiation into the sea. It was this issue which served as the basis for an experimental program designed to provide a quantitative comparison of the damping efficiencies of tuned dynamic absorbers applied to both high and low "Q" vibrating systems. As is noted in the summary, the program was not considered complete at the termination of the work period.

EXPERIMENTAL

OBJECTIVES

The major objectives of the experimental work were:

A. To demonstrate the effectiveness of dynamic absorbers (spot dampers) on both high and low "Q" plates utilizing the thick-plate test apparatus (Fig. 3);

B. To evaluate the effect of an internal damping treatment (including the dynamic absorbers) applied to a steel plate enclosed as one end of a massive cylinder in relation to the level of radiated energy in both air and water; and

C. To provide a submersible dynamic absorber model, if both A and B proved rewarding.

THE EFFECT OF TUNED DYNAMIC ABSORBERS ON BOTH HIGH AND LOW "Q" VIBRATING SYSTEMS

Instrumentation.—The initial experimental effort was directed toward changing the information-display segment of the decay-rate apparatus from an oscilloscope-camera unit to a chart-recorder unit. This was accomplished allowing faithful reproductions of decay rates ranging upwards of 800 db/sec and covering an amplitude range of some 50 db. Figure 4 illustrates the newly modified experimental apparatus.

Decay-Rate Experiments.—An initial attempt to evaluate the dynamic absorber (spot damper) utilized a plate having a natural frequency of 281 cps and an absorber developed previously (see Fig. 5). One of the primary aims was to determine the effectiveness of the absorber on relatively low "Q" (100-300) vibrational systems.

From the decay rates obtained for the 281-cps, 15-3/8-in. square plate, the "Q" of the plate was determined to be approximately 2700. By applying four layers of 3M-36PSL152311L82-2C3 damping tape, 6 in. square, to the center portion of the underside of the plate, and small rectangles of Fiberglas (7-3/4-in.-by-5-1/2-in.) plus a septum to the corners of the plate, the decay rate increased to 60 db/sec and the "Q" dropped to approximately 128. The application of the absorber further increased the decay rate to 200 db/sec and lowered the "Q" to 38. However, these data, although qualitatively indicative, were not considered a suitable basis for succeeding experiments. The absorber was unreliable in performance, and the use of the dynamic absorber resulted in a highly damped, two-degrees-of-freedom behavior, thereby invalidating the usual thick-plate decay-rate measurement, for which the damped test plate must still behave like a single-degree-of-freedom, relatively high "Q" system.

Steady-State Experiments.—Consequently, the steady-state response of the test plate to pure-tone excitation was investigated in a frequency range bracketing the resonant range of the plate-plus-absorber system. In addition, a simple dynamic absorber consisting of a rubber plug and an attached mass (which was variable) replaced the unreliable absorber previously mentioned. The apparatus shown in Fig. 4 was used once more, this time with a pure-tone frequency scan of 1 decade per 26.67 minutes. For convenience, the 129-cps, 24-1/8-in. square plate of mass 55.9 lb was used. From the decay-rate data taken on the bare plate (these data being valid under single-degree-of-freedom conditions), the "Q" of the plate was calculated to be approximately 3500. Similar measurements of the plate treated with four 7-3/4-in.-by-5-1/2-in. rectangles of Fiberglas and septum placed at each corner yielded an approximate "Q" rating of 200. This treatment is shown in Fig. 6. Armed with two plates, with a wide variation in "Q" values, it was possible to conduct a continuous spectrum analysis of the vibratory characteristics of each of these systems singly, and with the additional application of a simple dynamic absorber. The absorber in this case consisted of a No. 11 natural rubber stopper weighing 69 grams and an attached optimum mass of 236.8 grams. In all cases this absorber was cemented to the plate's center. A schematic of the absorber is shown in Fig. 2.

Figure 7 illustrates the excellent results obtained from these experiments. Figure 7a is a record of the response of the bare plate to pure-tone excitation from 100 to 400 cps. Several salient features may be noted. First of all, the primary mode at approximately 130 cps stands out clearly. Obviously, the peak shown here is probably not the maximum which one could expect even with manual tuning. However, it will do as a reference level. That the system has a high "Q" and therefore low internal dissipative forces is illustrated by the heavily marked high-frequency side of the 130-cps peak. This shading is due to the energy retained by the plate at its natural frequency beating with that energy imparted at the driving frequency. Other smaller peaks occur in the vicinity of 260-270 cps and at 350-370 cps. The dynamic absorber was now cemented to the plate with the results shown in Fig. 7b. As was expected, the 130-cps peak was completely obliterated and replaced by two much smaller humps at approximately 122 and 145 cps, both practically merging into noise. The peaks at 260-270 cps were also eradicated, the peaks showing at 350-370 cps were unchanged.

Fiberglas and septa were then applied to the bare plate (as mentioned above) with the ensuing results noted in Fig. 7c. First of all, the major peak at 130 cps, although somewhat depressed, is still prominent. However, the peak value recorded is likely to be real inasmuch as no beating phenomena are apparent. Furthermore, the peaks appearing at 260-270 cps and 350-370 cps in Fig. 7a also appear in Fig. 7c. This system now exhibits the characteristics of a low "Q" system. The dynamic absorber was now cemented to the damped plate, resulting in the dynamic changes as shown in Fig. 7d. A comparison of Figs. 7b and 7d show that, for all practical purposes, the end results obtained by applying the dynamic absorber to the high "Q" plate and the low "Q" damped plate are almost identical: the peaks occurring at 130 cps and 260-270 cps are no longer apparent. However, no significant reductions occurred at 350-370 cps. The results are tabulated in Table I.

Further efforts aimed at designing an absorber for these upper peaks were unsuccessful and subsequent examination of the plate supports showed that these vibrations emanated from the spring mountings. In short, it was shown that a tuned dynamic absorber is effective upon application to both high and low "Q" systems.

RELATION OF INTERNAL DAMPING TREATMENTS TO RADIATED ENERGY

Instrumentation.—To determine the effect on the radiated noise spectrum of an "enclosed" plate as a function of an internally applied damping treatment, the apparatus shown in Fig. 8 was assembled. The test plate, a 1/8-in.-thick steel plate, 14 in. in diameter, was clamped to the open end of a massive double-walled steel cylinder and driven through a range of 200-1600 cps by an 8-in. speaker carrying a 20-cps-bandwidth tone warbled at the rate of 32 cps. The internal and external sound-pressure levels were monitored, and their difference signal recorded. The use of a warble tone reduced the possibility of standing waves at the internal microphone position, while the utilization of the reverberation room and the rotating vane eliminated any standing-wave phenomena at the externally located hydrophone position.

Other factors remaining constant, the acoustic radiation from an object depends on its amplitude of vibration. In the cylindrical tank experiment with an internal acoustic drive, the amplitude of the plate vibration is conveniently measured by recording the transmission of energy through the plate. In these experiments, the points of major interest were those where maximum plate vibration or maximum transmission occurred, namely, at resonance.

After the reproducibility of the measurement system had been ascertained, data were recorded for three experimental procedures, all conducted in an air medium:

1. A recording of the transmission characteristics of the bare plate.

2. A transmission recording of the plate with an internally applied asphalted felt-septum damping treatment.

3. A transmission recording of the plate with an attached simple tuned dynamic absorber.

Transmission-Measurement Experiments.—Figure 9a shows the transmission record obtained with the bare plate. The major resonant peak is noted at approximately 470 cps coincident with the first mode of vibration, while the second mode resonance appears at 1400 cps. From a comparison between the results obtained at the relatively low scan rate and the manual scan at the major resonance, it was noted that a substantial difference in the peak values, some 8 db, was observed. This is due to the high "Q" of the undamped plate and the ensuing critical resonant frequency.

Figure 9b shows the transmission record obtained with a total surface coverage of the inside face of the test plate with the asphalt-felt damping treatment. The asphalt-impregnated felt (Ruberoid No. 260) was first cemented with 3M weatherstrip adhesive to the test plate, and a septum of 20-gauge sheet metal was then cemented to the felt. A substantial reduction of the major resonance-peak transmission level was noted and no difference in peak levels was obtained with either the manual or driven frequency scan. We were now dealing with a lower "Q" system.

Figure 9c denotes the transmission record obtained with a simple tuned dynamic absorber applied to the center of the test plate. The absorber consisted of a No. 4 natural rubber stopper weighing 12.9 grams cemented to the plate and an additional 1.5-gram weight cemented to the face of the stopper. From the recording, the large reduction of the major transmission peak can once more be noted. The results of the three transmission tests are illustrated in Table II. From these results it may be noted that approximately 27-db reductions of the level of the transmitted sound at the primary modal frequency are obtained by both the asphalted-felt-septum and the tuned-dynamic-absorber treatments. However, the total absorber mass measures only 3.4% of the felt-septum treatment.

SUMMARY AND CONCLUSIONS

The exploratory experiments carried out to determine the applicability of the tuned dynamic absorber as a device for the reduction of mechanical vibrations have yielded favorable results on vibratory systems ranging in "Q" values from 3500 to 200. Similar results can thus be expected when the absorber is used on similarly dynamic systems. The advantages of the tuned dynamic absorber as compared to extended treatment methods for reducing vibration amplitudes are its tunability, its relatively light weight, and its adaptability to surfaces marked by various protrusions and indentations.

Initial experiments conducted in an air medium showed that effective reductions in radiated noise emanating from an "enclosed" plate vibrating at its resonant frequency can be obtained by the application of an internal damping treatment. The same degree of reduction was obtained using an extended coverage treatment as was obtained using a tuned dynamic absorber. However, the absorber mass weighed only 3.4% of the extended coverage treatment. Unfortunately, curtailed contract time and funds did not allow completion of the correlation of internally located damping treatments with associated underwater acoustic radiation and the development of a stable practical tuned dynamic absorber.

TABLE I

RESPONSE AT THE MAJOR RESONANCE OF THE 129-CPS TEST PLATE, BARE AND TREATED, TO A TUNED DYNAMIC ABSORBER

Test-Plate Condition	Resonance Amplitude (db)	Reduction (db)
(1) Bare (Q = 3500)	51	
		33
(2) Tuned Absorber Attached	18	
(3) Fiberglas-Septum Treatment (Q = 200)	41	
		25
(4) Same as (3) with Tuned Absorber Attached	16	

TABLE II

RELATIVE TRANSMISSION VALUES AT THE PRIMARY MODAL FREQUENCY (APPROXIMATELY 470 CPS) FOR THE TEST PLATE WITH VARIOUS DAMPING TREATMENTS

Test-Plate Condition	Resonance Amplitude (db)	Reduction (db)
(1) Bare plate	63	
		25.5
(2) Asphalted Felt-Septum Treatment (wt. 425 gm)	37.5	
(3) Tuned Dynamic Absorber (wt. 14.4 gm)	37	26

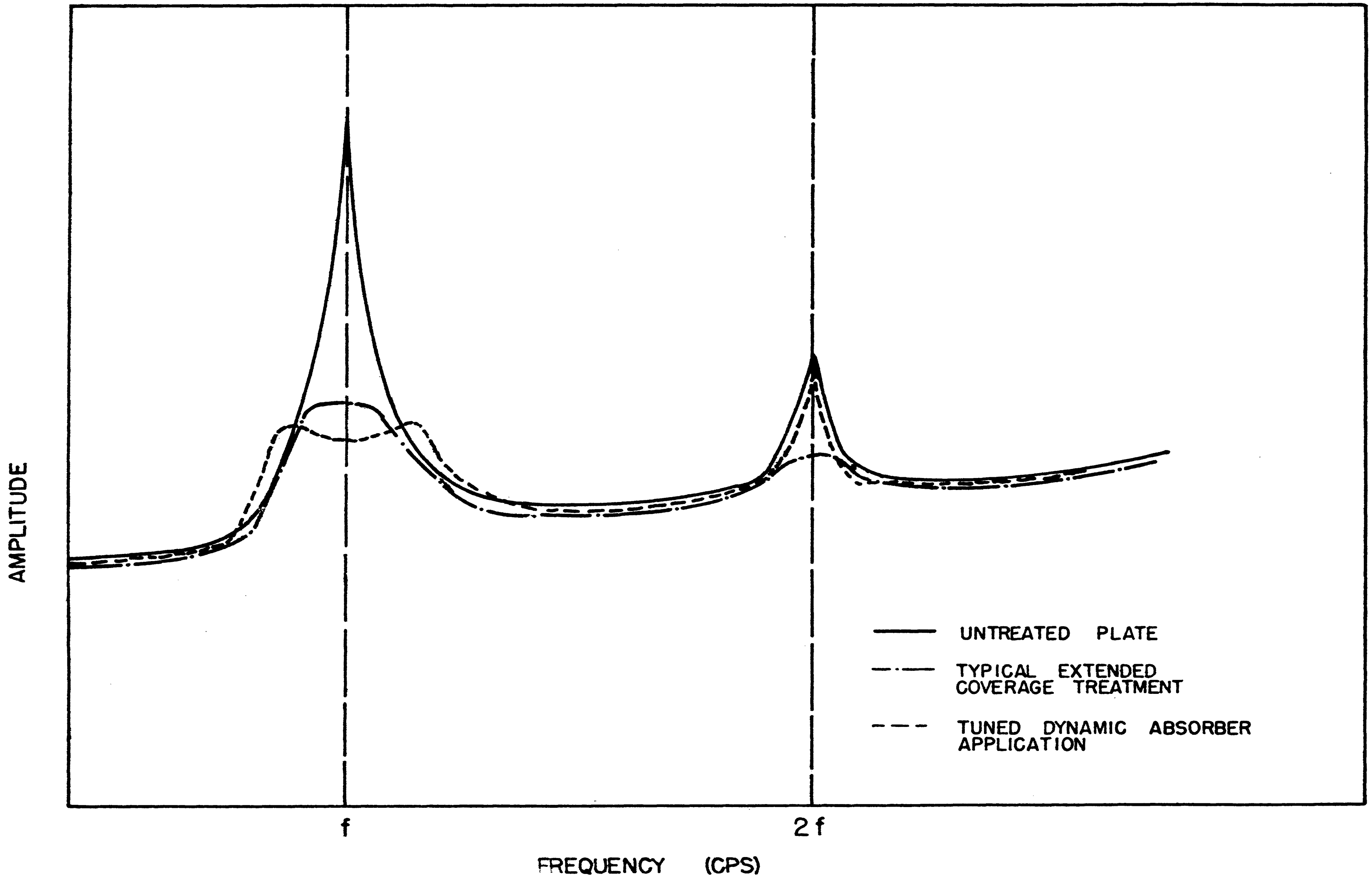


FIGURE 1. COMPARISON OF EXTENDED COVERAGE AND TUNED DYNAMIC ABSORBER BEHAVIOR.

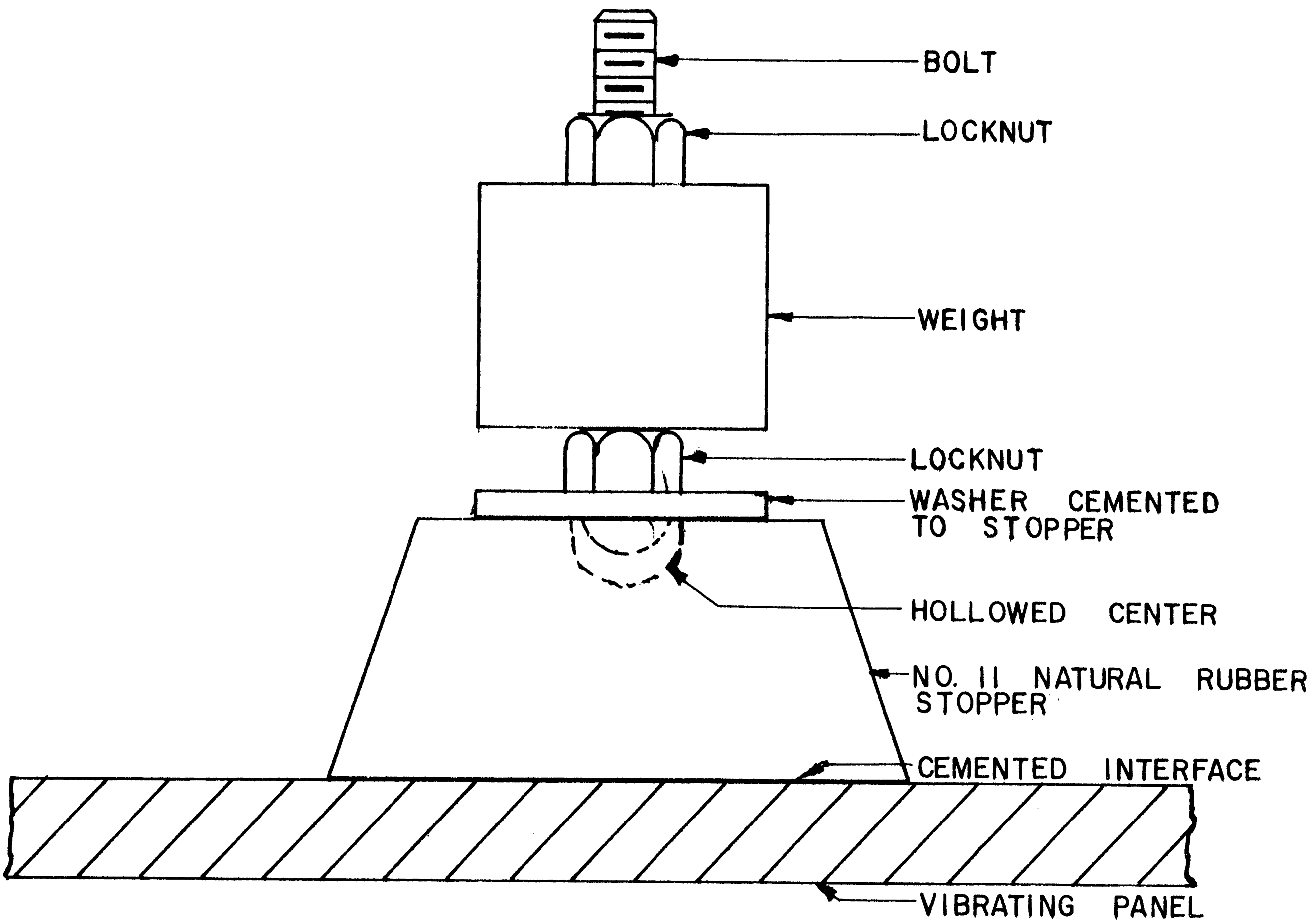


FIGURE 2. SIMPLE TUNED DYNAMIC ABSORBER.

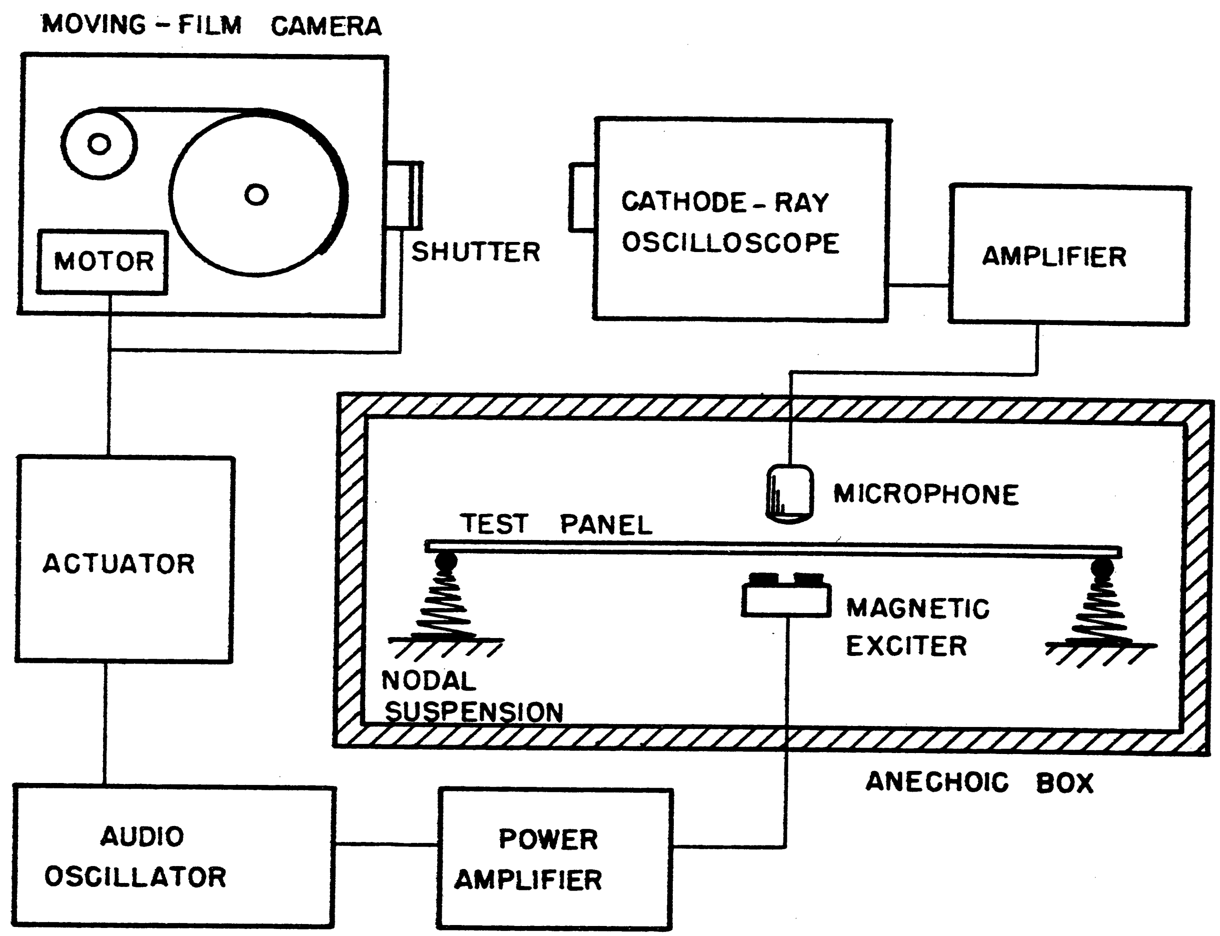


FIGURE 3. ORIGINAL THICK PLATE DECAY - RATE MEASUREMENT APPARATUS.

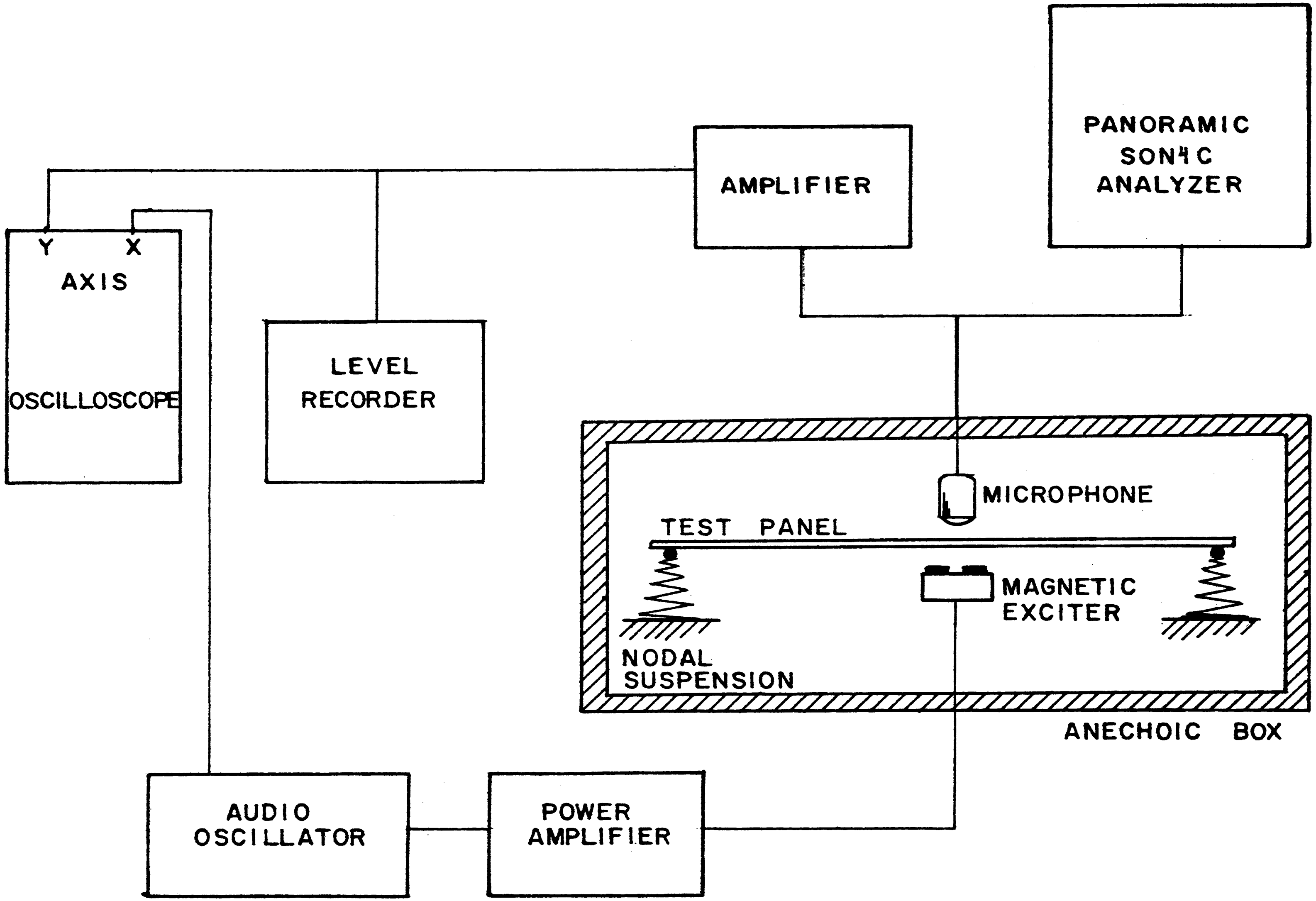


FIGURE 4. MODIFIED THICK PLATE DECAY – RATE APPARATUS.

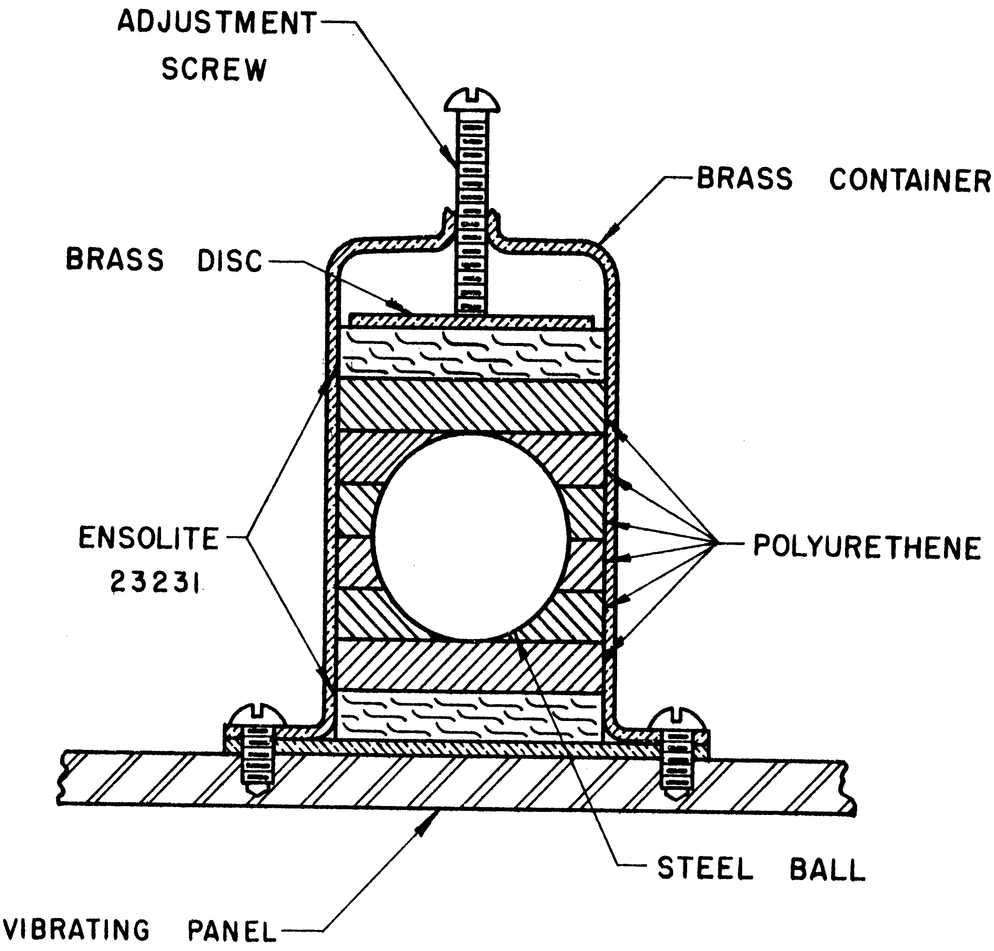
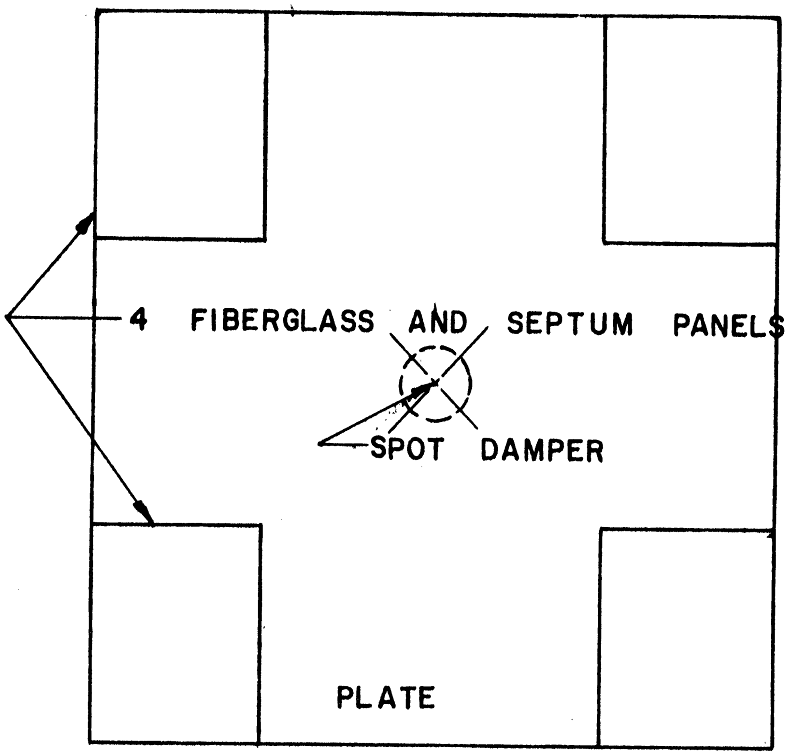
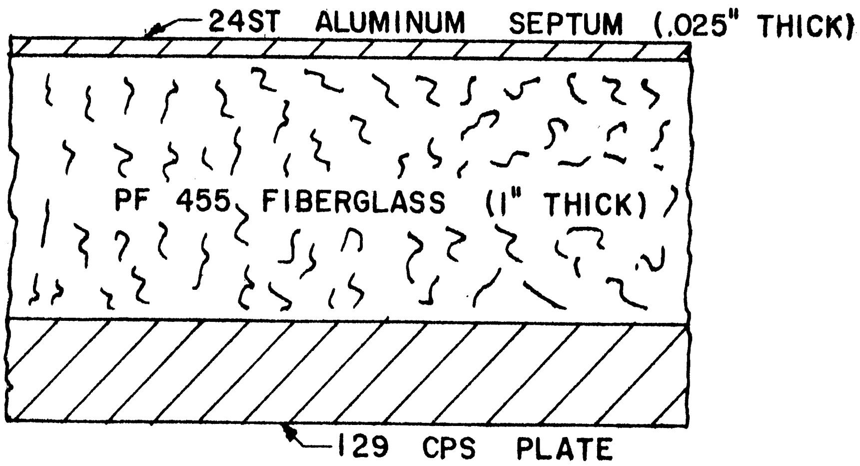


FIGURE 5. INITIAL MODEL OF SUBMERSIBLE DYNAMIC ABSORBER.





CROSS SECTION OF TREATMENT

FIGURE 6. FIBERGLASS-SEPTUM DAMPING TREATMENT USED TO LOWER THE Q OF THE 129 CPS TEST PLATE.

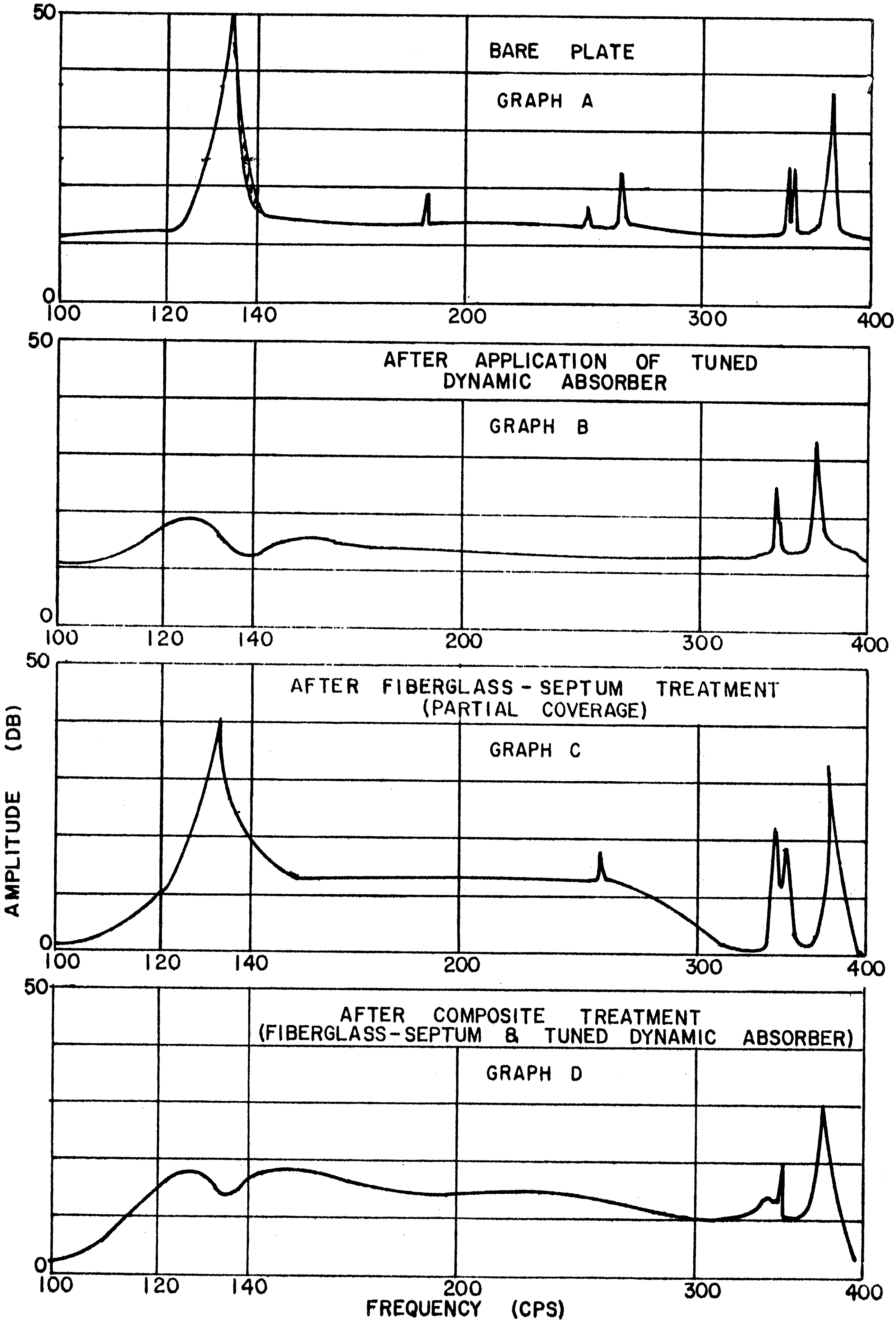


FIGURE 7. RESPONSE OF THE 129 CPS TEST PLATE TO PURE TONE SCANNING.

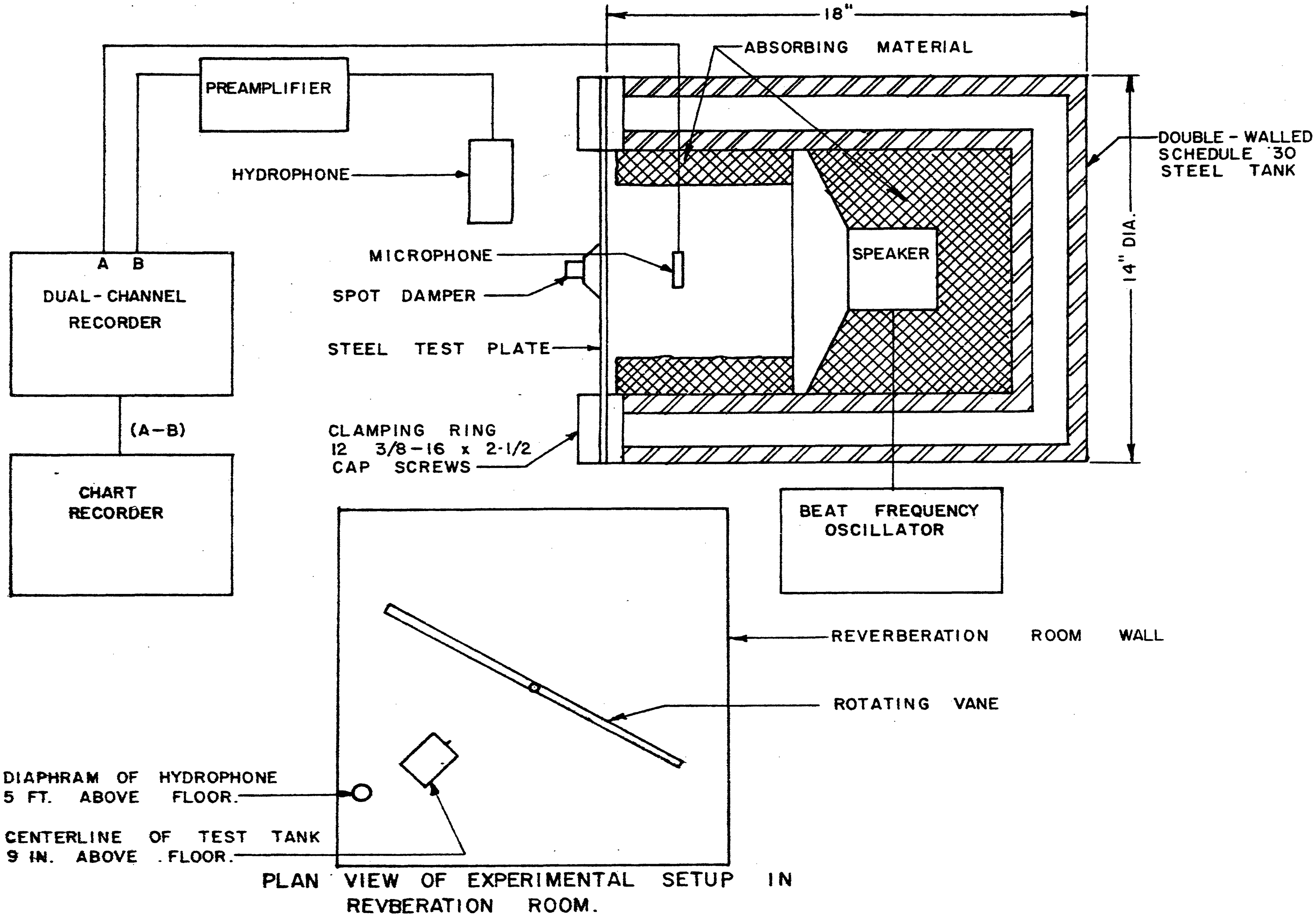
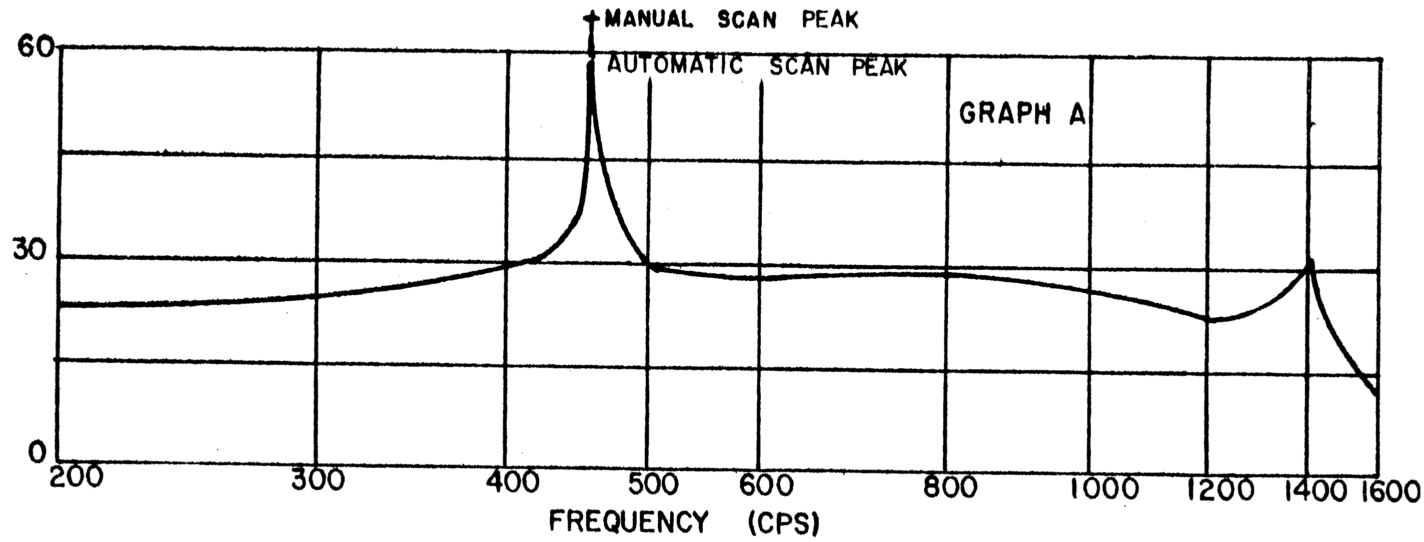
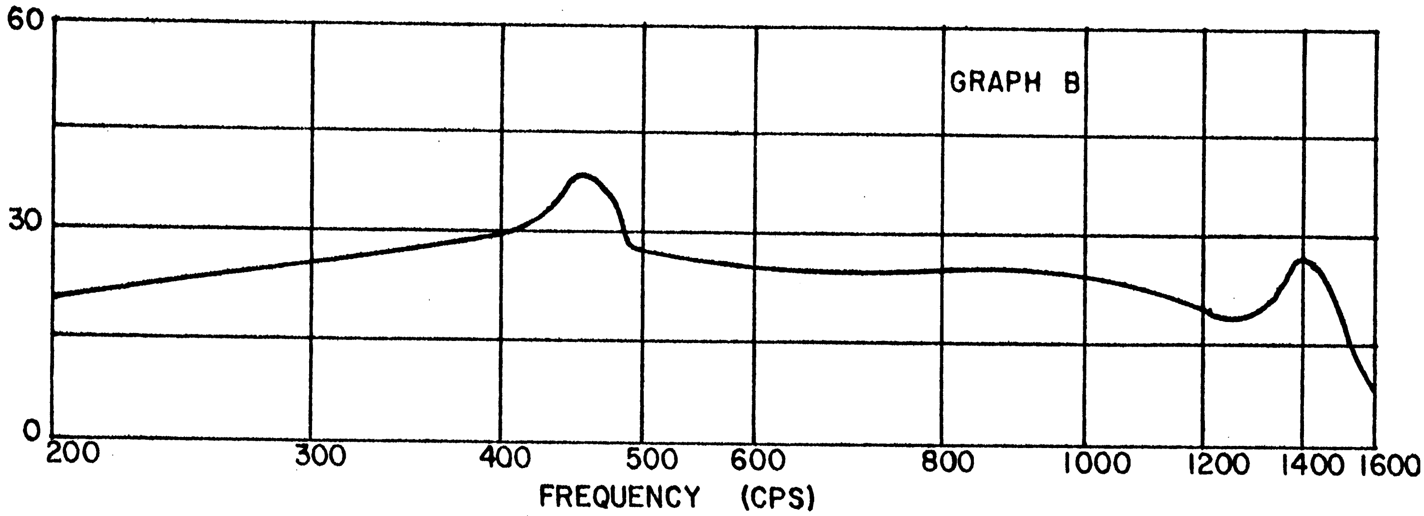


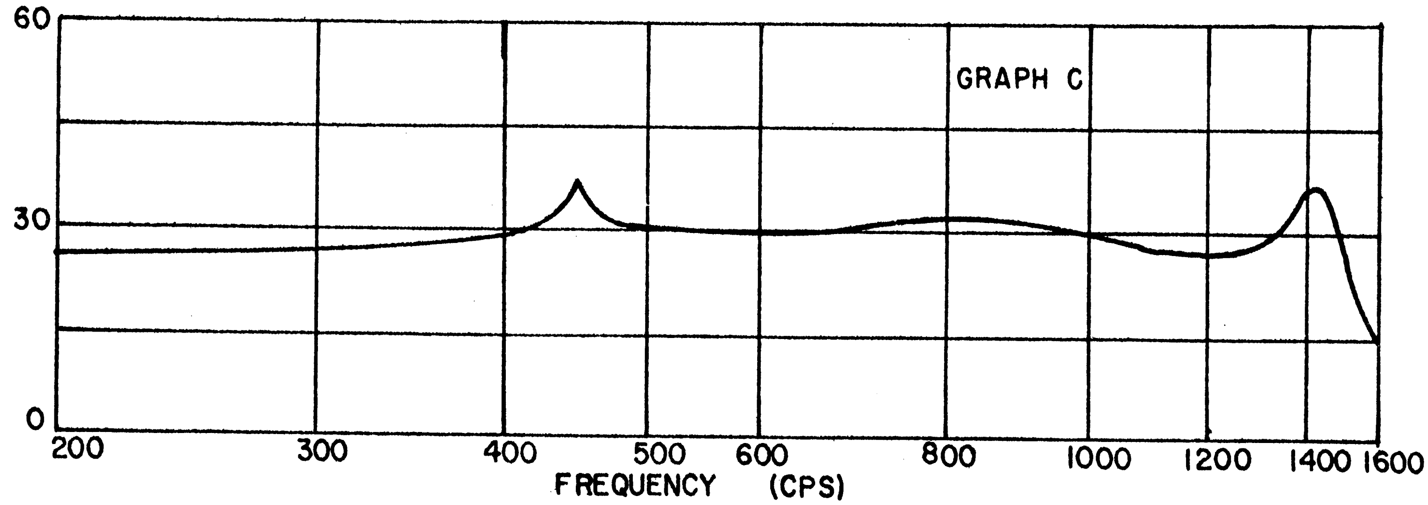
FIGURE 8. APPARATUS TO DETERMINE THE EFFECT ON THE ACOUSTIC RADIATION EMANATING FROM AN ENCLOSED PANEL BY APPLICATION OF INTERNAL DAMPING TREATMENTS.



BARE PLATE



AFTER ASPHALTED FELT – SEPTUM TREATMENT
(TOTAL SURFACE COVERAGE)



AFTER APPLICATION OF TUNED DYNAMIC ABSORBER

FIGURE 9. TRANSMISSION RECORD OF TEST PLATE.

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