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STRUCTURAL MATERIAL DAMPING FOR PACKAGING ELECTRONIC COMPONENTS

TECHNIQUES USED FOR MISSILE FUZE COMPONENTS CAN BE ADAPTED FOR MANY PACKAGING APPLICATIONS

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ABSTRACT

Viscoelastic shear damping is being used by the Naval Ordnance Laboratory, Corona, California, to control the mechanical resonances of structures housing guided missile fuze components. Four experimental structures were fabricated using cast aluminum, laminated aluminum, laminated fiberglass, and aluminum honeycomb. They were subjected to acceleration, shock, temperature and humidity, and salt-spray tests. This report describes the housings and presents test results.

ACKNOWLEDGMENT

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FOREWORD

A program was conducted in 1963 and 1964 at the Naval Ordnance Laboratory, Corona, California, to establish the relative merits of different types of structures for packaging the electronic circuits of target detecting devices for guided missiles and to identify the most promising design approach.

This report describes a series of tests performed to evaluate the efficacy of various combinations of housing and plate assemblies. The results have led to an advanced packaging technique, which may also prove to be of general interest to other designers of packaging in various missile and aerospace programs.

The work was authorized by WEPTASK RMMO-21-030/211-1/F009-08-01.

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INTRODUCTION

The vibrations caused by modern aerospace propulsion systems and ultrasonic flight produce resonance responses in structures, instrumentation, and equipment over an extended frequency range. Since such resonance responses can cause structural fatigue or malfunctions of critical components, they have long been of prime concern to those designers who are charged with packaging the electronic components of missiles.

In 1963 and 1964 a program was undertaken at the Naval Ordnance Laboratory, Corona, California, to ensure that packaging of the electronic circuits of target detecting device components of missile fuzes will afford adequate protection against all the severe environments which they could conceivably experience during missile flight and storage. Because packaging designs in use at that time, and such fabrication techniques as rigidization, detuning, and decoupling, did not meet this requirement, new techniques were sought and the use of structural material damping was developed.

Besides reducing structural vibration response, this structural material damping technique has been found to have the following advantages: (1) it provides an advantageous volume-to-weight ratio, and (2) it cuts down packaging fabrication costs. It also reduces the vibration amplification of the structural resonances of equipment, whereas vibration isolators merely reduce the environmental vibration that is transmitted to the equipment.

The tests that were performed at this Laboratory to evaluate the various housing and plate assemblies designed experimentally for packaging the electronic components consisted of (1) a vibration response test, (2) a steady-state flight acceleration test, (3) a temperature and humidity test in accordance with MIL-STD-354, and (4) a 96-hr salt-spray test in accordance with MIL-STD-356.

The materials used for the experimental housing and plate assemblies were (1) sand-cast aluminum for both housing and plate (Fig. 1); (2) sheet aluminum for the housing and multilayers of sheet aluminum laminated with viscoelastic compound for the plate (Fig. 2); (3) fiberglass for the housing and fiberglass laminated with viscoelastic compound for the plate (Fig. 3); and (4) honeycomb aluminum for the housing and Butyl-isolated aluminum honeycomb for the plate (Fig. 4). Each housing was finished in accordance with the applicable MIL-STD.

PHILOSOPHY

In the development of electronic fuze packages, it is usually necessary to base the design on estimates of environmental parameters. This is normal for programs involving missiles and space equipments, since the true environmental parameters cannot be determined until hardware has been designed, built, and tested in flight. Because of the nature of the vibration response in undamped structural materials, a gross failure in estimating vibration environments might cause component failures in localized areas. However, the response of damped structures to vibrations of widely varying frequencies and energy levels is more nearly constant and predictable than is that of similar undamped structures and therefore the risks inherent in estimating vibration environments can be greatly reduced if structural material damping is employed in packaging the electronic components.

As used herein, structural material damping refers to the energy-dissipation properties of materials or systems under cyclic stress whereby energy is dissipated within the vibratory system, in most cases by conversion of mechanical energy to heat. The damping here of interest arises from slip and other boundary shear effects at interfaces. Energy dissipation during cyclic shear strain at an interface may occur as a result of strain in a separating adhesive, such as a viscoelastic layer between interfaces (Ref. 1).

Structural material damping is a manifestation of the inelasticity of the structural material. Stress as a function of strain in a sheet of metal, for instance, is represented by a single curve—a hysteresis loop, pointed at the ends, that doubles back upon itself—but the curve of laminated material forms an elliptical hysteresis loop (Ref. 2). When a bending wave travels through a multilayer plate in which the central or viscous layers are relatively soft, the deformation will be either pure bending or shear bending. In shear bending, the viscoelastic layers undergo a shearing action, whereas metal or other layers undergo pure bending. The theory of bending waves in such a system has been developed by Kerwin (Refs. 3, 4, and 5). He showed that when the wavelength is long, the deformation is pure bending, but as the frequency is increased, a steady transition toward shear bending occurs. Thus, at low frequencies the bending stiffness of a multilayer plate corresponds with that of a thick nonlaminated plate, but as the frequency increases, the bending stiffness decreases toward the lower values representative of the combined stiffness of the nonviscous layers, which are bending simultaneously and with the same radius of curvature. The damping of the composite plate arises from the shear distortion of the viscoelastic layers.

Although the relative damping factors of certain structures can be determined theoretically, and estimates of the relative vibration and noise control can be provided in terms of the relative damping factors, how much damping material is required in a given vibration-control application is best determined experimentally. A few accepted rules and simple computations should guide the trial-and-error determination of an effective combination (Ref. 6).

TEST SETUP AND RESULTS

For the vibration response test, each of the four housings described above was assembled with 10 lb of mock electronic modules (various size aluminum blocks) mounted to its plate. Two triaxial accelerometers (Fig. 5) were attached to the mock hardware to measure the excitation vibration and equipment response. One accelerometer was mounted near the center of each plate, and one near the edge. No appreciable differences in frequency and transmissibility were registered at the two locations.

The No. 8 screws used for assembling the modules to the plates were tightened to 10 in. lb, and the five 1/4-in. screws used for assembling the plates to the housings were torqued to 20 in. lb. The holes drilled to accept the Rivnuts[1] used in assembling the modules to the viscoelastic-laminated plates allowed 0.020-in. minimum clearance so that the laminations could slip relative to one another during vibration or dynamic action.

The four assemblies were hard-mounted to a Ling Model A174 electrodynamic shaker (Fig. 6) capable of maintaining a constant acceleration over a broad frequency range (5 to 5000 cps) for harmonic vibration excitation in the lateral (X) and pitch (Y) axes. The assemblies were later similarly mounted on a Ling Model C-25 shaker (5 to 3000 cps) for vibration in the thrust (Z) axis. The assemblies were submitted to the harmonic vibration excitation input of 5 g from 20 to 3000 cps in three mutually perpendicular axes.

Typical curves showing transmissibility as a function of frequency for the four housing and plate assemblies are shown in Figs. 7-18. The four housing assemblies are further compared as to their volume-to-weight ratio and resonant transmissibility ratio in Table 1.

The findings of the above-described vibration response test were then corroborated by vibrating (1) the fiberglass housing assembled with the multilayer viscoelastic-laminated sheet aluminum plate (Figs. 19-21) and (2) the sheet aluminum housing assembled with the viscoelastic-laminated

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[1] B. F. Goodrich product.

TABLE 1. Volume-to-Weight Ratios and Resonant Transmissibility

Housing	Plate	Wt. (lb)	Vol. (cu in.)	Ratio of Vol. to Wt. (cu in./lb)	Resonant Transmissibility
Cast aluminum	Cast aluminum	11	330	30	4 to 55
Sheet aluminum	Viscoelastic-laminated aluminum	7	350	50	2 to 9
Fiberglass	Viscoelastic-laminated fiberglass	5.25	350	65	2 to 9
Honeycomb aluminum	Butyl-isolated honeycomb aluminum	4.9	350	70	8 to 25

fiberglass plate. No marked differences in transmissibility or resonant frequencies were noted; also, no great differences in transmissibility were registered by the two accelerometers. In general, the response curves of transmissibility for the multilayer viscoelastic-laminated aluminum plate reflected a slightly higher gain but narrower resonant bandwidth than those for the viscoelastic-laminated fiberglass plate, although the resonant frequencies were similar. Figure 22 shows the vibration responses of the sheet aluminum housing (1) when combined with the viscoelastic-laminated fiberglass plate and (2) when combined with the multilayer viscoelastic-laminated sheet aluminum plate.

In addition to the vibration tests, all the assemblies were subjected for a minimum of 5 sec to a steady-state flight acceleration of 40 g in the Z-axis and 28 g in the X- or Y-axes. All combinations responded successfully.

Standard temperature and humidity tests (MIL-STD-354) were performed on all the housing and plate assemblies. They were also subjected for 96 hr to salt spray (MIL-STD-356). After these tests, inspection showed no structural damage and no deterioration of the dynamic quality of the viscoelastic compound.

CONCLUSIONS

On the basis of the tests performed with the several experimental structures that were designed for protecting the electronic components of fuzes, it is concluded that viscoelastic shear damping (structural damping) can be employed with considerable success to control the vibration response of the structural resonances of equipment. This packaging technique is, in fact, currently used successfully for the electronic circuits of missile fuzes.

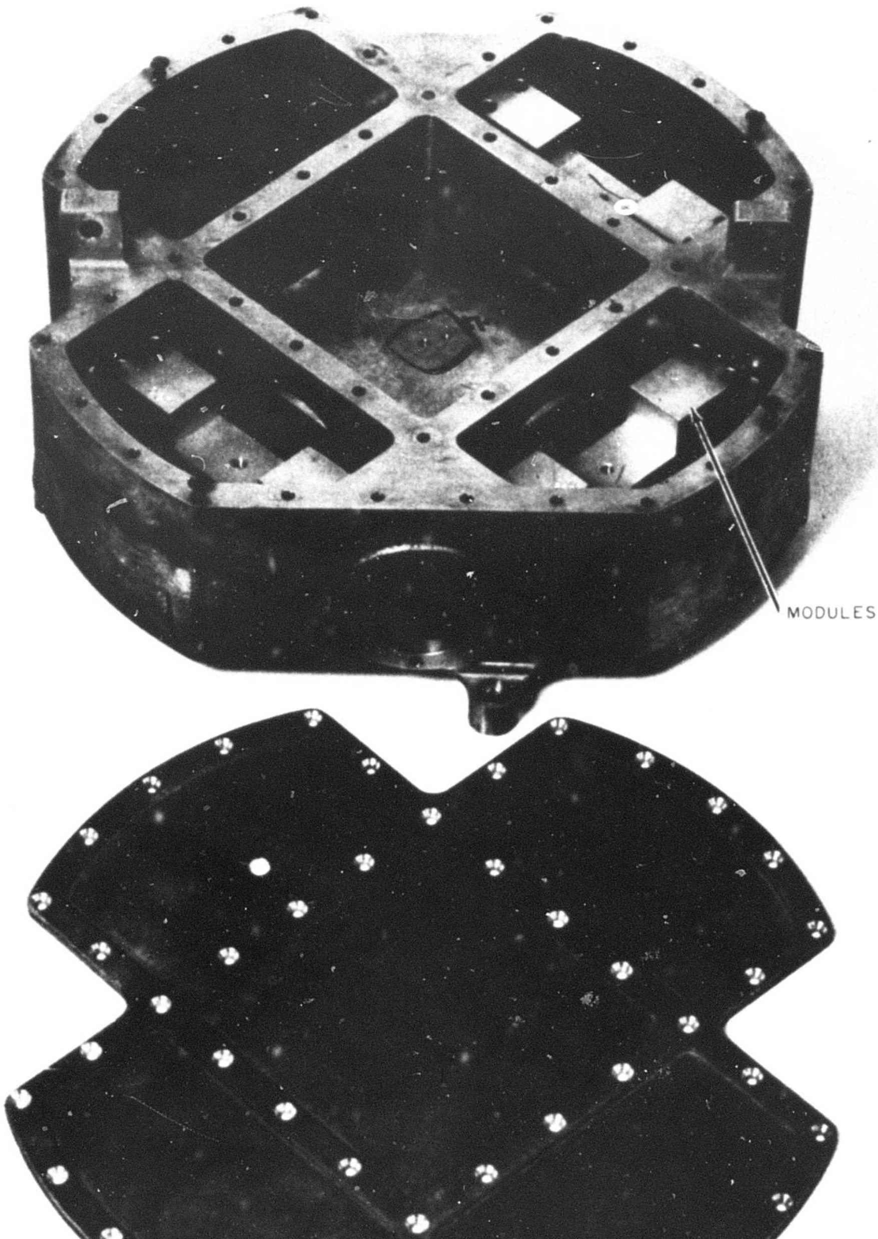

FIGURE 1. Cast Aluminum Housing and Plate Assembly Showing Mounted Mock Electronic Modules

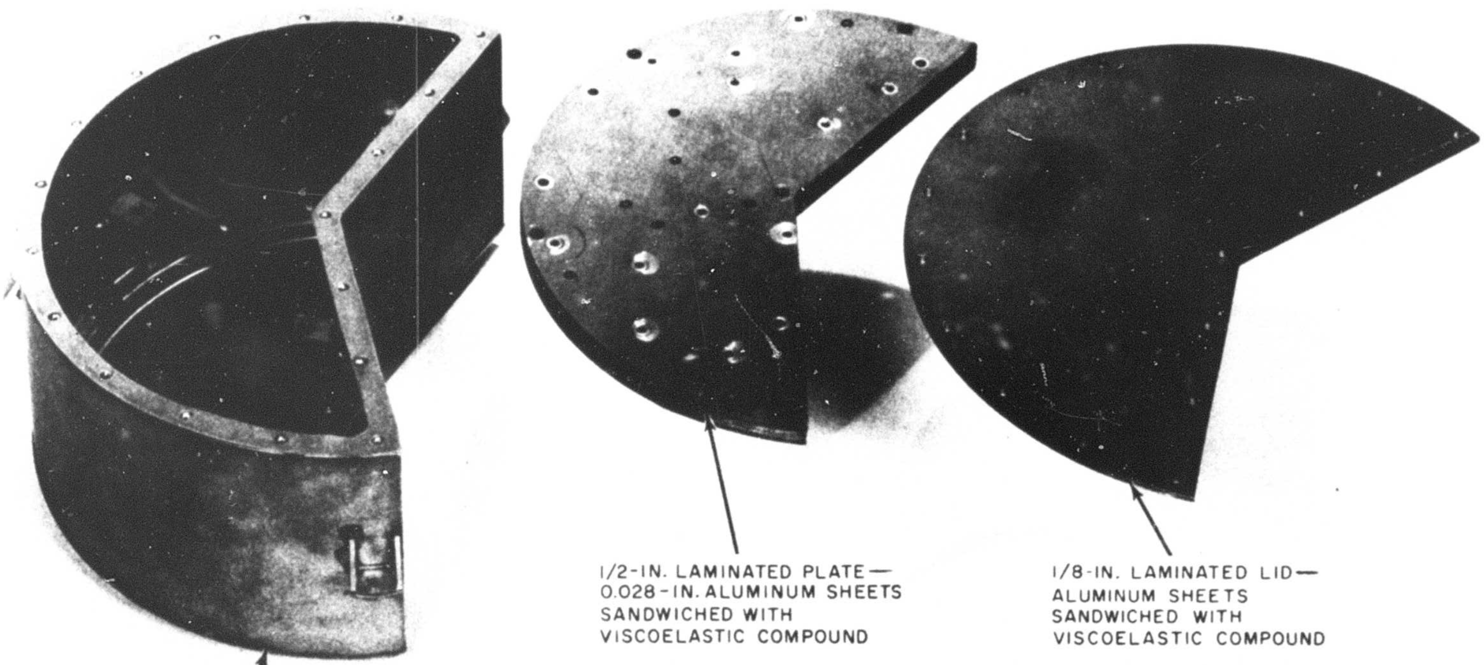

FIGURE 2. Sheet Aluminum Housing With Viscoelastic-Laminated Sheet Aluminum Plate

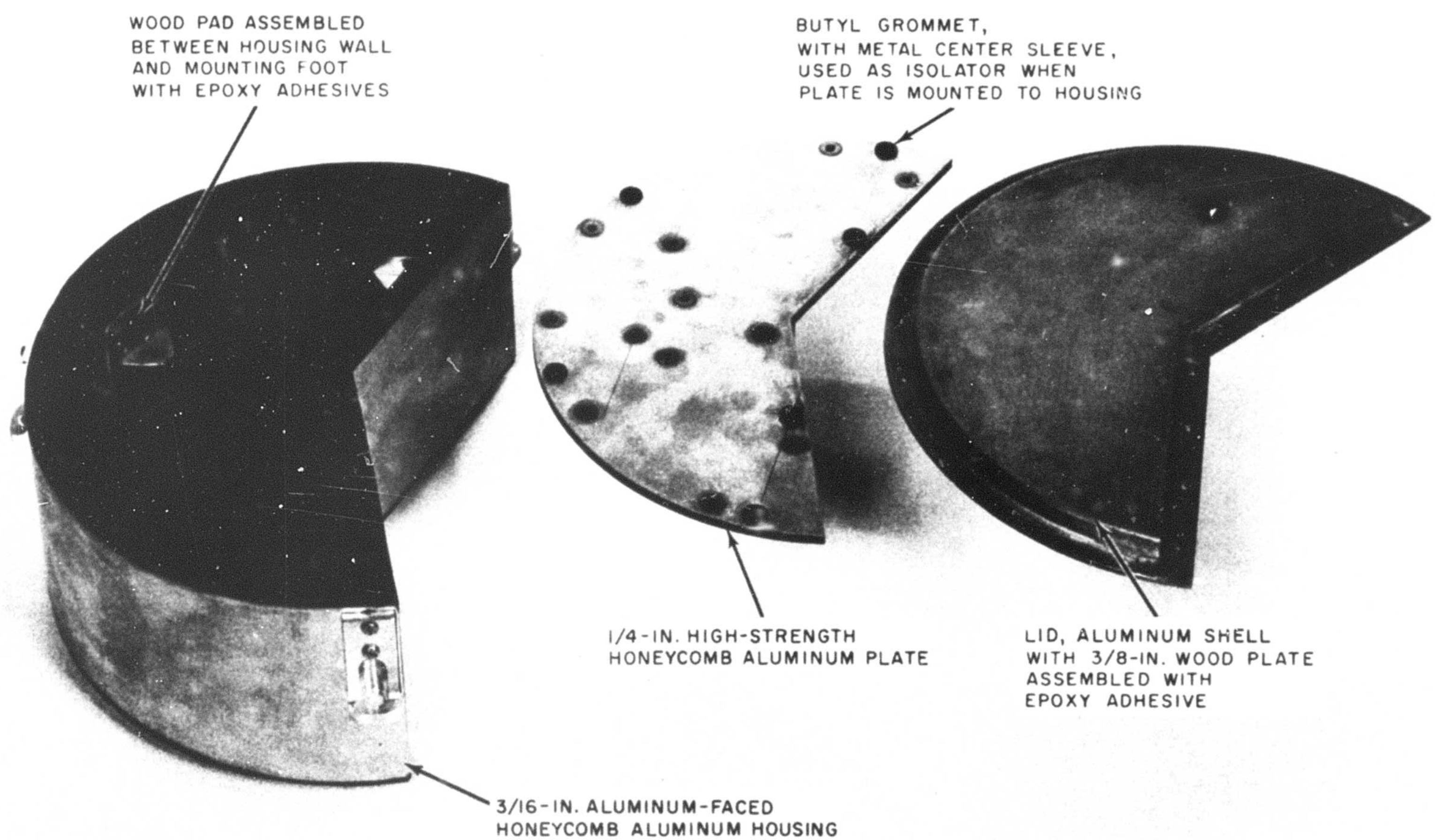

FIGURE 4. Honeycomb Aluminum Housing With Honeycomb Aluminum Plate

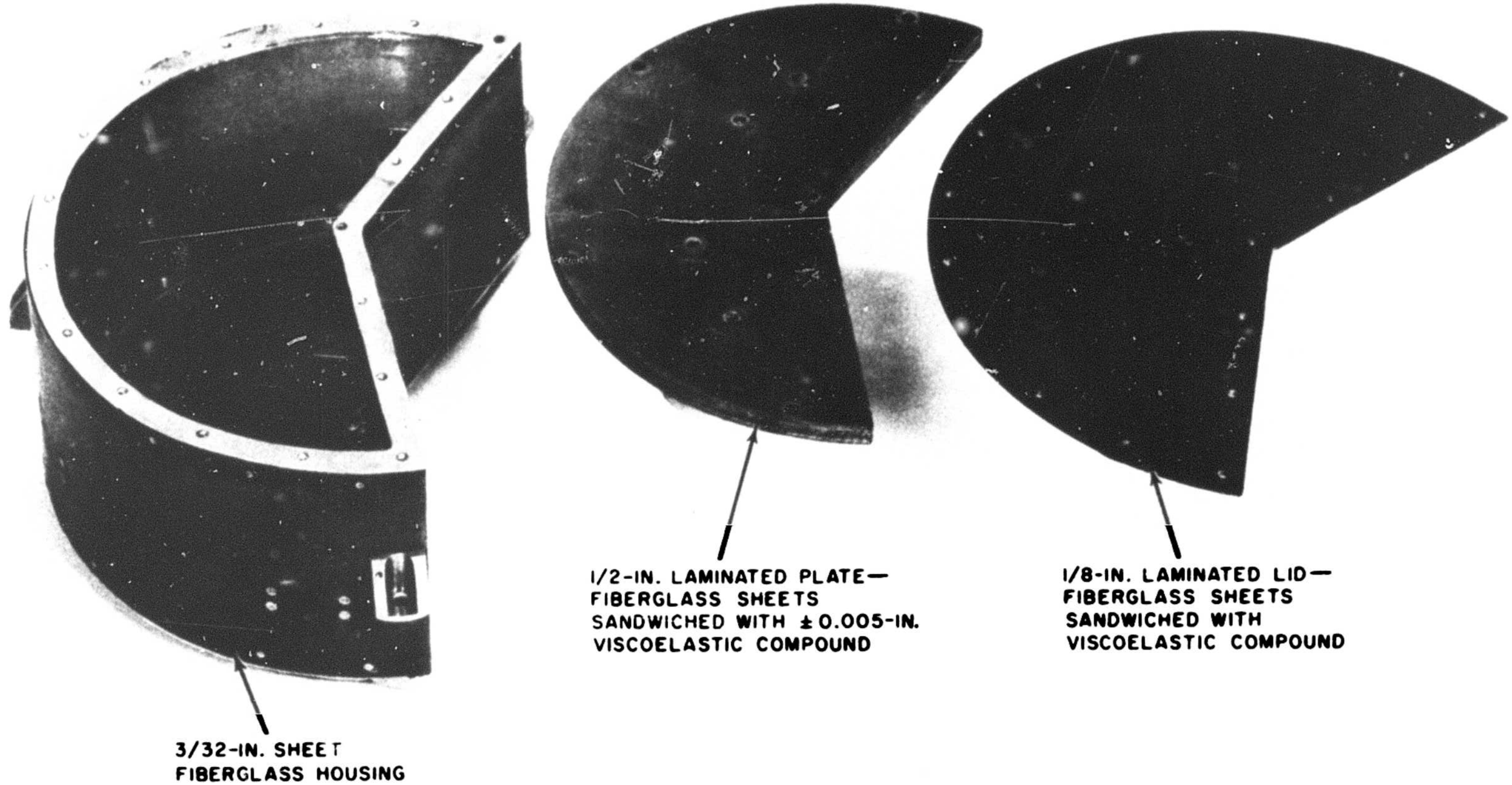

FIGURE 3. Fiberglass Housing With Viscoelastic-Laminated Fiberglass Plate

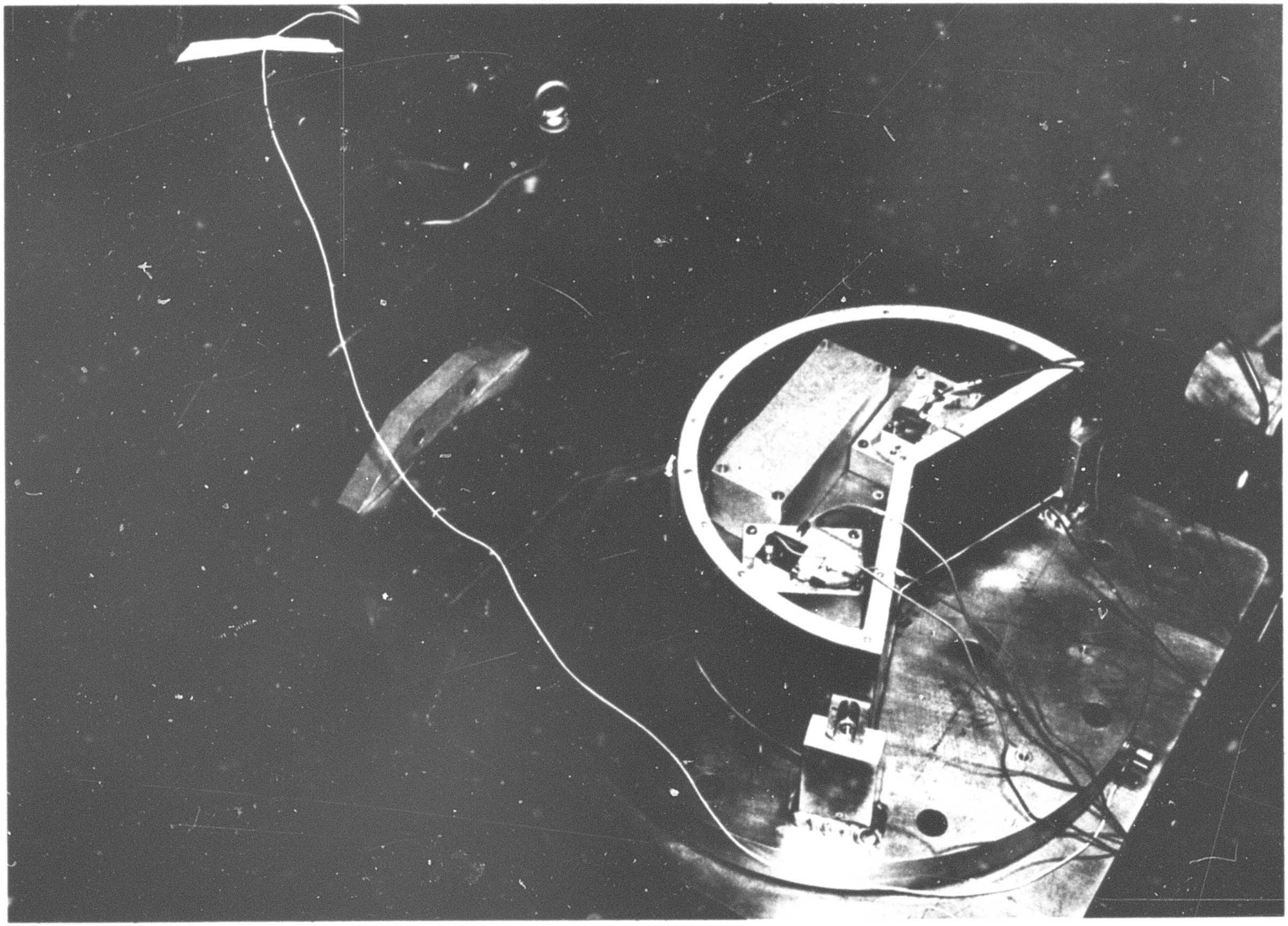

FIGURE 6. Fiberglass Housing and Plate Assembly Mounted on Ling A174 Oil-Slide, Air-Bearing Shaker Table

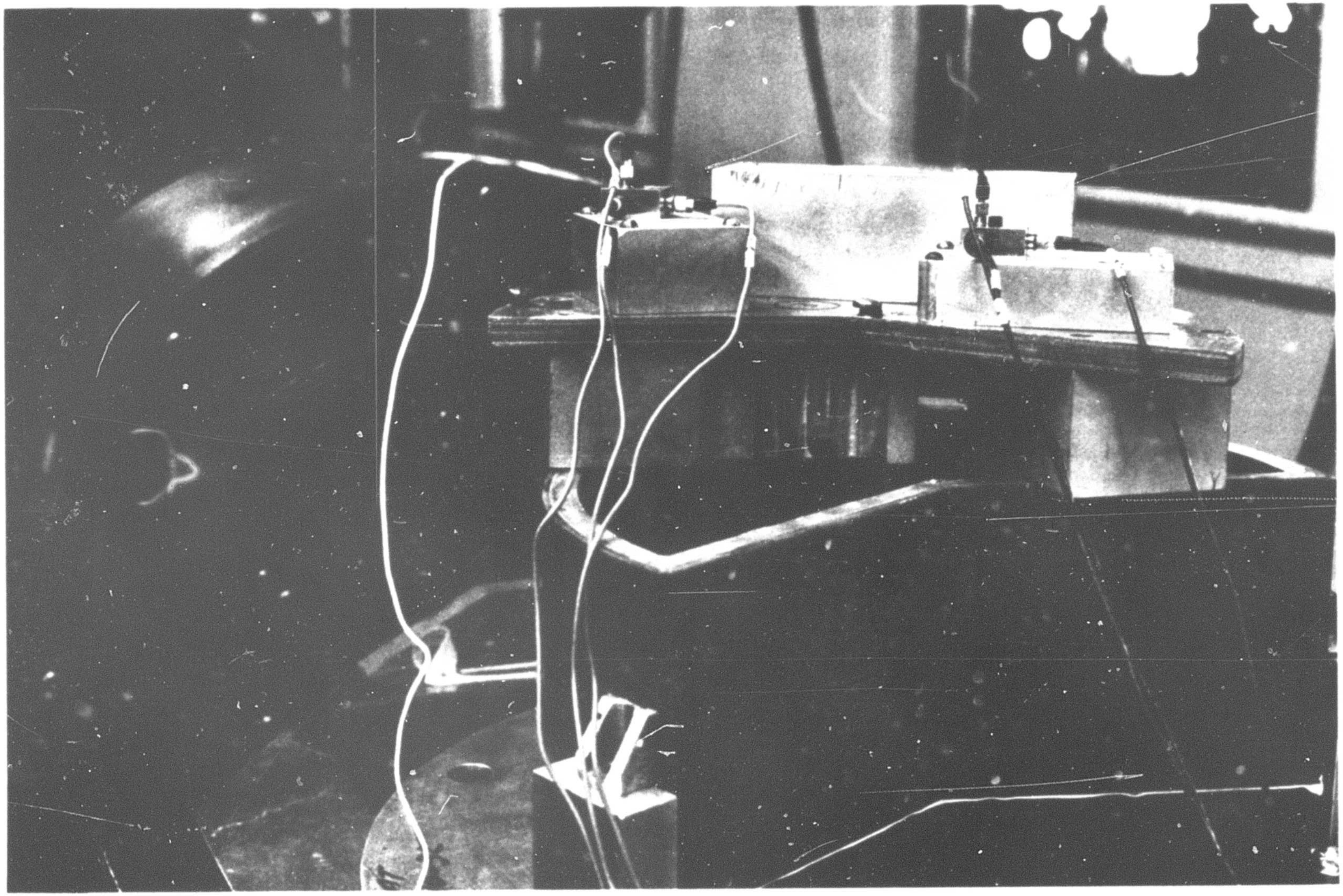

FIGURE 5. Fiberglass Housing Showing Plate Assembled with Mock Electronic Hardware and Triaxial Accelerometers

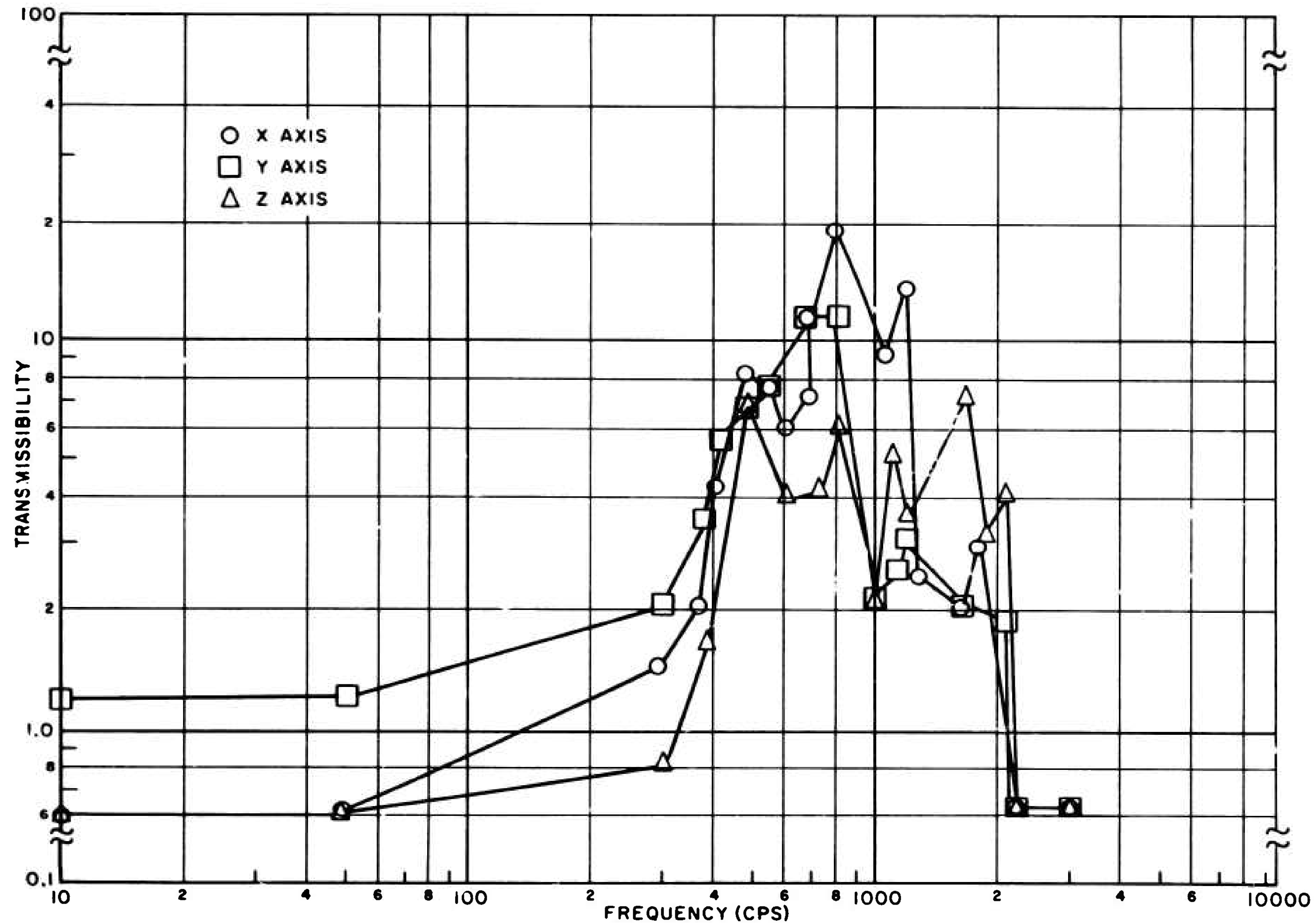

FIGURE 8. Cast Aluminum Housing and Plate Assembly Driven in Y Axis

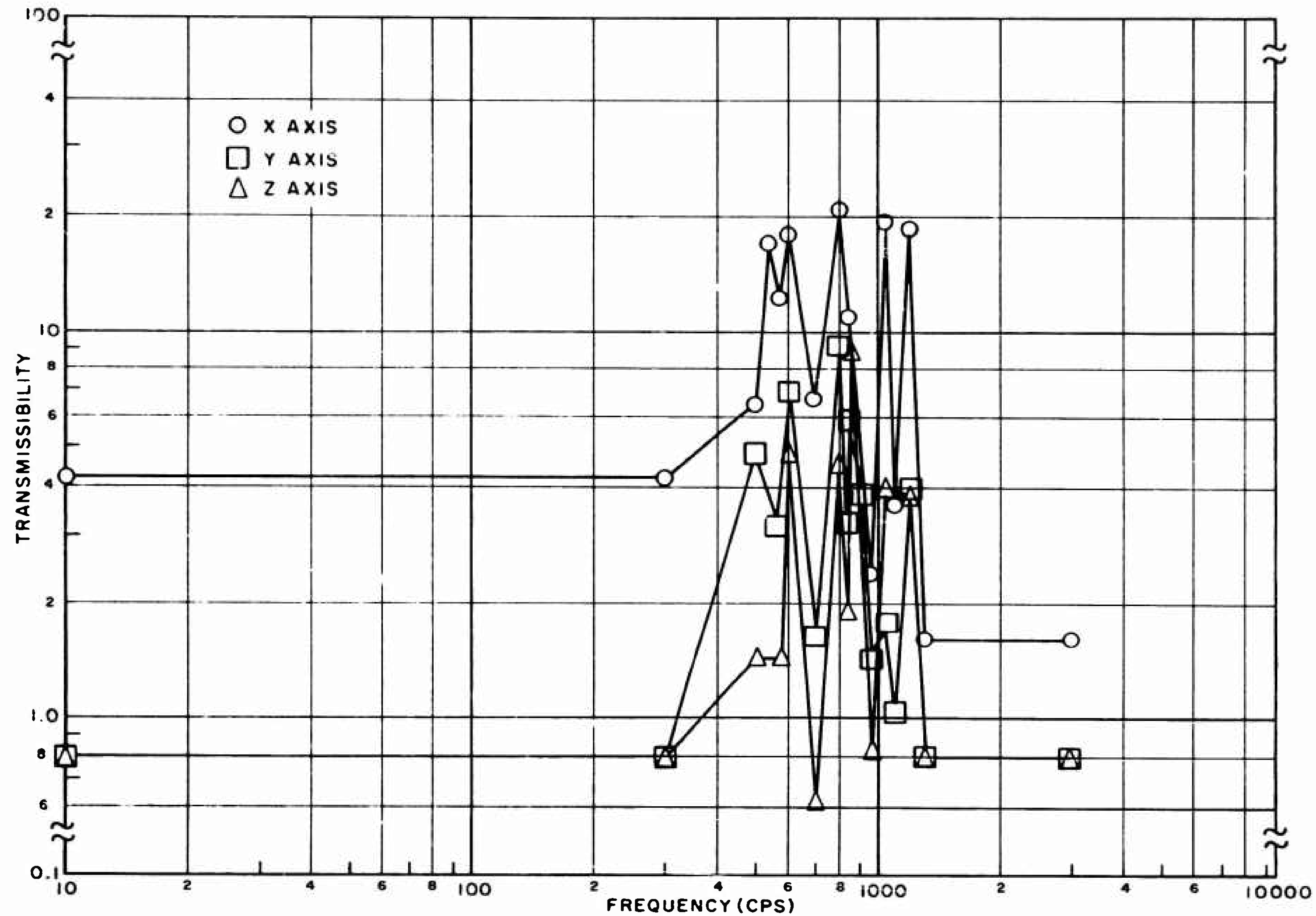

FIGURE 7. Cast Aluminum Housing and Plate Assembly Driven in X Axis

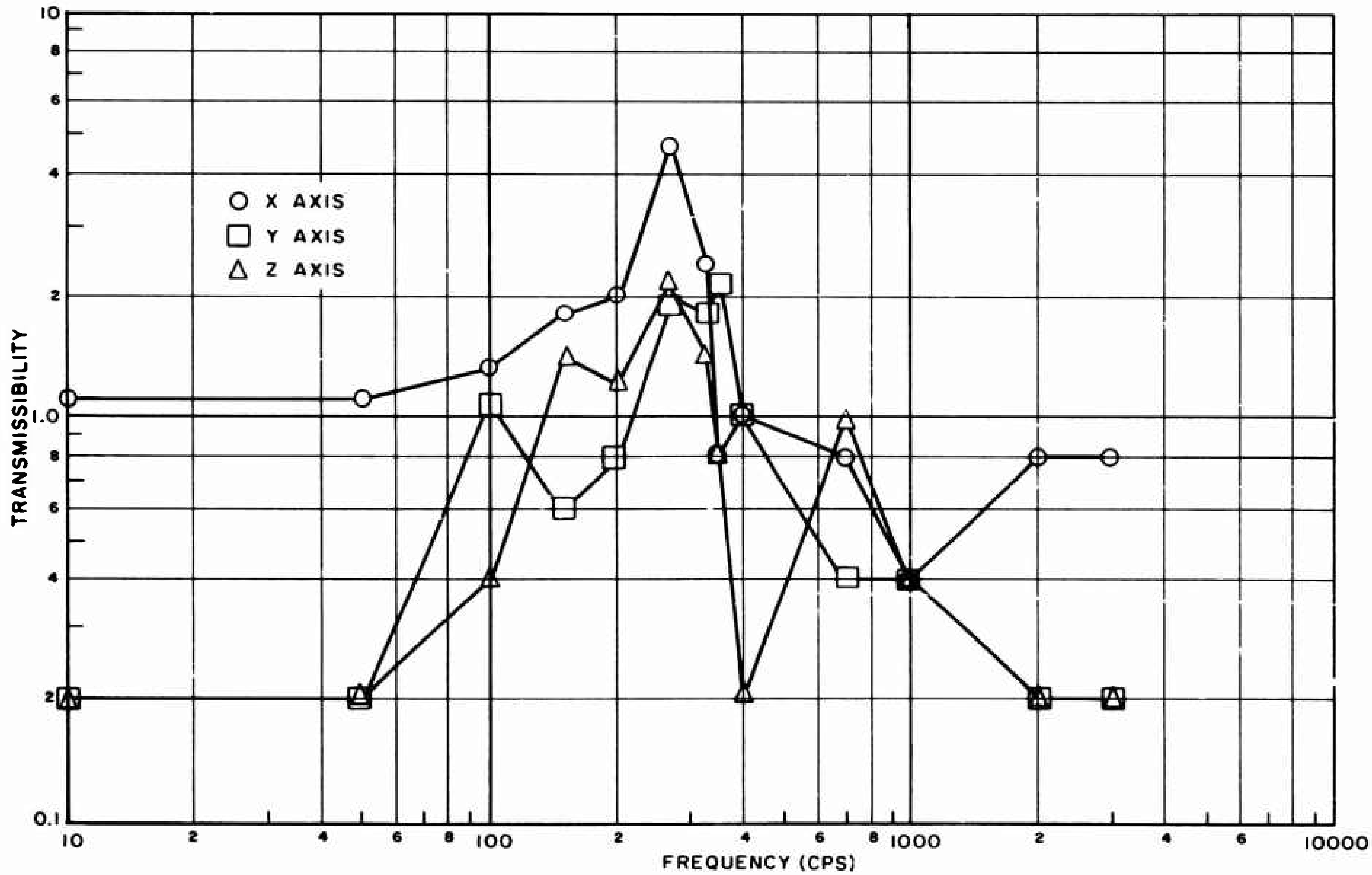

FIGURE 10. Sheet Aluminum Housing With Multilayer Viscoelastic-Laminated Sheet Aluminum Plate Driven in X Axis

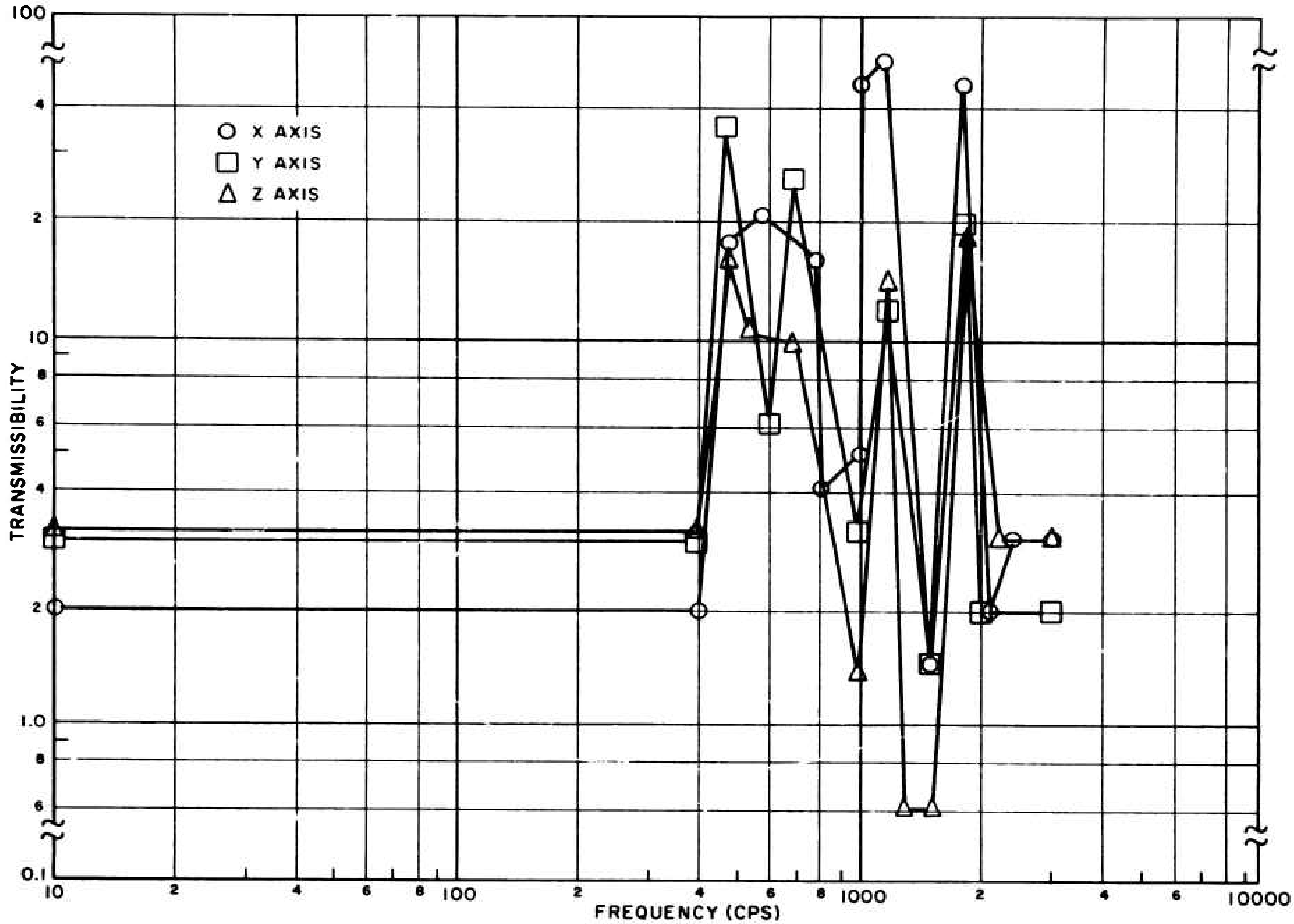

FIGURE 9. Cast Aluminum Housing and Plate Assembly Driven in Z Axis

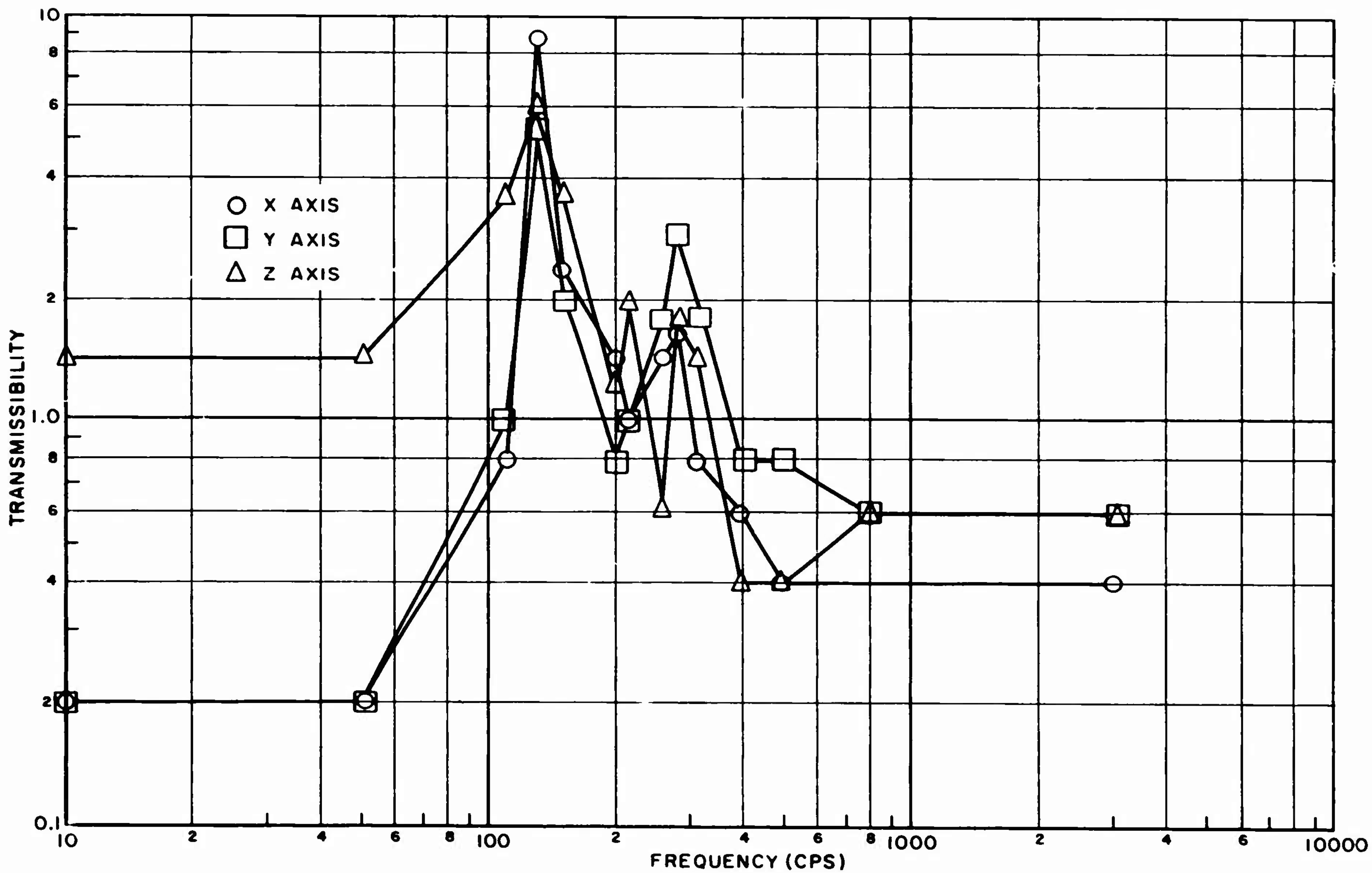

FIGURE 12. Sheet Aluminum Housing With Multilayer Viscoelastic-Laminated Sheet Aluminum Plate Driven in Z Axis

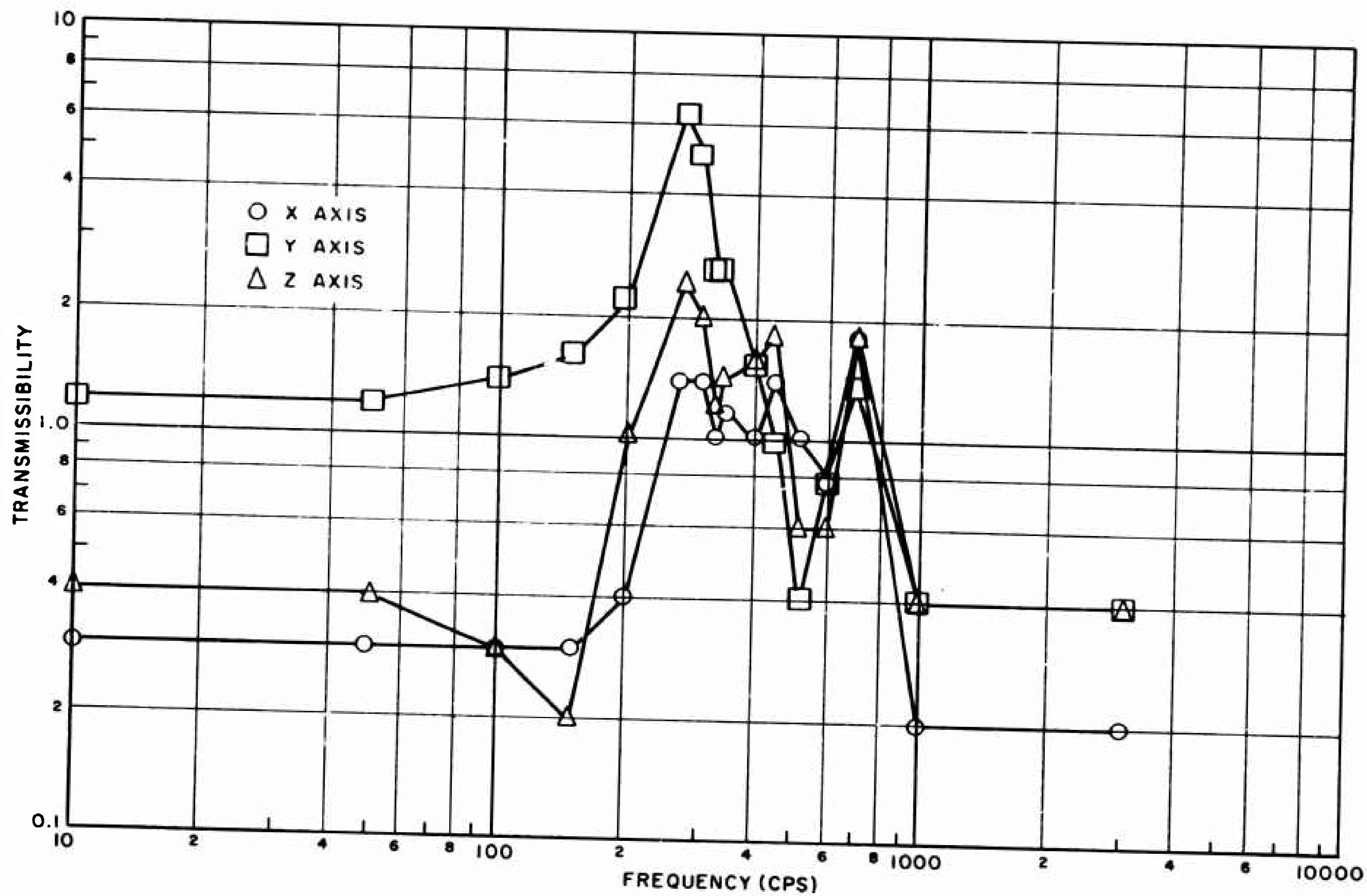

FIGURE 11. Sheet Aluminum Housing With Multilayer Viscoelastic-Laminated Sheet Aluminum Plate Driven in Y Axis

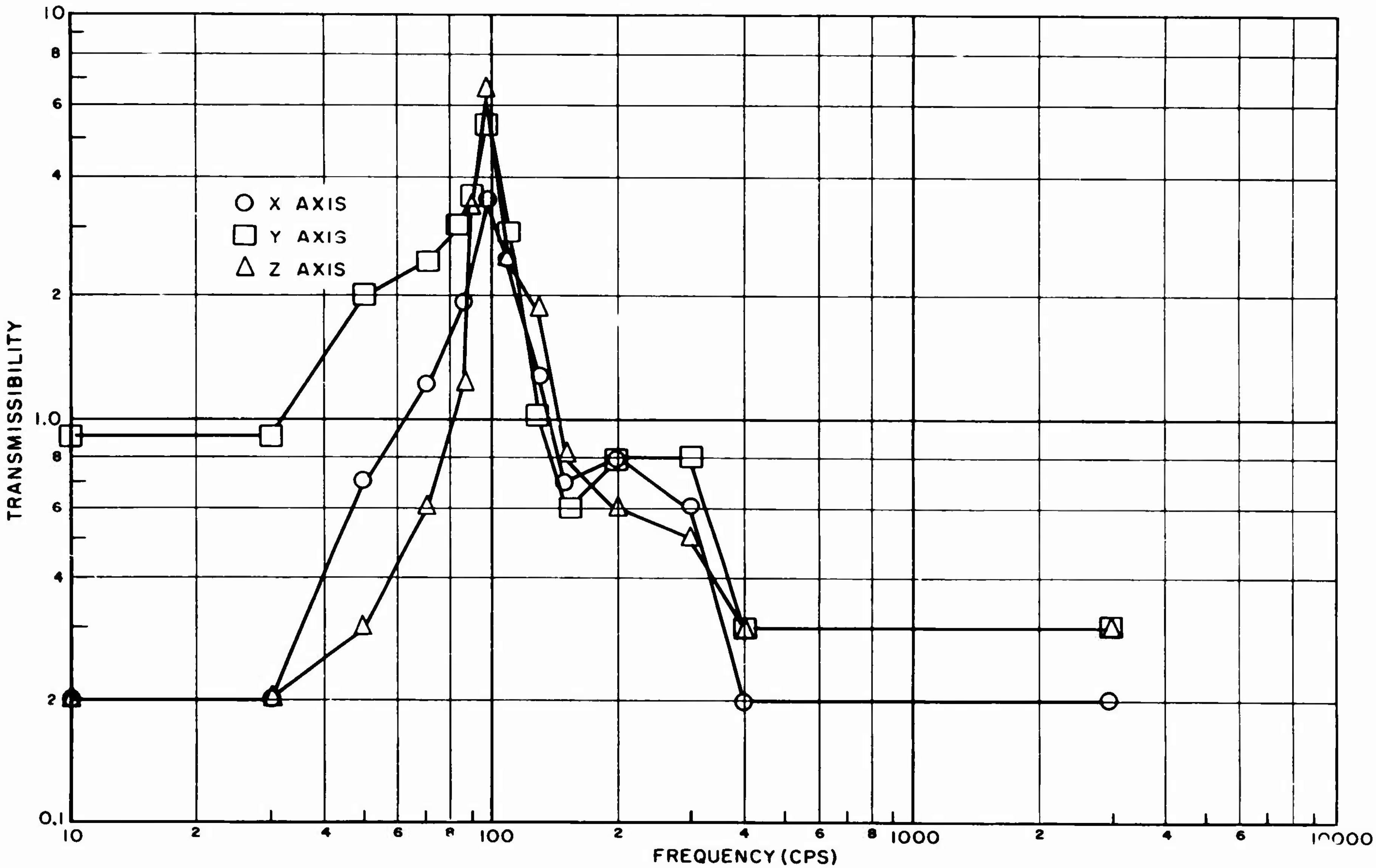

FIGURE 14. Fiberglass Housing With Viscoelastic-Laminated Fiberglass Plate Driven in Y Axis

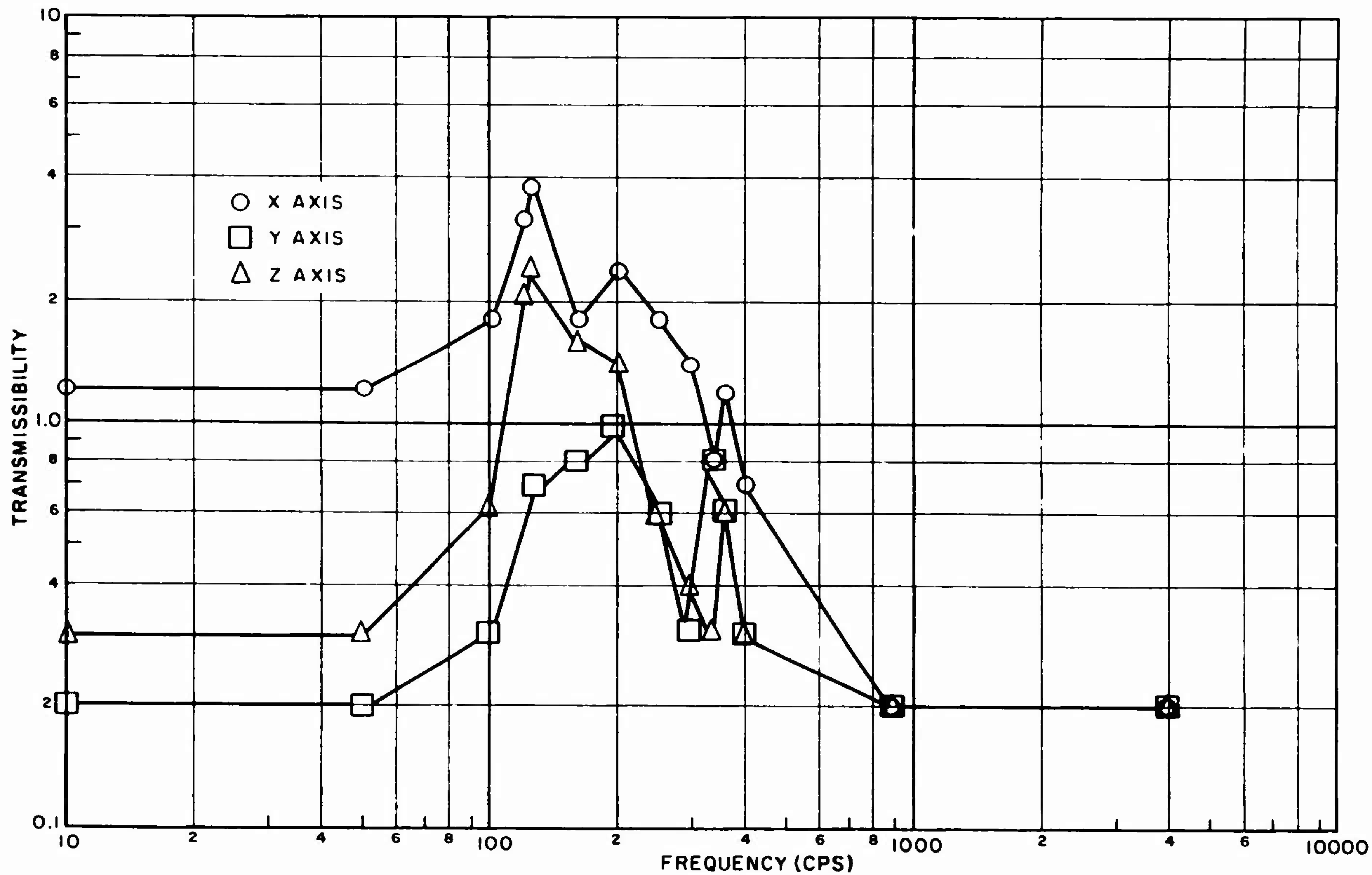

FIGURE 13. Fiberglass Housing With Viscoelastic-Laminated Fiberglass Plate Driven in X Axis

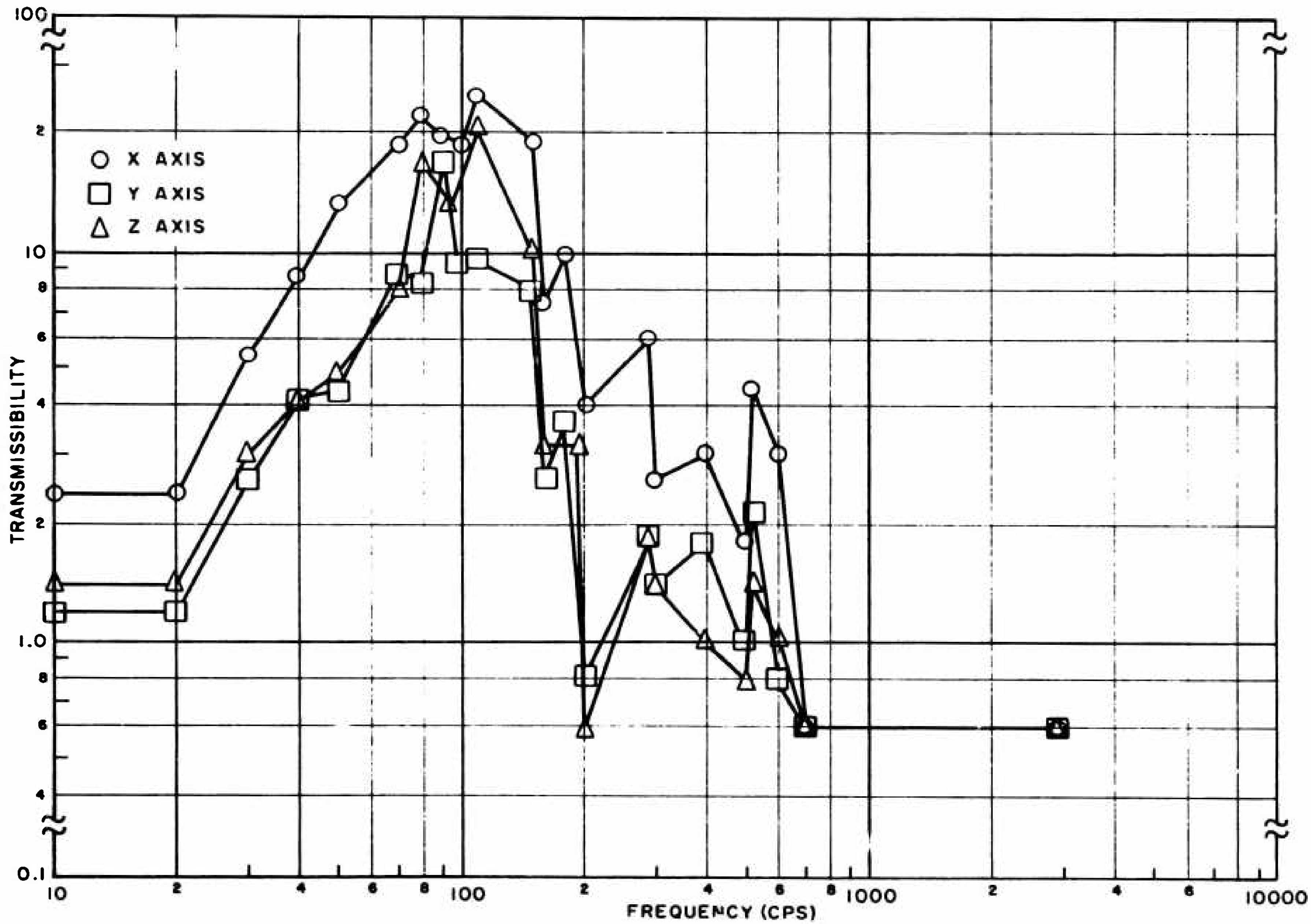

FIGURE 16. Honeycomb Aluminum Housing With Butyl-Isolated Honeycomb Aluminum Plate Driven in X Axis

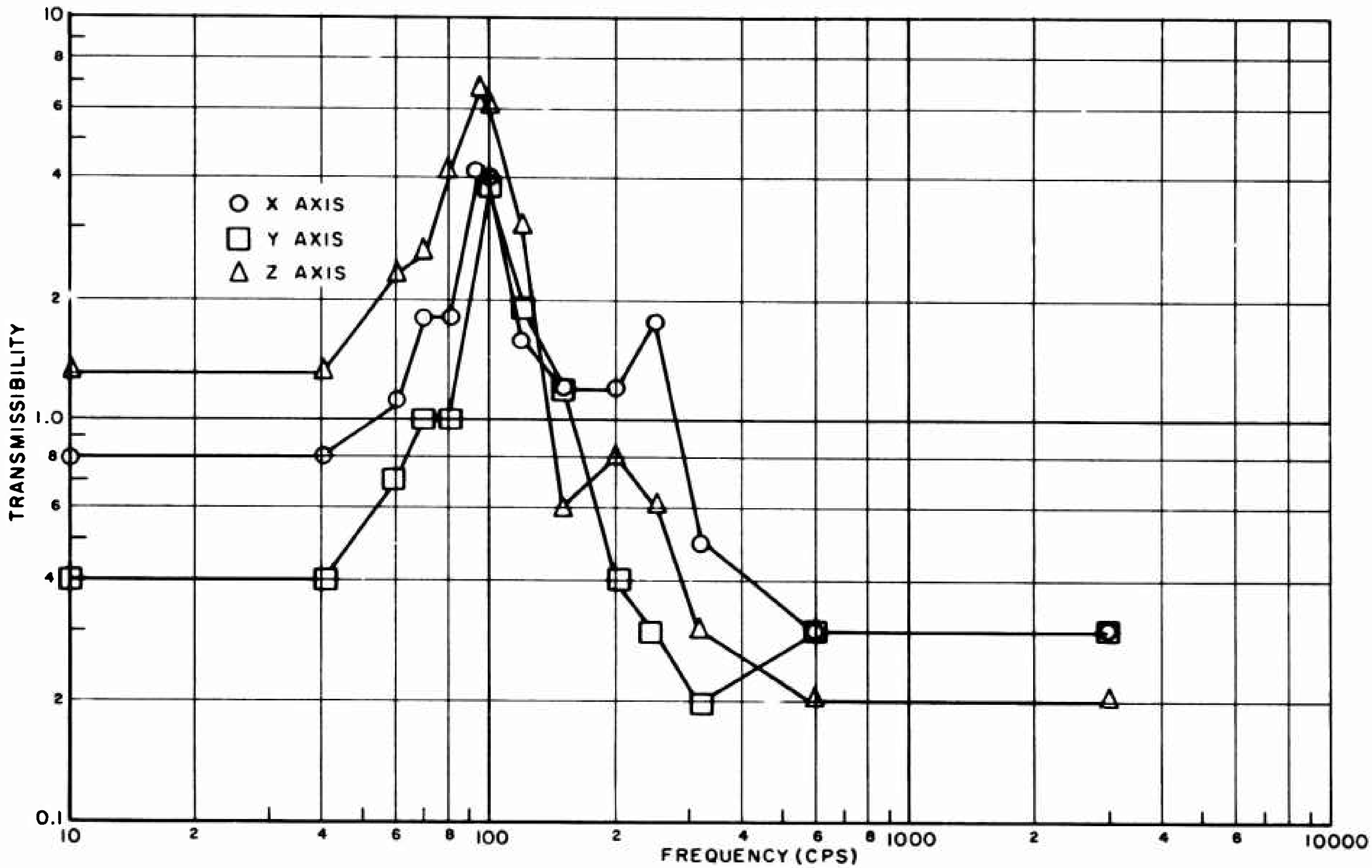

FIGURE 15. Fiberglass Housing With Viscoelastic-Laminated Fiberglass Plate Driven in Z Axis

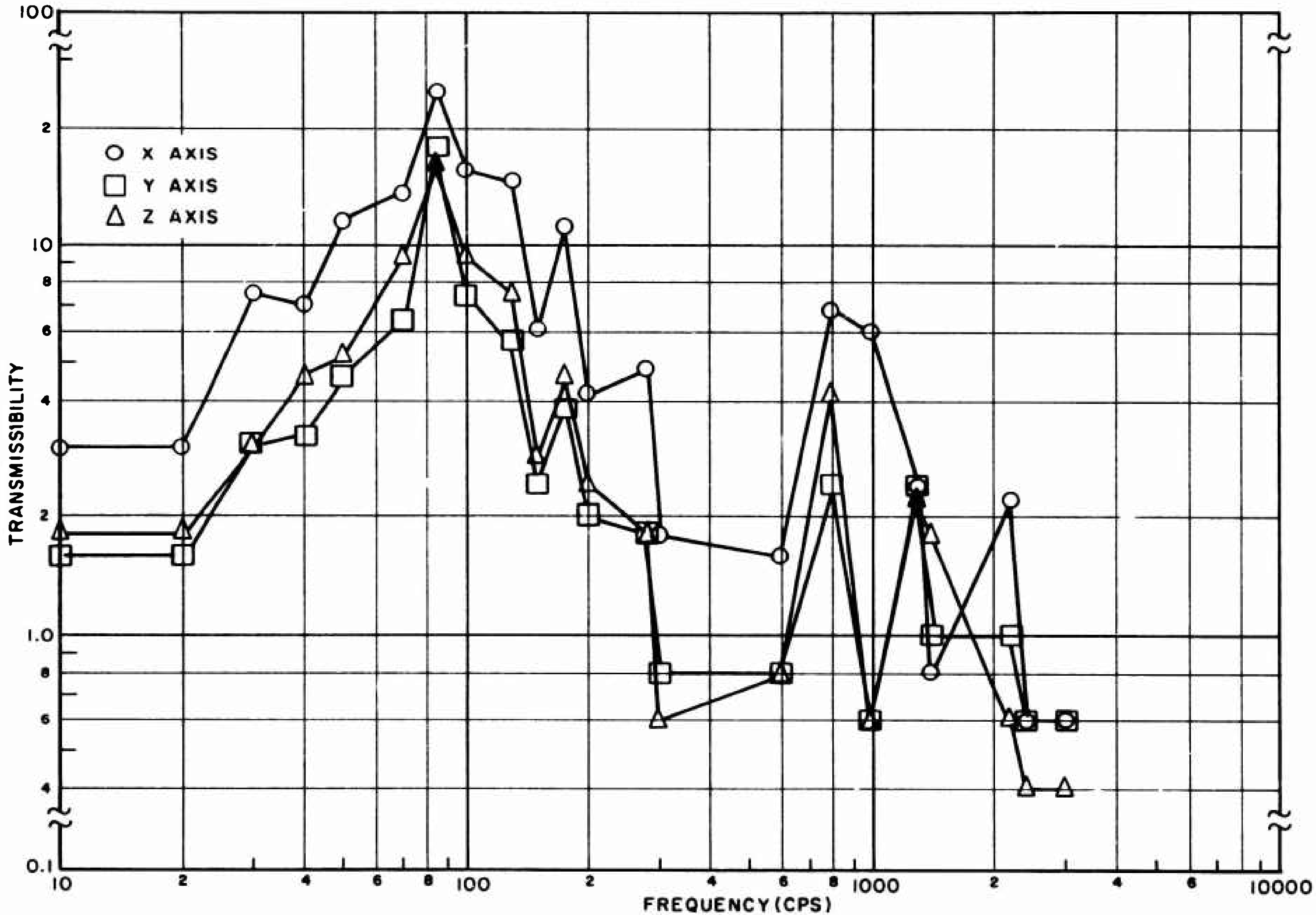

FIGURE 18. Honeycomb Aluminum Housing With Butyl-Isolated Honeycomb Aluminum Plate Driven in Z Axis

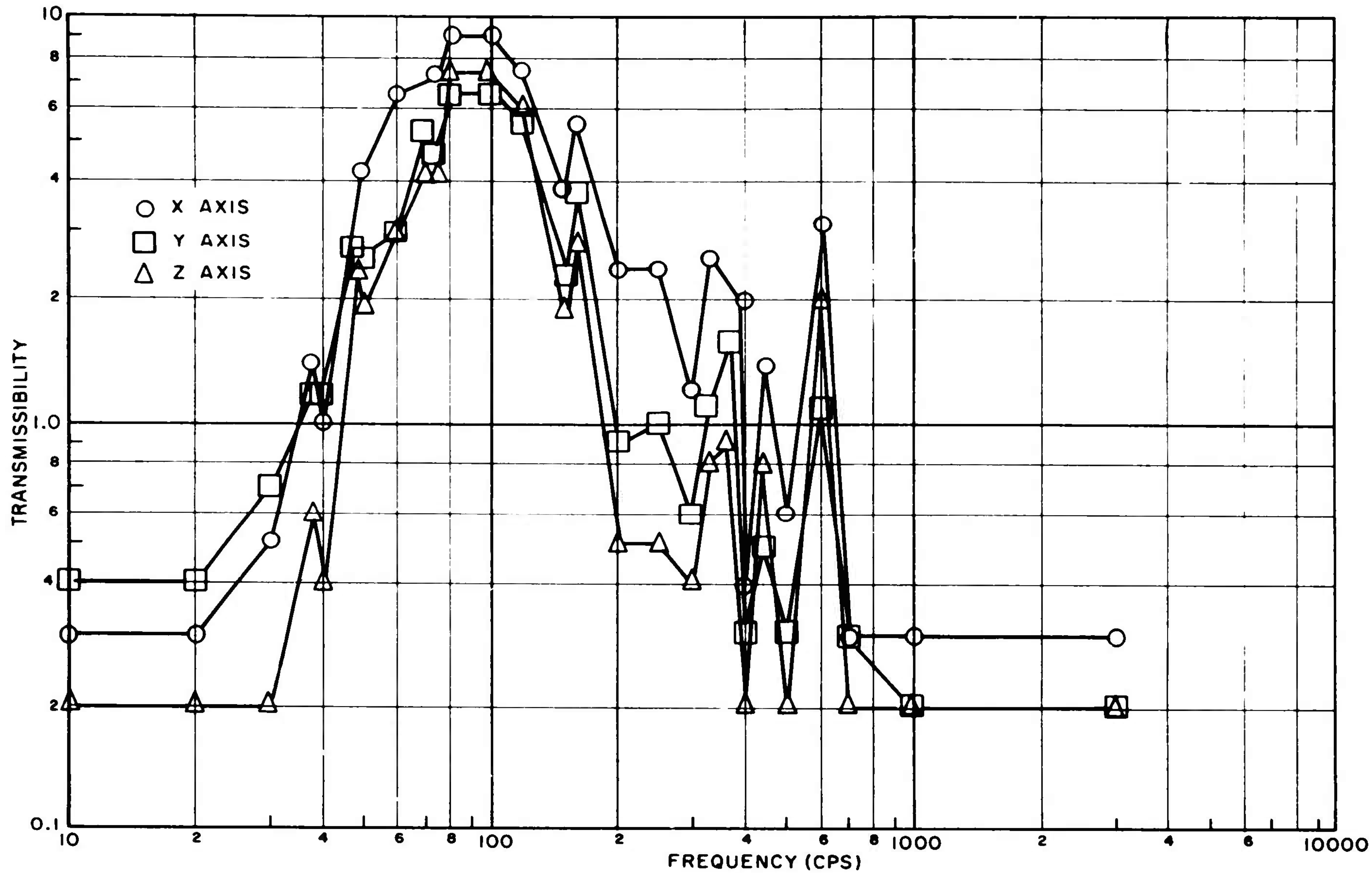

FIGURE 17. Honeycomb Aluminum Housing With Butyl-Isolated Honeycomb Aluminum Plate Driven in Y Axis

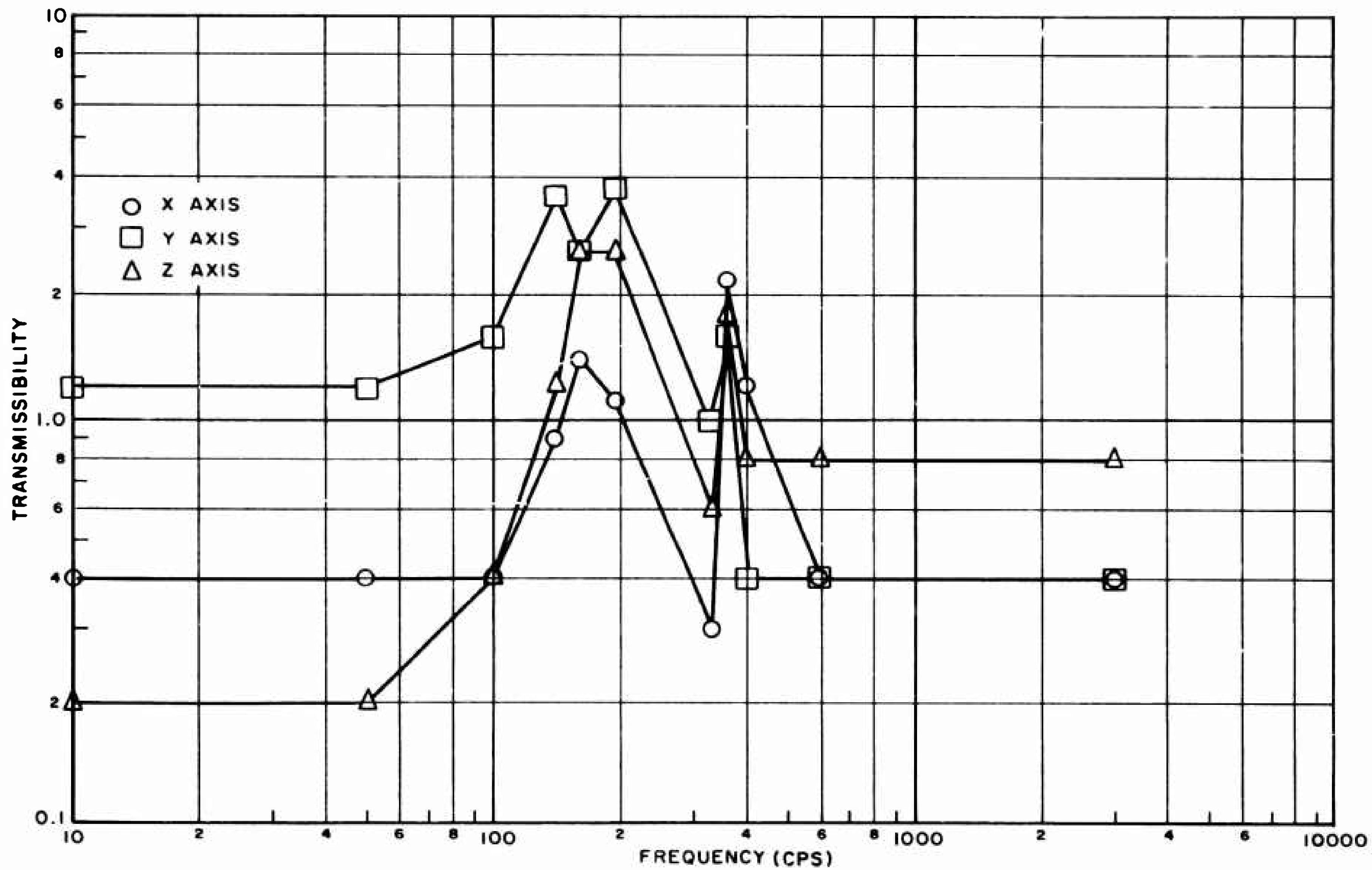

FIGURE 20. Fiberglass Housing With Multilayer Viscoelastic-Laminated Sheet Aluminum Plate Driven in Y Axis

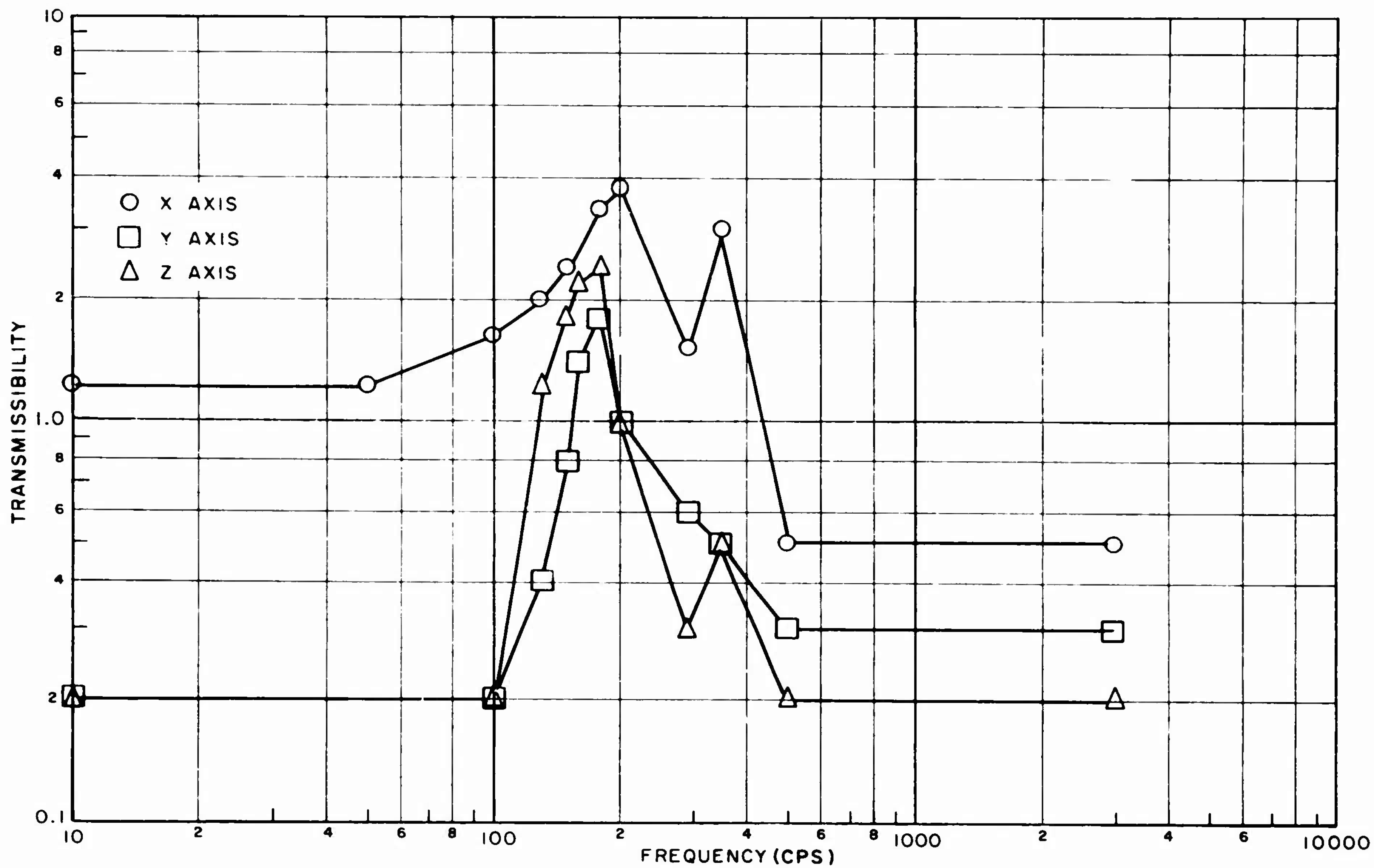

FIGURE 19. Fiberglass Housing With Multilayer Viscoelastic-Laminated Sheet Aluminum Plate Driven in X Axis

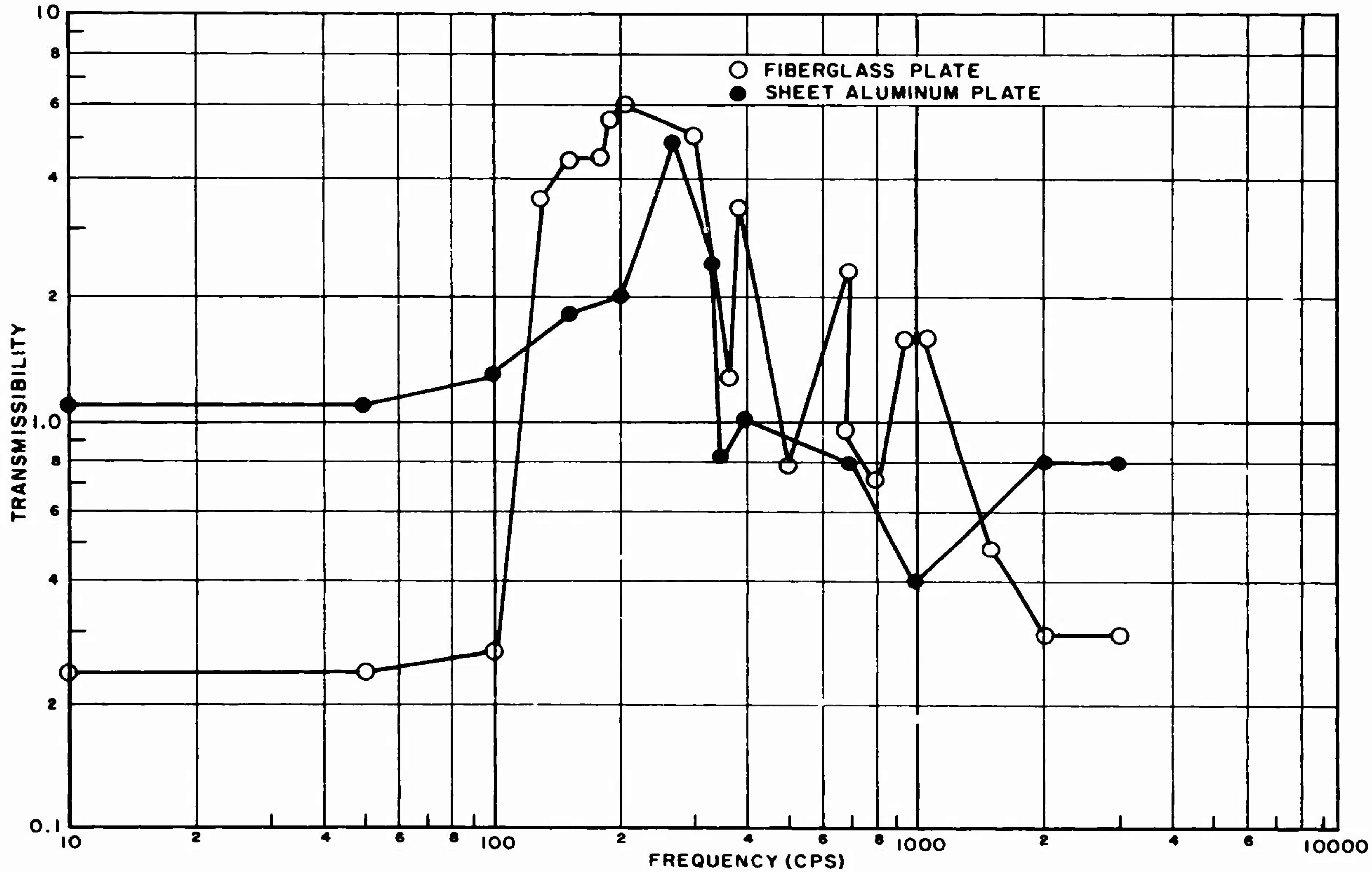

FIGURE 22. Sheet Aluminum Housing With Viscoelastic-Laminated Fiberglass Plate Compared With Sheet Aluminum Housing With Multilayer Viscoelastic-Laminated Sheet Aluminum Plate; Both Driven in X Axis

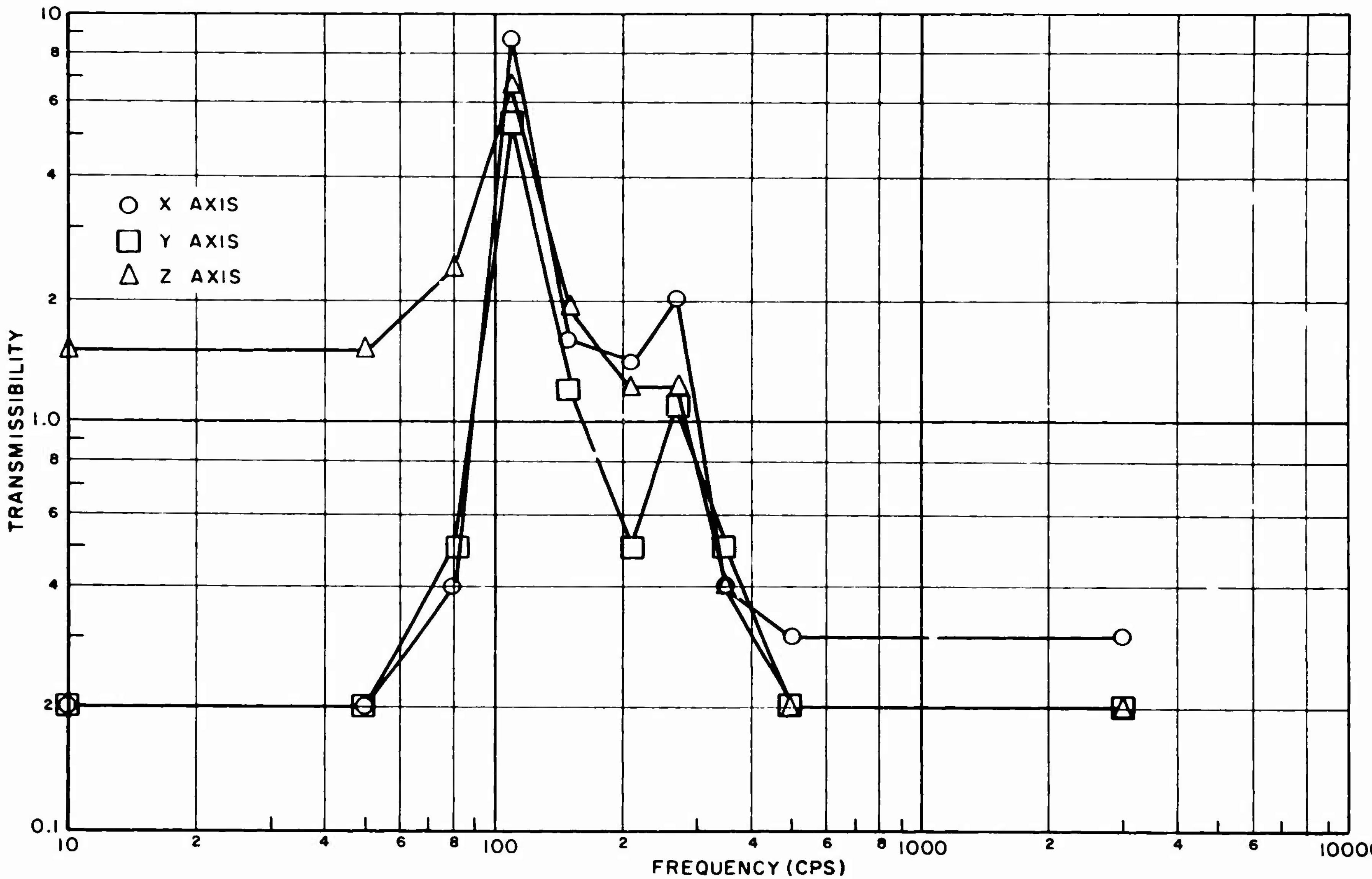

FIGURE 21. Fiberglass Housing With Multilayer Viscoelastic-Laminated Sheet Aluminum Plate Driven in Z Axis

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13 ABSTRACT

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14 KEY WORDS	LINK A		LINK B		LINK C	
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Vibration damping Resonant vibration—damping Viscoelasticity Hysteresis, mechanical						

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