General Disclaimer

One or more of the Following Statements may affect this Document

- This document has been reproduced from the best copy furnished by the organizational source. It is being released in the interest of making available as much information as possible.

- This document may contain data, which exceeds the sheet parameters. It was furnished in this condition by the organizational source and is the best copy available.

- This document may contain tone-on-tone or color graphs, charts and/or pictures, which have been reproduced in black and white.

- This document is paginated as submitted by the original source.

- Portions of this document are not fully legible due to the historical nature of some of the material. However, it is the best reproduction available from the original submission.

Produced by the NASA Center for Aerospace Information (CASI)

N.I.

BOLT BERANEK AND NEWMAN INC

CONSULTING · DEVELOPMENT · RESEARCH

Report 1702
Job 11326

DEVELOPMENT OF IMPROVED VIBRATION TESTS OF SPACECRAFT ASSEMBLIES

Contract No. 951990

This work was performed for the Jet Propulsion Laboratory, California Institute of Technology, sponsored by the National Aeronautics and Space Administration under Contract NAS7-100.

October 1968

Prepared by:

Bolt Beranek and Newman Inc.
15808 Wyandotte Street
Van Nuys, California 91406

Submitted to:

Jet Propulsion Laboratory
California Institute of Technology
4800 Oak Grove Drive
Pasadena, California 91103

RECEIVED
NASA STI FACILITY
INPUT BRANCH
DEC 1968

N69-11383
(ACCESSION NUMBER)
44
(PAGES)
CR-97796
(NASA CR OR TMX OR AD NUMBER)
(THRU)
1
(CODE)
31
(CATEGORY)
FACILITY FORM 602

Report 1702
BBN Job 11326

DEVELOPMENT OF IMPROVED VIBRATION TESTS OF SPACECRAFT ASSEMBLIES

NOTICE

This report was prepared as an account of Government-sponsored work. Neither the United States, nor the National Aeronautics and Space Administration (NASA) nor any person acting on behalf of NASA:

a. Makes warranty or representation, expressed or implied, with respect to the accuracy, completeness, or usefulness of the information contained in this report, or that the use of any information, apparatus, method, or process disclosed in this report may not infringe privately-owned rights; or

b. Assumes any liabilities with respect to the use of, or for damages resulting from the use of any information, apparatus, method, or process disclosed in this report.

As used above, "person acting on behalf of NASA" includes any employee or contractor of NASA, or employee of such contractor, to the extent that such employees or contractor of NASA, or employee of such contractor prepares, disseminates, or provides access to, any information pursuant to his employment with such contractor.

Requests for copies of this report should be referred to:

National Aeronautics and Space Administration
Office of Scientific and Technical Information
Washington 25, D.C.

TABLE OF CONTENTS

	Page
SUMMARY	1
INTRODUCTION	1
Assembly and System Level Tests	1
Rigid Versus Multimodal Fixtures	2
Objectives of Investigation	3
DEVELOPMENT OF MULTIMODAL TEST FIXTURE	4
Description of Fixture Development Tests	4
Modal Density Enrichment	5
Damping	8
TEST RESULTS	8
Description of Excitation and Mounting Configurations	8
Sine-Sweep Response of Assembly	9
Octave-Band Random Transfer Functions	10
Spatial Variation	12
DESIGN AND TESTING GUIDELINES	13
Definition of a Multimodal Test Fixture	13
Design Procedure	14
Excitation Configuration	18
Specification and Control	19
Data Analysis	21
Possible Problem Areas	21
CONCLUSIONS AND RECOMMENDATIONS	23
GLOSSARY	26
REFERENCES	28
FIGURES	

DEVELOPMENT OF IMPROVED VIBRATION TESTS FOR SPACECRAFT ASSEMBLIES

SUMMARY

This report presents the results of an experimental research program to develop a novel type of spacecraft assembly-level vibration test. A multimodal test fixture which is designed to simulate the high-frequency (100-2000 Hz) dynamic characteristics of a Mariner spacecraft structure is developed. The data obtained in research vibration tests of a simulated Mariner electronic assembly mounted in the multimodal fixture indicate that the novel testing technique results in a more realistic and more uniform high-frequency vibration environment on the assembly than conventional assembly-level tests utilizing rigid fixtures. The report presents guidelines for designing and utilizing multimodal type fixtures in future spacecraft assembly-level vibration test programs.

INTRODUCTION

Assembly and System Level Tests

It is common practice in aerospace development programs to conduct both assembly-level and system-level vibration tests. In the assembly level tests, which are usually conducted early in the program, the assemblies are attached to stiff fixtures and mounted rigidly to the vibration exciter or vibration slip table. In the system-level test, the complete system with all of its assemblies in place is attached to a stiff fixture which is then mounted on a vibration exciter or slip table.

In a previous study, the vibration environment of a Mariner C electronic assembly in simulated assembly-level and spacecraft-level vibration tests was investigated.[1] The results of the study indicate that the stiff test fixture mounting used in the assembly-level tests result in overtesting of the assembly at frequencies above 300 Hz, and that fixture resonance problems render the test data meaningless at frequencies above 1000 Hz. The results also indicate that the spatial variations in the assembly response are much greater in the assembly-level tests than in the spacecraft-level tests. The problem of performing assembly-level tests which simulate the vibration environment of the assembly mounted in the complete spacecraft or space vehicle is common to all spacecraft development programs. Therefore the present study was undertaken to develop improved vibration tests of spacecraft assemblies.

Rigid Versus Multimodal Fixtures

The problems which arise in assembly vibration tests at high-frequencies (above approximately 100 Hz) are associated primarily with the fact that current vibration test fixtures are designed to produce a unidirection in phase excitation. At low frequencies the vibration environment of the assemblies is governed by overall vibration modes of the launch vehicle or spacecraft, and the vibration wavelength is long compared to a typical assembly dimension. Therefore the assembly-spacecraft interface moves as a rigid body. Assembly vibration tests with rigid fixtures simulate this low-frequency rigid body motion.

In the high-frequency regime, where the spacecraft vibration environment is the result of acoustic, aerodynamic, or shock excitation, the vibration wavelength in the spacecraft structure is typically short compared to the characteristic dimensions of the assemblies, and the vibration modes of the assemblies, and spacecraft, and the launch vehicle interact in a very complex way. Conventional assembly test fixtures, which are designed to be rigid in order to place their first resonance frequency as high as possible, do not adequately simulate the high-frequency vibration environment of aerospace assemblies.

In order to develop assembly-level tests which provide realistic results at high frequencies, the assembly test fixture must be designed to approximate the multimodal high-frequency response characteristics of the spacecraft structure adjacent to the assembly. Previous results indicate that a multimodal vibration test fixture which exhibits many vibration resonances in the excitation frequency band, exhibits very uniform frequency and spatial response characteristics.

Objectives of Investigation

The primary objective of this study is to develop a multimodal-type vibration test fixture for spacecraft assembly-level tests and to perform vibration tests with the multimodal fixture using a model of a real spacecraft assembly. The results of this study are compared with the results from conventional assembly-level and spacecraft-level vibration tests in order to determine the relative advantages and disadvantages of the multimodal vibration

testing concept. A secondary objective of this study is to develop design and testing guidelines which can be used to incorporate the multimodal vibration testing concept into future Jet Propulsion Laboratory (JPL) vibration test programs.

DEVELOPMENT OF MULTIMODAL TEST FIXTURE

The development of a multimodal test fixture is at the present time basically a cut-and-try procedure. This section describes the steps followed in designing, fabricating, and testing a multimodal fixture for high-frequency research vibration tests of a dynamic model of an electronic assembly from the Mariner 1969 spacecraft. In a later section of this report, some general guidelines for designing and testing multimodal fixtures are presented.

Description of Fixture Development Tests

Figure 1 presents a perspective view of the Mariner 1969 spacecraft electronic assembly and the multimodal fixture construction. The basic fixture resembles a three-eighths section of the Mariner spacecraft bus which consists of eight assemblies arranged in the shape of an octagon. As Fig. 1 illustrates, the electronic assembly used in this test program mounts in the center bay of the three-bay fixture. Since the development of a multimodal test fixture is a cut-and-try process, it is necessary to perform sine-sweep and octave-band random development vibration tests on the fixture during the course of its development.

Figure 2 shows the two small shaker mounting configurations used in development tests of the multimodal fixture. In the first configuration, two 25-lb force Goodman shakers are attached to the backplates of the fixture, and in the second configuration the shakers are attached to the fixture feet. Figure 3 shows a photograph of the electronic assembly mounted in the multimodal fixture excited with small shakers attached to the fixture backplates. In the actual development tests, the fixture was supported on a large cushion of foam rubber which is not shown in the photograph.

Modal Density Enrichment

The Mariner multimodal fixture is built-up starting with a three-eighths Mariner bus fixture identified in Fig. 1 as the fixture frame. This frame was previously developed by JPL for assembly-level experimental tests. The sine-sweep acceleration measured on the bare fixture frame excited with small shakers attached at its feet is shown in Fig. 4. The acceleration is measured on the horizontal frame structure of the center bay. The sine-sweep response shown in Fig. 4 exhibits many vibration modes at frequencies above 1000 Hz. The objective of the fixture development program was to add secondary structure to the bare fixture frame in order to enrich its modal density in the frequency region from 100 to 1000 Hz.

The first modification to the bare fixture involves adding two fixture backplates to the fixture bays adjacent to the test bay as shown in Fig. 1. These plates are designed to approximate the dynamic effects of the backplates of the Mariner spacecraft electronic assemblies.

A quantitative criterion for the simulation of the dynamic properties of a flat late can be formulated in terms of the point force impedance of an infinite flat plate given by Eq. 1.[2]

$$Z_p(f) = \frac{4}{\sqrt{3}} \rho_s h^2 c_\ell \qquad (1)$$

In Eq. 1, ρ_s is the plate mass per unit area, h is the plate thickness, and c_ℓ is the speed of sound in the plate structure (17,000 ft per second in steel and aluminum).

The acceleration response of the fixture frame with the backplates added is shown in Fig. 5. In this test, the shakers are attached to the fixture backplates as shown in Fig. 2a. The sine-sweep response exhibited in Fig. 5 shows slightly higher modal density in the 100 to 1000 Hz range than that of the bare fixture shown in Fig. 4.

The second step in enriching the modal density of the fixture involves adding a large aluminum plate to the back side of the fixture and two small plates connecting the plate with the fixture frame as shown in Fig. 1. Again, these plates are designed so that their equivalent infinite plate point impedance as given by Eq. 1 approximates the infinite plate point impedance of the Mariner electronic assembly backplates. In addition, wire-mesh, fin-like elements are added to the back of both fixture backplates in order to simulate electronic module boards and further enrich the modal density of the fixture as shown in Fig. 1.

The increase in modal density which may be expected by adding plates to enrich the modal density of a fixture may be estimated from Eq. 2[3]

$$\delta_p(f) = \frac{hc_\ell}{\sqrt{3}A} \qquad (2)$$

where δ_p is the average frequency interval between resonant modes of the plate and A is the total plate area. The total area of the modal enrichment plates is approximately 16 ft^2 and the plates are constructed of 1/8-inch-thick aluminum. Equation 2 indicates that the plates should exhibit a vibration resonance approximately every 6 Hz. The increase in modal density to be expected from the wire-mesh is more difficult to estimate, but the effect can be approximated by considering the wire-mesh as a series of beams. The modal separation for a beam is given by Eq. 3[3]

$$\delta_b(f) = [h_b c_\ell f/2\sqrt{3}]^{1/2}/L \qquad (3)$$

where h_b is the beam depth and L is the total length of the beams. Equation 3 indicates that the wire-mesh has a very high modal density indeed. Of course, many of the wire-mesh modes will not couple into the multimodal fixture because of the impedance mismatch between the wire-mesh and the fixture backplate structure.

The sine-sweep acceleration response on the fixture frame with the backplates, modal enrichment plates, and wire-mesh added is shown in Fig. 6. The sine-sweep response in Fig. 6 exhibits approximately 20 vibration resonances

in the frequency range from 100 to 200 Hz or approximately six resonances per 1/3-octave band. At frequencies above 200 Hz it is difficult to resolve the resonance frequencies, but the modal density is very large. Therefore this fixture meets the multimodal criterion (see glossary), and it is concluded that this configuration is adequate.

Damping

The final step in the development of the multimodal fixture involves adding strips of damping tape (Scotch Brand Pressure Sensitive Tape No. 428A) to the fixture backplates and modal enrichment plates. The application of this damping tape reduces the peak-to-valley variations in the frequency response of the modal enrichment plates and backplates but does not significantly affect the frequency response of the basic fixture frame.

TEST RESULTS

Description of Excitation and Mounting Configurations

Two basic excitation configurations were used in the multimodal fixture tests of the Mariner spacecraft assembly. The first configuration involved small shaker excitation of the fixture backplates and the fixture feet as shown in Fig. 2. The second configuration involved mounting the multimodal fixture on a conventional slip table and exciting the fixture at its four feet as shown in Fig. 7a and at the fixture mounting block as shown in Fig. 7b. Figure 8 shows a picture of the complete fixture and assembly mounted on a conventional shaker and slip table at the fixture mounting blocks. The small

shaker excitation tests were performed at Bolt Beranek and Newman Inc., Van Nuys, California, and the slip table tests were performed at JPL. Both sine-sweep and octave-band random tests were conducted with the small shaker and slip table excitation configurations.

Sine-Sweep Response of Assembly

The sine-sweep response of the Mariner electronic assembly with the multimodal fixture excited with small shakers at the fixture backplates is shown in Fig. 9. The assembly backplates exhibit a high modal density at frequencies above 100 Hz. The input to the shaker in this test consists of a constant voltage swept sinusoid, so no attempt is made to control the acceleration response at a constant level.

The sine-sweep response of the Mariner assembly backplate with the multimodal fixture excited with a conventional slip table at the mounting blocks is shown in Fig. 10. The sine-sweep response obtained with the slip table mount shows a strong resonance frequency at approximately 180 Hz and exhibits few modes between 100 and 300 Hz. The rigid mounting on the conventional slip table suppresses the modal interaction of the multimodal fixture and the electronic assembly in the low frequency regime. The results presented in Figs. 9 and 10 indicate that small shaker excitation is preferable over conventional slip table excitation for exciting the multimodal fixture.

Octave-Band Random Transfer Functions

The mechanical vibration transfer functions from the multimodal fixture to the assembly are the most important parameters for evaluating the performance of the multimodal fixture. Two important transfer functions are considered: 1) the transfer function from the fixture-assembly interface to the assembly backplate; and 2) the transfer function from the fixture feet to the assembly backplate. The first transfer function is useful for comparing the multimodal fixture test results with the results obtained in conventional assembly-level and spacecraft-level tests. The second transfer function is useful for assessing the use of multimodal fixture assembly-level tests to approximate system-level tests with control at the spacecraft feet.

The octave-band random transfer functions from the fixture-assembly interface to the assembly backplate are shown in Fig. 11. The assembly backplate acceleration levels and the fixture-assembly interface acceleration levels used to compute these transfer functions represent the average of acceleration measurements at many different points on the backplate and the interface. The solid line in Fig. 11 represents the transfer function obtained in JPL system-level vibration tests using a dynamic model spacecraft, the Development Test Model (DTM). The short dashed line represents the transfer function obtained in conventional rigid fixture assembly-level tests at JPL. The data presented in Fig. 11 indicate that the transfer functions obtained in all the multimodal fixture tests more closely simulate the transfer function in

the spacecraft-level DTM tests than the transfer function obtained in rigid fixture assembly-level tests. The multimodal fixture tests with small shaker excitation of the fixture backplates most closely simulates the system-level DTM test results. The data presented in Fig. 11 indicate that the Mariner assembly multimodal test fixture indeed provides a means of conducting more realistic high-frequency assembly-level tests than conventional rigid assembly-level fixtures.

Figure 12 shows the transfer functions from the fixture feet to the Mariner electronic assembly backplate. These transfer functions again are based on the average of many acceleration measurements on the fixture backplate and the average of acceleration measurements at the four fixture feet. The solid line again presents the transfer function obtained in the system level DTM vibration tests conducted at JPL. The long-dash short-dash line presents the transfer function obtained in the multimodal fixture tests with the small shakers attached at the fixture feet. The small shaker feet excitation multimodal fixture tests clearly provide a more realistic simulation of the transfer function from the feet to the assembly backplate than the slip table feet-mount or block-mount multimodal fixture tests. Thus, the space-average transfer functions presented in Figs. 11 and 12 also confirm that small shaker excitation of the multimodal fixture is preferable to excitation with the conventional slip table.

Spatial Variation

Figure 13 presents the measured spatial variances of the assembly backplate mean-square acceleration response in various test configurations. The normalized variance plotted in Fig. 13 is ten times the log of the spatial variance in the mean-square acceleration response divided by the square of the spatial-average mean-square acceleration response. In order to provide a controlled and uniform vibration environment of the electronic assembly, it is advantageous to minimize the spatial variance in the assembly backplate response at high frequencies. The data presented in Fig. 13 indicate that the spatial variances in the conventional assembly-level tests greatly exceed the variances in the DTM test, and that the spatial variances in the small shaker and slip table excitation multimodal fixture tests are well below those measured in the DTM tests. The data indicate that the variances in the small shaker multimodal fixture tests are significantly lower than those of the slip table multimodal fixture tests.

Also presented in Fig. 13 is a theoretical value of the assembly backplate spatial variation calculated from Eq. 4[4]

$$\sigma^2/m^2 = 5\delta/\Delta \qquad (4)$$

where σ^2 divided by m^2 is the normalized spatial variance, δ is the modal separation of the backplate as given by Eq. 2, and Δ is the octave-band excitation bandwidth. Equation 4 is valid when the modal bandwidth $\frac{\pi}{2}\eta f$ (where

η is the plate damping loss factor) is less than the plate modal separation δ. Equation 4 indicates that the normalized variance is inversely proportional to the number of vibration modes which contribute to the assembly backplate response in the excitation frequency band. This theoretical result indicates that the multimodal test fixture configuration should result in significantly less spatial variation in the assembly response than conventional assembly-level tests.

DESIGN AND TESTING GUIDELINES

Definition of a Multimodal Test Fixture

A multimodal assembly test fixture is a fixture which is designed to simulate the high-frequency vibration characteristics of the mounting structure adjacent to the spacecraft assembly. The fundamental characteristic of a multimodal test fixture is that the fixture operates at frequencies well above its fundamental resonance frequency, and the fixture has three or more resonance frequencies in each one-third octave band over the entire frequency range of its application. It generally follows from this definition, that the fixture will exhibit modal overlap (the bandwidth of the fixture resonance peaks are greater than the modal frequency separation). The effect of high modal density and modal overlap is to reduce the frequency and spatial variations in the fixture response.

The modal separation for the fixture can be calculated from expressions similar to Eqs. 2 or 3 or determined experimentally by counting the number of resonant peaks in the measured sine-sweep response. The equivalent

bandwidth of the fixture resonant peaks is given by $\frac{\pi \eta f}{2}$, where η is the damping loss factor defined as twice the critical damping ratio. Thus a quantitative definition of modal overlap is[5]

$$\frac{\pi}{2}\eta f \geq \delta \qquad (5)$$

As an example, consider the sine-sweep response of the Mariner multimodal fixture shown in Fig. 6. The sine-sweep response shows approximately 20 resonance peaks in the frequency range from 100 to 200 Hz, or an average of approximately 7 peaks per third-octave band. Therefore the modal separation is approximately 5 Hz and Eq. 5 indicates that modal overlap can be achieved with a damping loss factor value of approximately 2×10^{-2} which is approximately equal to the measured value of the Mariner multimodal fixture damping loss factor at a frequency of 150 Hz. Therefore we conclude that the Mariner fixture indeed constitutes a multimodal fixture and exhibits modal overlap at frequencies of 100 Hz and above.

Design Procedure

In the preceeding section it was stated that the multimodal fixture should simulate the portion of the spacecraft mounting structure adjacent to the assembly. The first question that arises in designing a multimodal fixture is "How much of the adjacent structure should the multimodal fixture physically simulate?" Let us define the characteristic length associated with the assembly spacecraft interface (in the case of a Mariner

spacecraft assembly, the characteristic length would be equal to the width of the assembly backplate or approximately 2 ft). The multimodal fixture should simulate the spacecraft mounting structure one characteristic length on each side of the assembly. Thus the multimodal fixture developed for the Mariner spacecraft assembly simulates the assembly bays on each side of the test assembly (Fig. 1). This criterion breaks down for the case of an assembly mounted essentially at a point on the spacecraft, for example consider a hypothetical antenna which might be cantilevered from the spacecraft structure. In this case engineering judgment must be used to determine the amount of spacecraft structure which the fixture should physically simulate.

Insofar as possible, the fixture should simulate the dynamic characteristics of the spacecraft mounting structure adjacent to the assembly. That is the thickness, density, and modulus of elasticity of the fixture primary structure should simulate the actual mounting structure. However it is not necessary to simulate the exact details of the mounting structure. One criterion for simulation of the mounting structure is to simulate the infinite system point impedance of the mounting structure. For example, if the mounting structure resembles a flat plate, one would design a multimodal fixture which basically consists of a flat plate with the same infinite point impedance as given by Eq. 1. In the case of the Mariner multimodal fixture, the primary fixture frame very closely resembles a three-bay section of the Mariner spacecraft (see Fig. 1).

After the primary structure of the fixture is constructed, secondary structure must be attached to the structure to enrich its modal density. The process of adding the secondary structure to enrich the modal density of the fixture is basically a cut-and-try process. The secondary structure utilized to enrich the modal density must be very rich in modes. Equations 2 and 3 which give the modal separation of plates and beams indicate that the modal enrichment structure must by nature be large in area or length and small in thickness. Thus the requirement for high modal density requires that the secondary structure be of a rather flimsy and flexible nature. However, in order for the secondary structure to couple into the primary structure and affect the response of the primary structure, the impedance of the secondary structure must be relatively well matched to that of the primary structure. The thickness and stiffness of the secondary structure at the attachment point must be similar to that of the primary structure in order to achieve good coupling.

Thus considerable trade-off is required in order to design the secondary structure so that it exhibits many modes and yet couples well in the primary structure. In the case of the Mariner multimodal fixture, it was found that wire-mesh and large plates constituted the best configuration for the secondary structure (see Fig. 1). Sine-sweep experiments should be conducted frequently throughout the course of the development of the multimodal fixture in order to determine the effectiveness of the various schemes for enriching the modal density.

After the modal density of the fixture has been enriched so as to simulate the mounting structure modal density or at least so that three or more modes in each third-octave band are realized, the damping of the fixture should be increased until the modal overlap condition given by Eq. 5 is achieved. In most cases this will require the addition of some exterior damping material in order to raise the fixture damping. Two possible means of increasing the damping of the fixture are to apply damping tape (for example, Scotch Brand Pressure Sensitive Tape No. 428A) or a sprayable damping compound (for example, Lord Manufacturing Company sprayable damping material no. LSD 501). In general it is possible to raise the fixture damping loss factor to a value of approximately 5×10^{-2}. In most cases, the damping material should be applied to the basic fixture structure rather than to the secondary modal enriching structure, because the application of damping to the secondary structure may inhibit the modal enrichment.

Care should be taken not to apply more damping than that required to achieve modal overlap, because after the modal overlap condition is achieved the addition of more damping increases the spatial and frequency variations of the fixture response.[5] The application of damping should be such as to minimize the frequency and spatial variations in the fixture acceleration. The frequency variations in the fixture response can be determined from sine-sweep tests, and the spatial variations determined from measurements made at a number of points on the fixture in octave-band random tests. The spatial and frequency variations in the multimodal fixture response at the control

points should be equal to or less than the spatial and frequency variations measured on corresponding points of the actual mounting system.

Excitation Configuration

The results of this investigation indicate that small shaker excitation is preferable to conventional slip table or rigid fixture excitation of the multimodal fixture. In order to excite the most modes of the fixture and to minimize spatial and frequency variations in the fixture response, the output of the small shakers should ideally be uncorrelated. In order to insure that the output of the shakers are uncorrelated at high frequencies, it is probably best to drive each shaker from an individual source. Engineering judgment must be exercised in order to determine the number and size of the shakers that should be used to excite the fixture. In the case of the present investigation two 25-lb force shakers were adequate to achieve a uniform reverberant acceleration response of a fixture of approximately 1 g rms over the frequency range from 100 to 2000 Hz.

In JPL spacecraft programs it would probably be preferable to use slightly larger shakers than those used in this investigation. The small shakers used in the JPL modal vibration tests would be ideal. A sufficient number of shakers should be used in order to insure that the fixture response in the immediate vicinity of the shakers is not significantly larger than the response at points far removed from the shaker attachment points. The number of shakers required will of course depend on the amount

of damping present in the multimodal test fixture, a highly damped structure will require a more uniformly distributed excitation. In order to insure that the vibration modes of the multimodal fixture are not constrained or inhibited, it is desirable to support the multimodal fixture on a soft cushion or spring mount, or to suspend the fixture from wires in a free-free configuration.

Specification and Control

The vibration environment in multimodal fixture tests can be specified in one of three ways. One can specify: 1) the space-average reverberant acceleration level on the multimodal fixture, 2) the space-average acceleration level at the fixture-assembly interface, or 3) the space-average acceleration response level of the assembly. The specification of reverberant vibration levels on the multimodal fixture is preferred to the specification at the interface or response levels because the interface and response levels are strongly dependent on the details of the assembly construction and the assembly mounting configuration. Unfortunately it is common practice in inflight vibration measurement programs to place the accelerometers at assembly-spacecraft interfaces (for contractual or other reasons) or to place the accelerometers on assemblies at points of particular concern. In cases where measurements of this type constitute the only link between the vibration test and the flight environment, it may be necessary to control interface or response measurements in multimodal fixture tests.

It is preferable to specify reverberant vibration levels on the multimodal fixture because these levels are rather insensitive to the details of the assembly mounting conditions or to the slight changes in the assembly construction or weight. Also acceleration measurements on uniform multimodal structure tend to exhibit less spatial and frequency variation than measurements at interfaces or at particular points on the assembly, and therefore represent more reliable and meaningful data. In the case of a Mariner assembly multimodal fixture, it would be desirable to specify the space-average acceleration level on the fixture backplates adjacent to the test assembly. Unfortunately it is not common practice to measure inflight acceleration levels on the reverberant structure such as the launch vehicle skin, a cylindrical adapter, or an assembly backplate. Thus for multimodal fixtures to be used in a most advantageous way will necessitate a change in the basic philosophy of measuring inflight high-frequency vibration environments.

The requirement that multimodal fixture vibration environments be controlled in terms of space-average acceleration levels requires that the vibration environment of the fixture be measured at a number of points simultaneously. Therefore it is recommended that multimodal fixture test control be based on the average (or in the case of sinusoidal tests, commutated averages) of a number of accelerometers mounted on the multimodal test fixture. It is difficult to formulate a quantitative criterion for how many control accelerometers should be used. In the case of the Mariner Spacecraft Assembly, we would recommend three control accelerometers on each of the two backplates if the backplate level is specified or

one accelerometer on each of the four feet if the foot level is specified. It is also recommended that control and equalization be based on 1/3-octave frequency bands rather than constant frequency bands such as 25 Hz.

Data Analysis

It is recommended that sine-sweep excitation of multimodal fixtures be limited to design or diagnostic test applications only, and that environmental simulation tests with multimodal fixtures be performed with 1/3 octave-band, octave-band, or broadband excitation.

The data obtained in the environmental simulation tests should be analyzed in 1/3 or full octave bands. This type of analysis has been investigated (Reference 6), and empirical factors have been determined for relating the results of broadband analyses to the peak statistics obtained from narrow-band analyses.

Possible Problem Areas

Since the concept of multimodal fixture vibration testing is relatively new, some problems with which the vibration test engineer is unfamiliar may be encountered. The first problem area concerns the frequency limitations of the multimodal vibration testing concept. It must be clearly understood that the multimodal concept is valid only in the high-frequency regime, and if a multimodal fixture is used in the low-frequency regime in the vicinity of its first few resonances the problems and difficulties encountered will be very similar to those when conventional rigid fixtures are used at

high-frequencies in the vicinity of their first few resonance frequencies. Thus some engineering judgment is necessary to determine which frequency range should be covered by conventional fixture tests and which frequency range should be covered by a multimodal fixture test. The frequency range for the multimodal fixture developed for the Mariner assemblies is 100-10,000 Hz.

A second problem area involves the efficiency associated with multimodal fixture excitation of the assembly. The vibration test engineer may complain that multimodal fixtures are less efficient in exciting the high-frequency vibration response of the assembly than conventional fixtures. This is certainly true. However one must realize that inflight mounting structures are also less efficient exciters of high-frequency vibration of spacecraft assemblies than conventional fixtures, and in this sense the multimodal fixture provides a realistic excitation efficiency. If the multimodal fixture is properly designed, one should have no difficulty in establishing inflight vibration levels with some reasonable factor of safety on the spacecraft assembly. In rare cases where it is desired to test the assembly to failure rather than to simulate the flight environment, the efficiency obtainable with a multimodal test fixture may be inadequate.

A third problem area is associated with the reliability and repeatability of test results. If the multimodal test fixture is properly designed to eliminate rattles and local failures, the vibration test data obtained with the multimodal fixture setup should be more reliable and repeatable than high frequency data obtained with conventional fixtures.

Other problems may arise because of the fact that multimodal fixture testing technology is not standardized and as well established as conventional rigid fixture technology. However if one follows the basic philosophy of designing a multimodal fixture which simulates the inflight spacecraft mounting structure, this problem should be minimized.

A fifth problem area concerns the durability and fatigue resistance of multimodal fixtures. Since multimodal fixtures are lightweight and of rather flimsy construction by nature, they will not withstand the stress levels and long-life abuse that a simple rigid fixture will withstand. However since multimodal fixtures are designed for high-frequency use, the stress levels in a properly designed fixture will ordinarily be well below those required to cause fatigue failure.

CONCLUSIONS AND RECOMMENDATIONS

The results of this investigation indicate that it is possible to design a multimodal vibration test fixture for performing high-frequency vibration tests of spacecraft assemblies, and that the multimodal test fixture approach alleviates many of the significant problems associated with the use of conventional rigid test fixtures for high-frequency vibration tests.

The specific conclusions that can be drawn from this investigation are:

1) A multimodal test fixture for performing assembly-level vibration tests can be designed to operate successfully at frequencies above 100 Hz (Fig. 6).

2) The multimodal test fixture concept eliminates the overtesting problem associated with rigid fixtures at high frequencies and provides a vibration environment on the spacecraft assemblies which simulates the assembly vibration environment in spacecraft-level tests (Fig. 11).

3) The use of several small shakers to excite the multimodal fixture is preferable to mounting the multimodal fixture and assembly on a conventional slip table or rigid fixture (Figs. 9 to 12).

4) At high frequencies, the spatial variations in the response of spacecraft assemblies in multimodal fixture tests is much less than the spatial variations in assembly response in high-frequency tests utilizing conventional rigid fixtures (Fig. 13).

5) Multimodal fixtures offer the advantage that an omnidirectional vibration environment is generated at the fixture-assembly interfaces in a single test.

On the basis of the results of this investigation it is recommended that:

1) Multimodal vibration test fixtures be constructed and utilized in assembly-level high-frequency vibration tests of spacecraft assemblies in future spacecraft development programs conducted by the Jet Propulsion Laboratory, Government agencies, and other aerospace contractors.

2) Other research be conducted to assess the feasibility of utilizing multimodal test fixtures for high-frequency transient vibration and shock tests of spacecraft assemblies.

GLOSSARY

modal density - the number of vibration modes of a structure with resonant frequencies in a frequency band divided by the frequency bandwidth.

modal separation - the average frequency interval between modal resonance frequencies, the reciprocal of modal density.

multimodal - exhibiting three or more vibration modes with resonant frequencies in a third-octave band.

reverberant - exhibiting many lightly damped vibration modes.

mean-square acceleration - the time-average of the squared acceleration at a particular point on the structure.

space-average acceleration - the average of the mean-square acceleration measurements at a number of points on a uniform structure.

spatial variance - the variance of the mean-square acceleration measurements at a number of points on a uniform structure.

frequency variation - the peak-to-valley amplitude of sine-sweep response data.

transfer function - the ratio of the vibration environment on an indirectly excited section of structure to the vibration environment on the exciting section of structure.

point force impedance - the force produced on a point on the structure by a sinusoidal velocity source of unit amplitude applied at the same point.

average point force impedance - the spatial and frequency average of the point force impedance measured at a number of points and at a number of different frequencies on a structure.

REFERENCES

1. "Comparison of Fixture and Spacecraft Vibration Tests of a Mariner C Electronic Assembly", Bolt Beranek and Newman Report 1384, submitted to the Jet Propulsion Laboratory, Pasadena, California, July 1966.

2. M. A. Heckl, "Compendium of Impedance Formulas", Bolt Beranek and Newman Report 774, submitted to the Office of Naval Research, 26 May 1961, Eq. III-2a.

3. E. E. Ungar and T. D. Scharton, "Analysis of Vibration Distribution in Complex Structures", Bulletin 36, Part V, Shock and Vibration Bulletin, January 1967.

4. R. H. Lyon, "Statistical Analysis of Power Injection and Response in Structures and Rooms", Bolt Beranek and Newman Inc. Report No. TIR-65, 14 June 1967, Eq. 69C.

5. Reference 4, p. 13.

6. M. C. Trummel and D. B. Wiksten, "A Broad-Band Versus Narrow-Band Measure of Spectral Peaks in Vibration Data", paper presented at the 74th meeting of the Acoustical Society of America, Miami, Florida, November 1967.

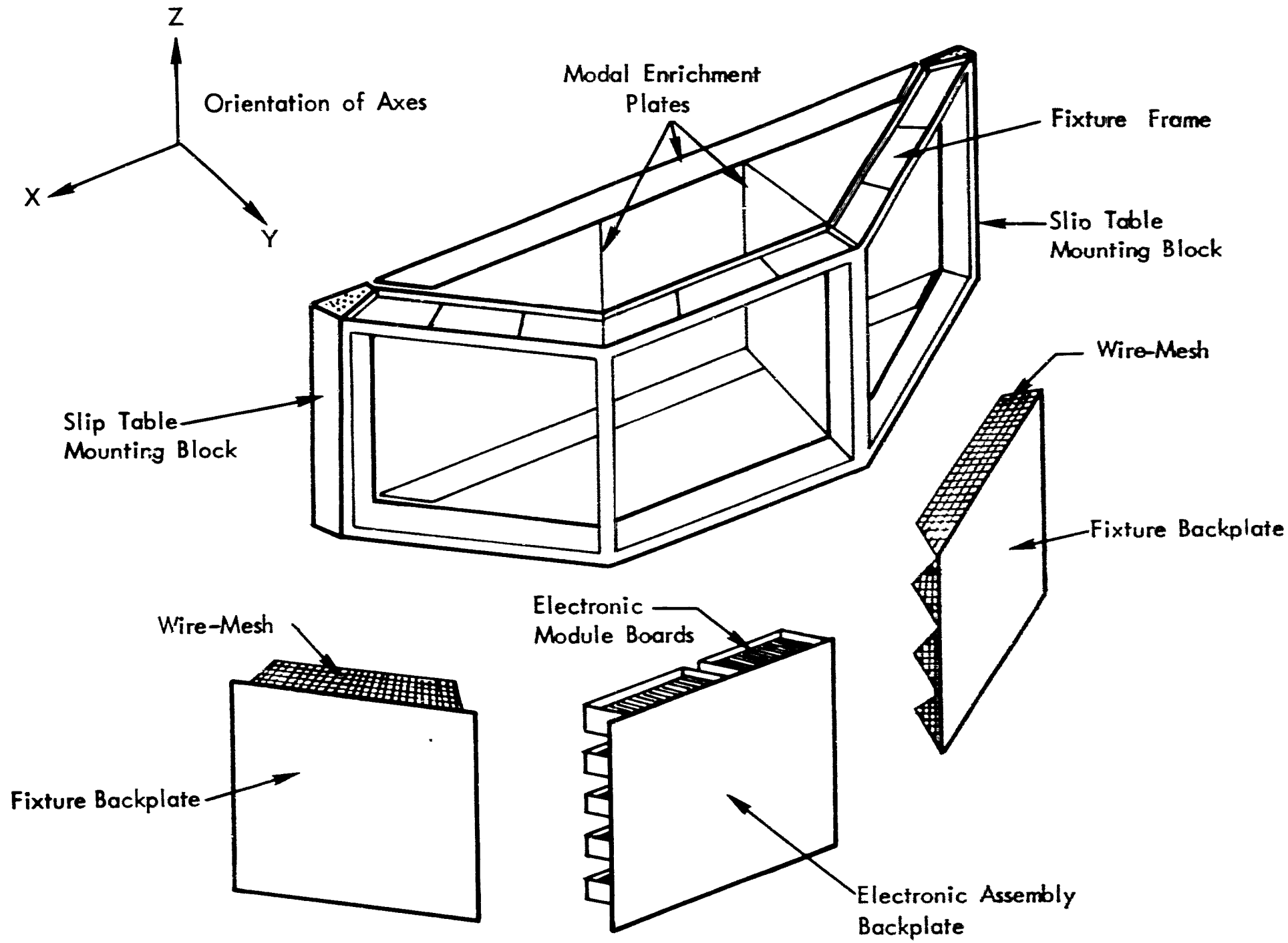
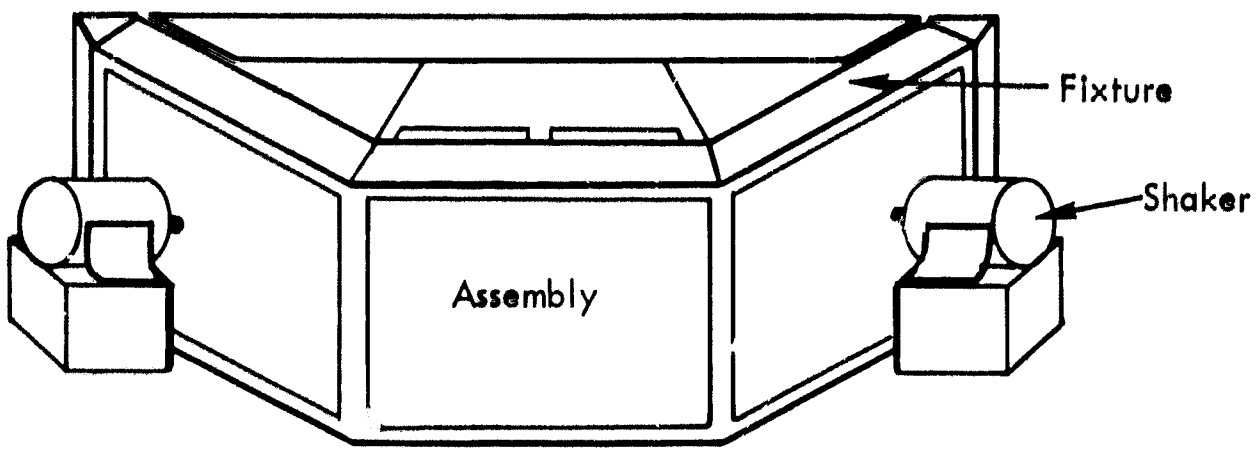
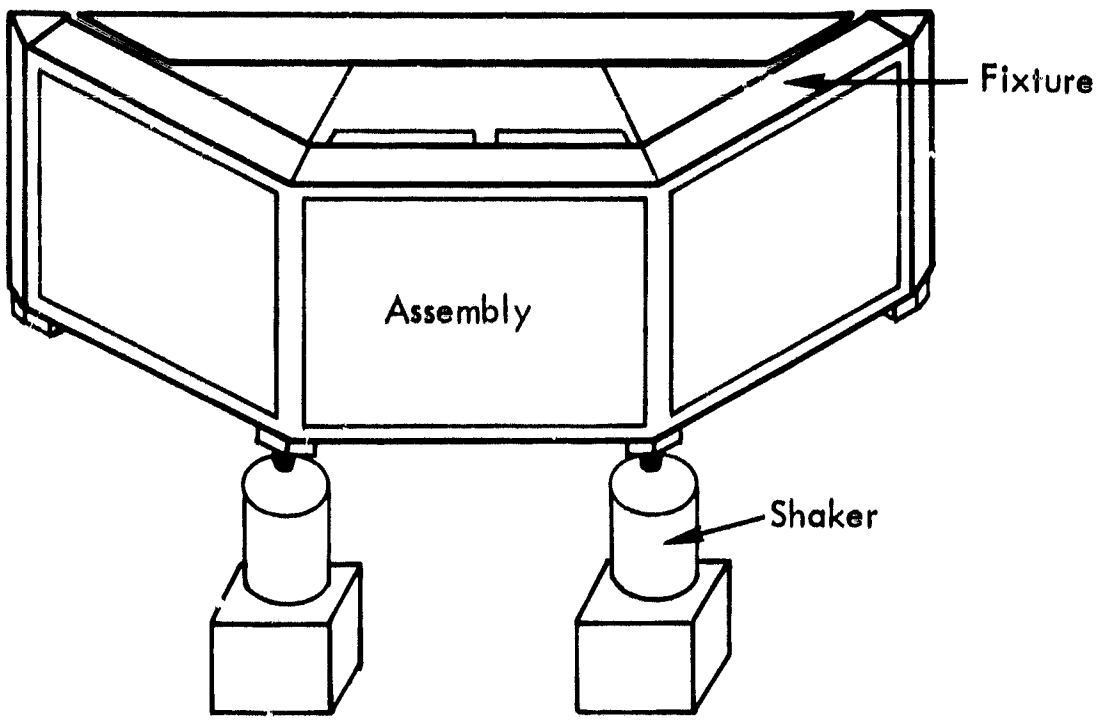


FIGURE 1. PERSPECTIVE VIEW OF MARINER '69 SPACECRAFT ELECTRONIC ASSEMBLY AND MULTIMODAL FIXTURE CONSTRUCTION



(a) SMALL SHAKERS ATTACHED TO BACKPLATES



(b) SMALL SHAKERS ATTACHED TO FEET

FIGURE 2. MOUNTING CONFIGURATIONS OF MULTIMODAL FIXTURE EXCITED WITH SMALL SHAKERS

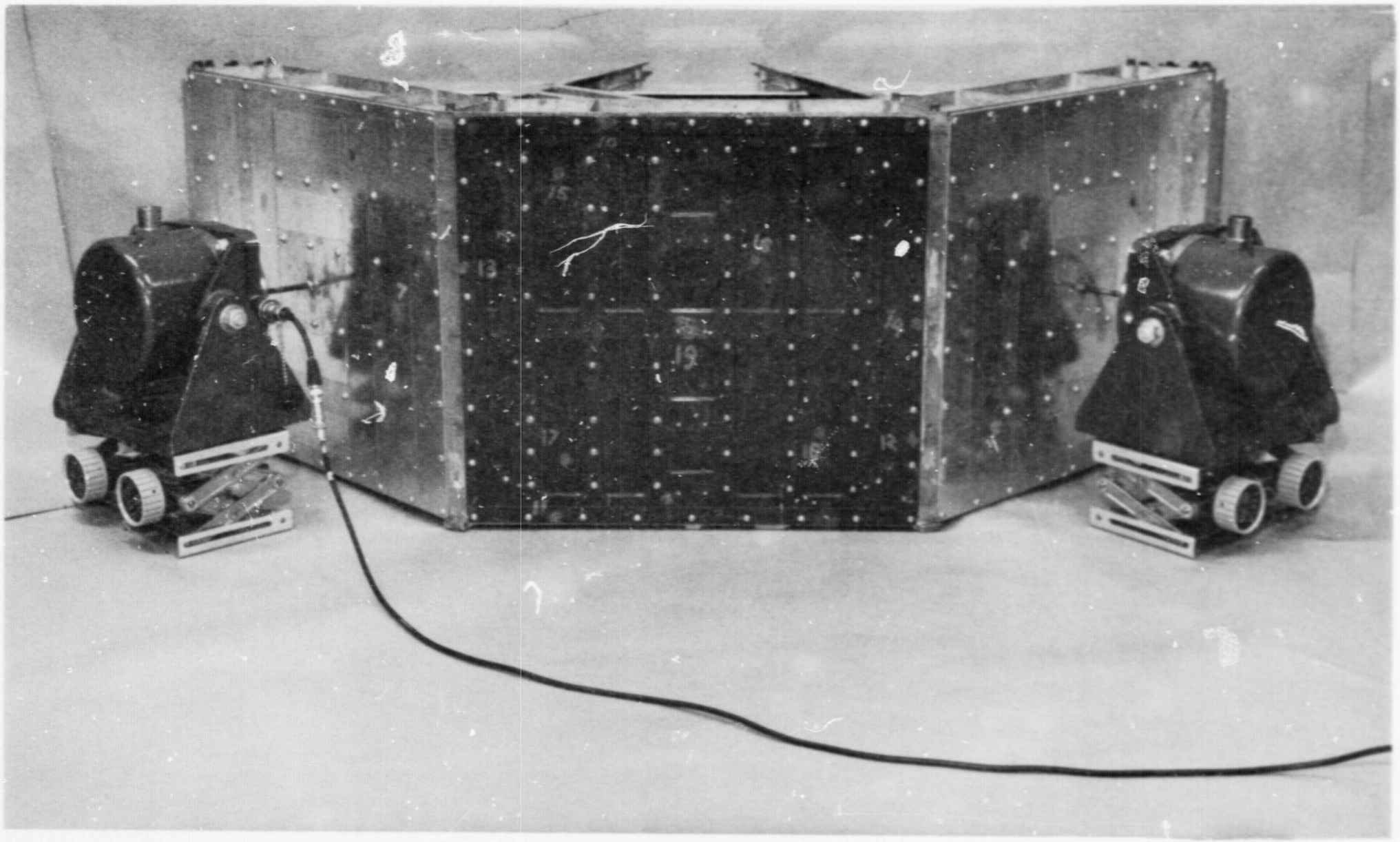


FIGURE 3. ELECTRONIC ASSEMBLY AND MULTIMODAL FIXTURE EXCITED WITH SMALL SHAKERS ATTACHED TO FIXTURE BACKPLATE

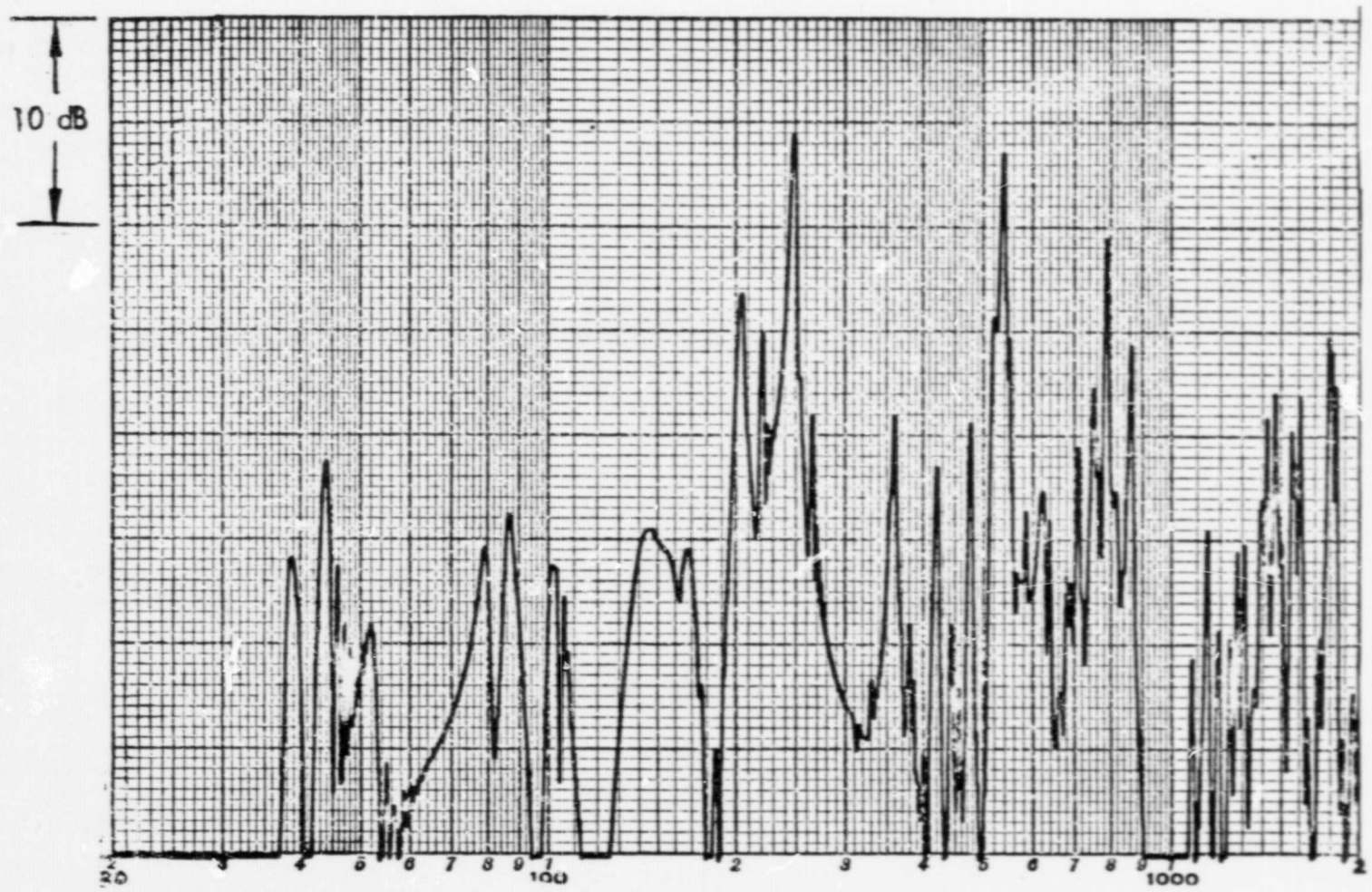


FIGURE 4. FRAME EXCITED WITH SMALL SHAKERS ATTACHED TO FEET

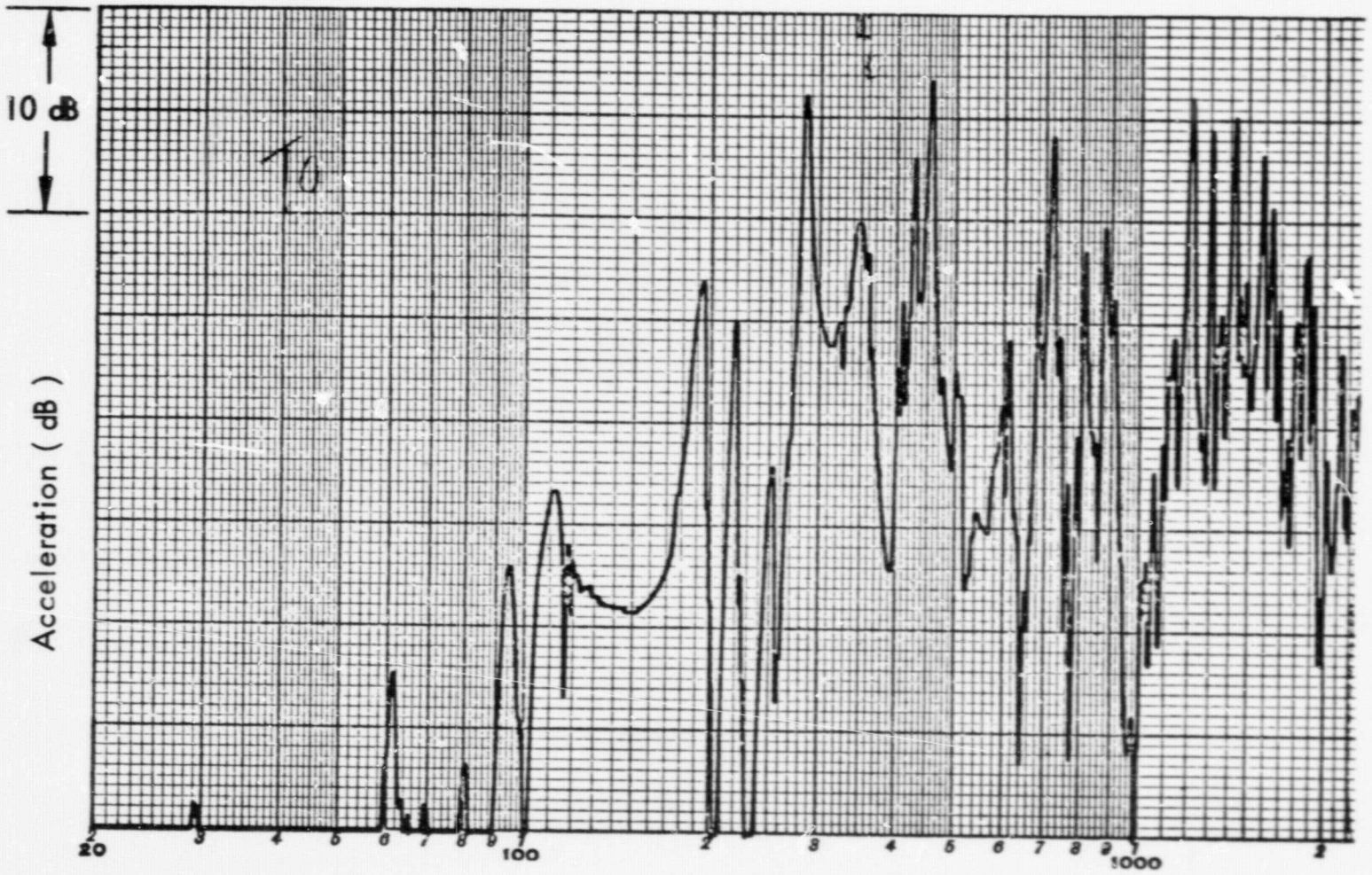


FIGURE 5. RESPONSE OF FIXTURE FRAME WITH BACKPLATES ADDED
(Small Shaker Excitation at Fixture Backplates)

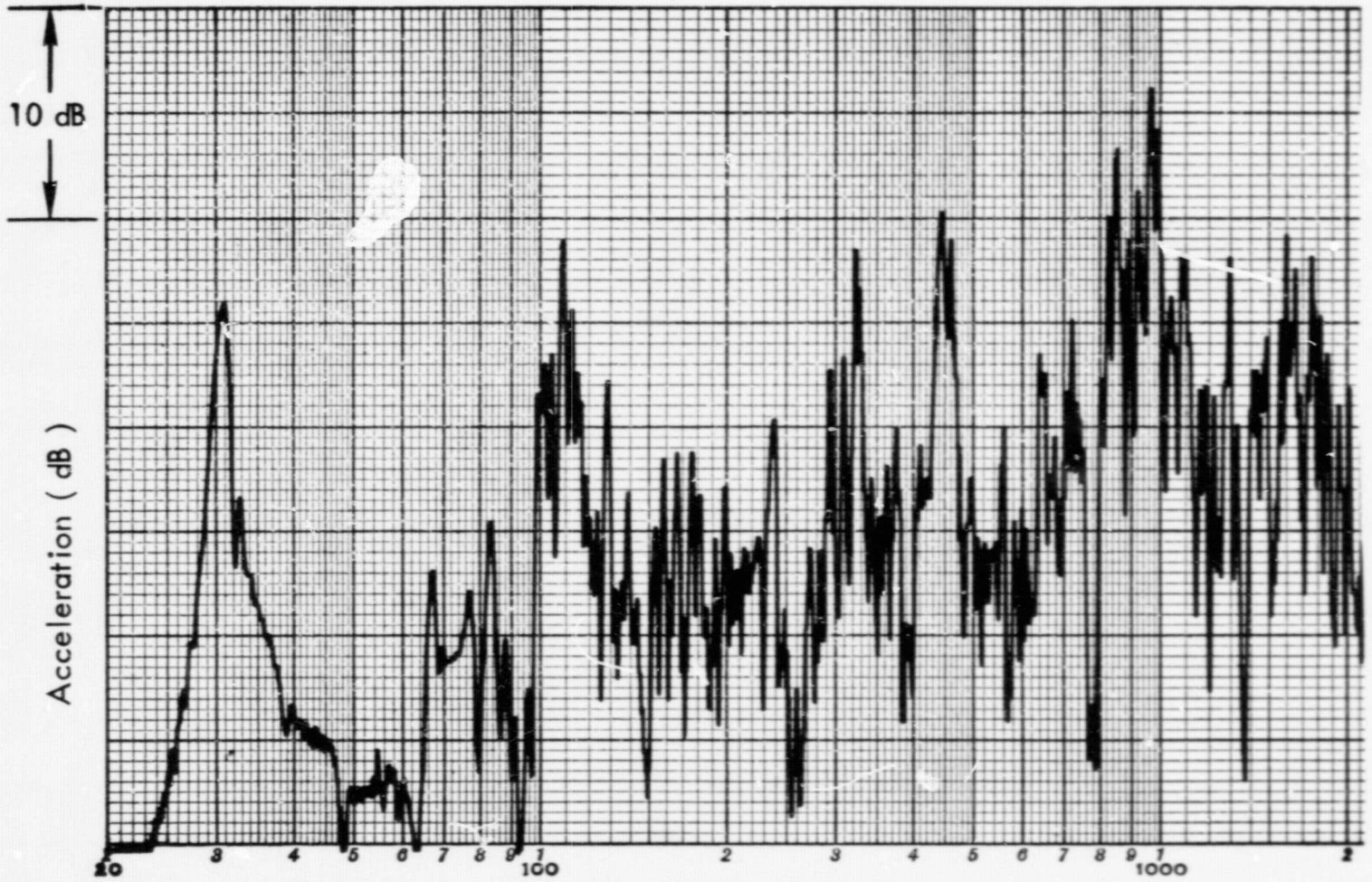
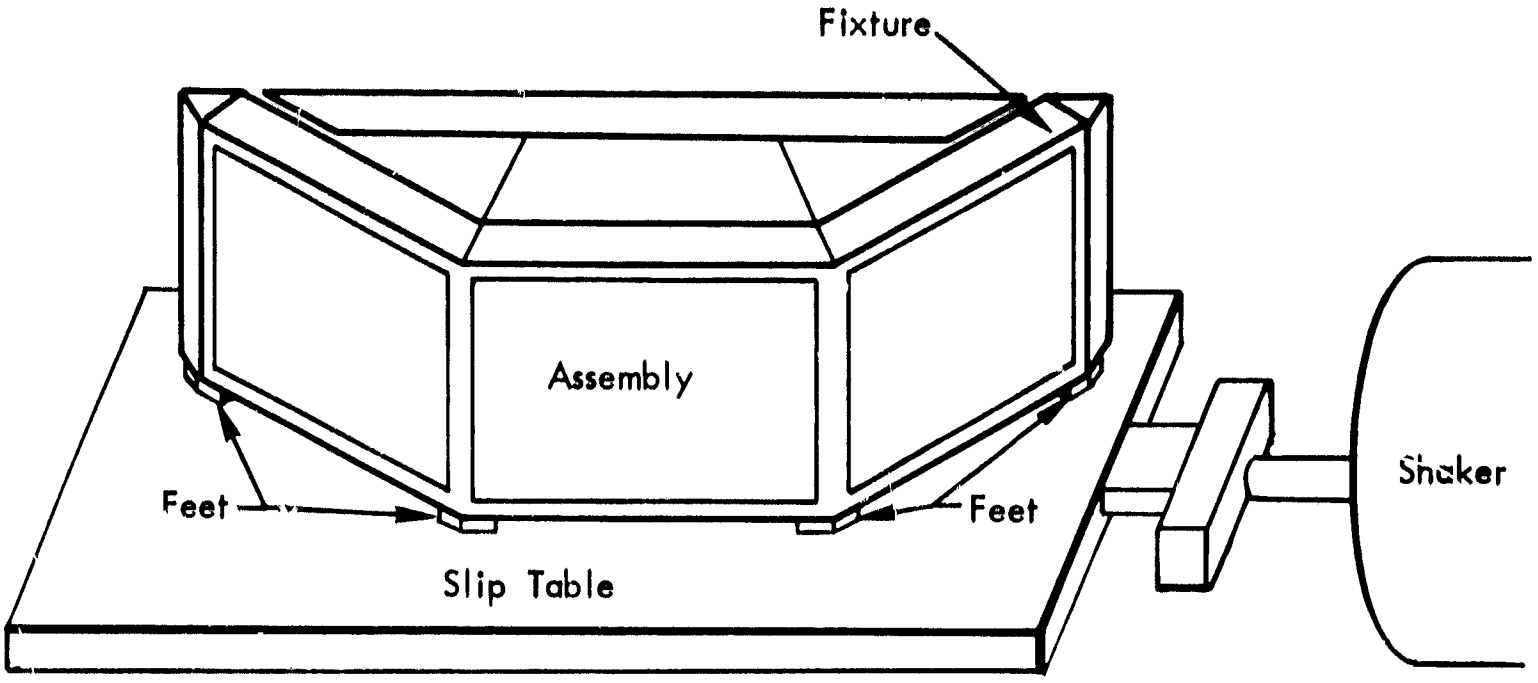
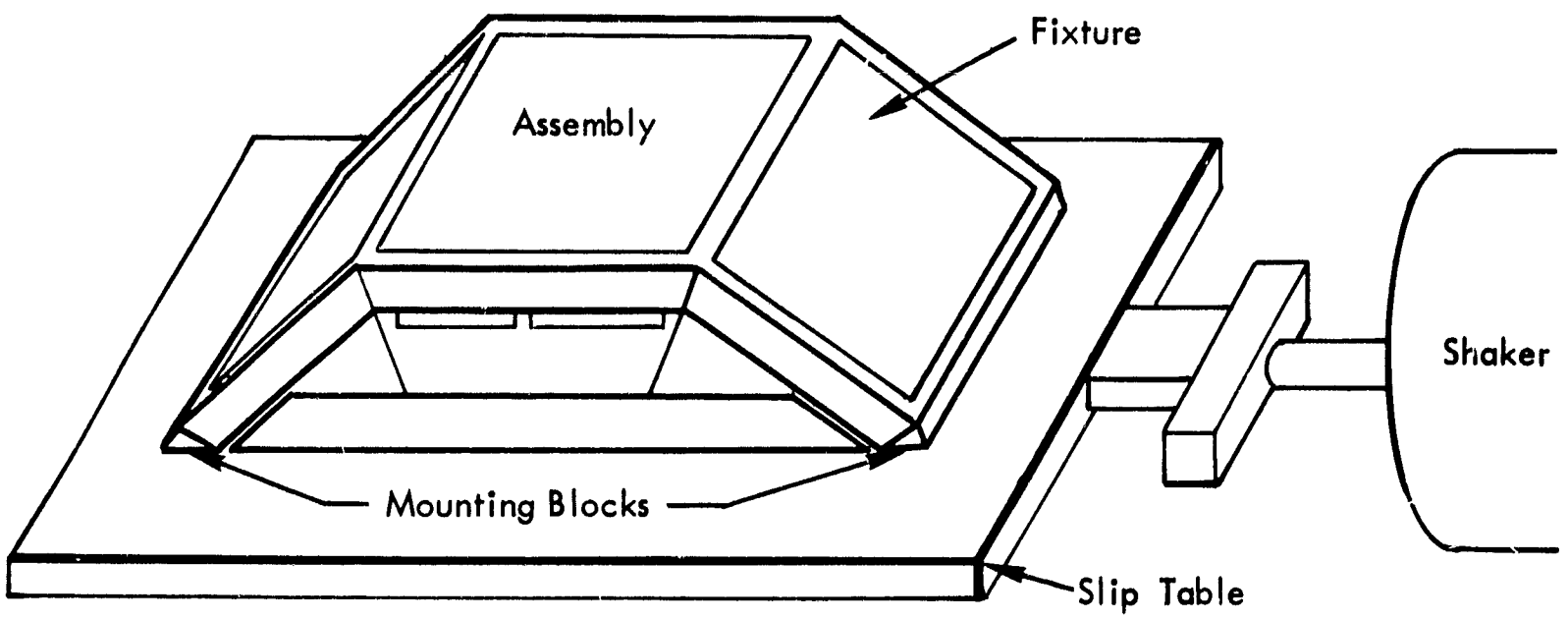


FIGURE 6. RESPONSE OF FIXTURE FRAME WITH BACKPLATES, MODAL ENRICHMENT PLAYES, AND WIRE-MESH ADDED
(Small Shaker Excitation at Fixture Backplates)



(a) SLIP-TABLE EXCITATION AT FEET



(b) SLIP-TABLE EXCITATION AT MOUNTING BLOCKS

FIGURE 7. MOUNTING CONFIGURATIONS OF MULTIMODAL FIXTURE EXCITED WITH CONVENTIONAL SLIP TABLE

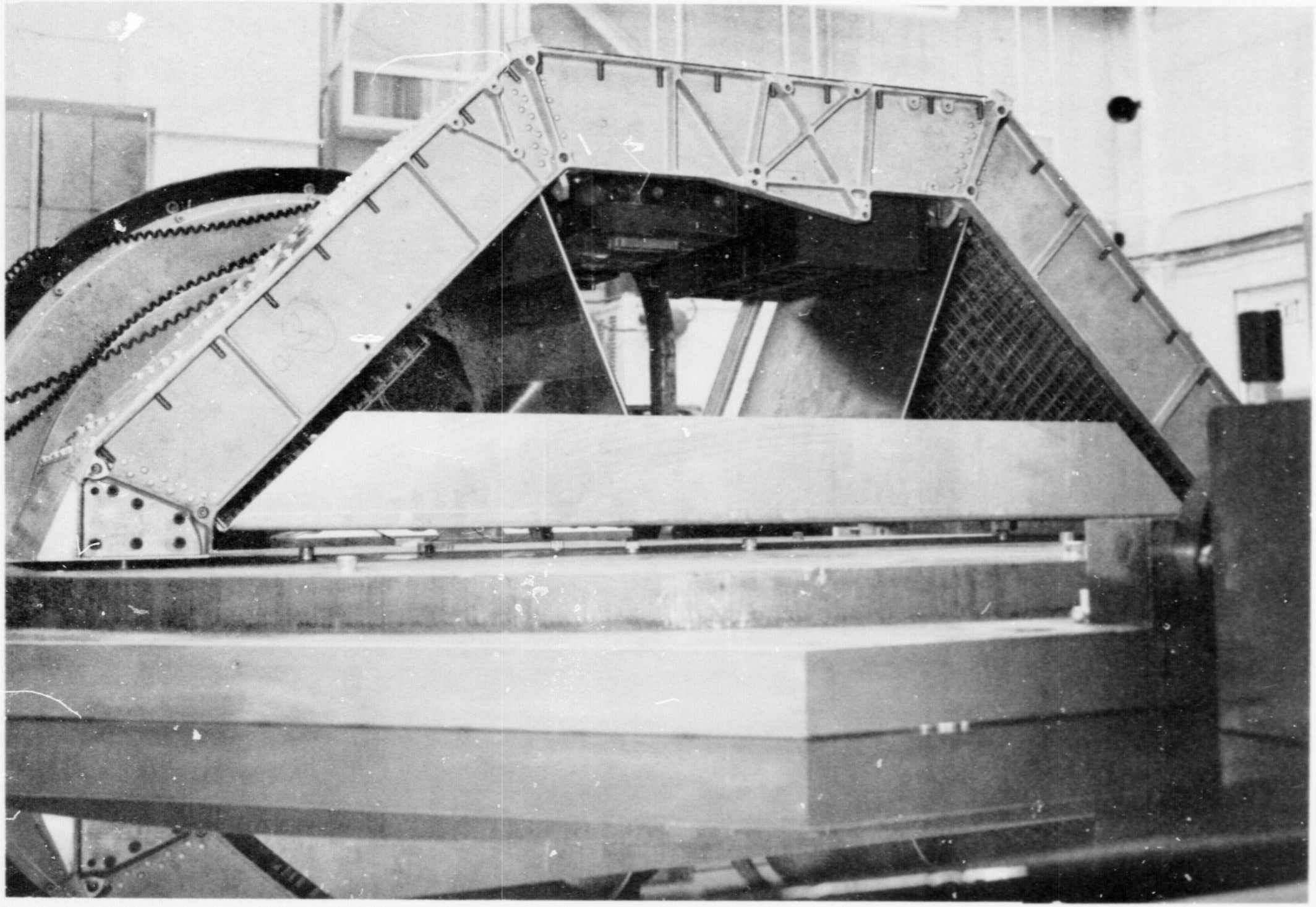


FIGURE 8. ELECTRONIC ASSEMBLY AND MULTIMODAL FIXTURE EXCITED WITH SLIP-TABLE AT MOUNTING BLOCKS

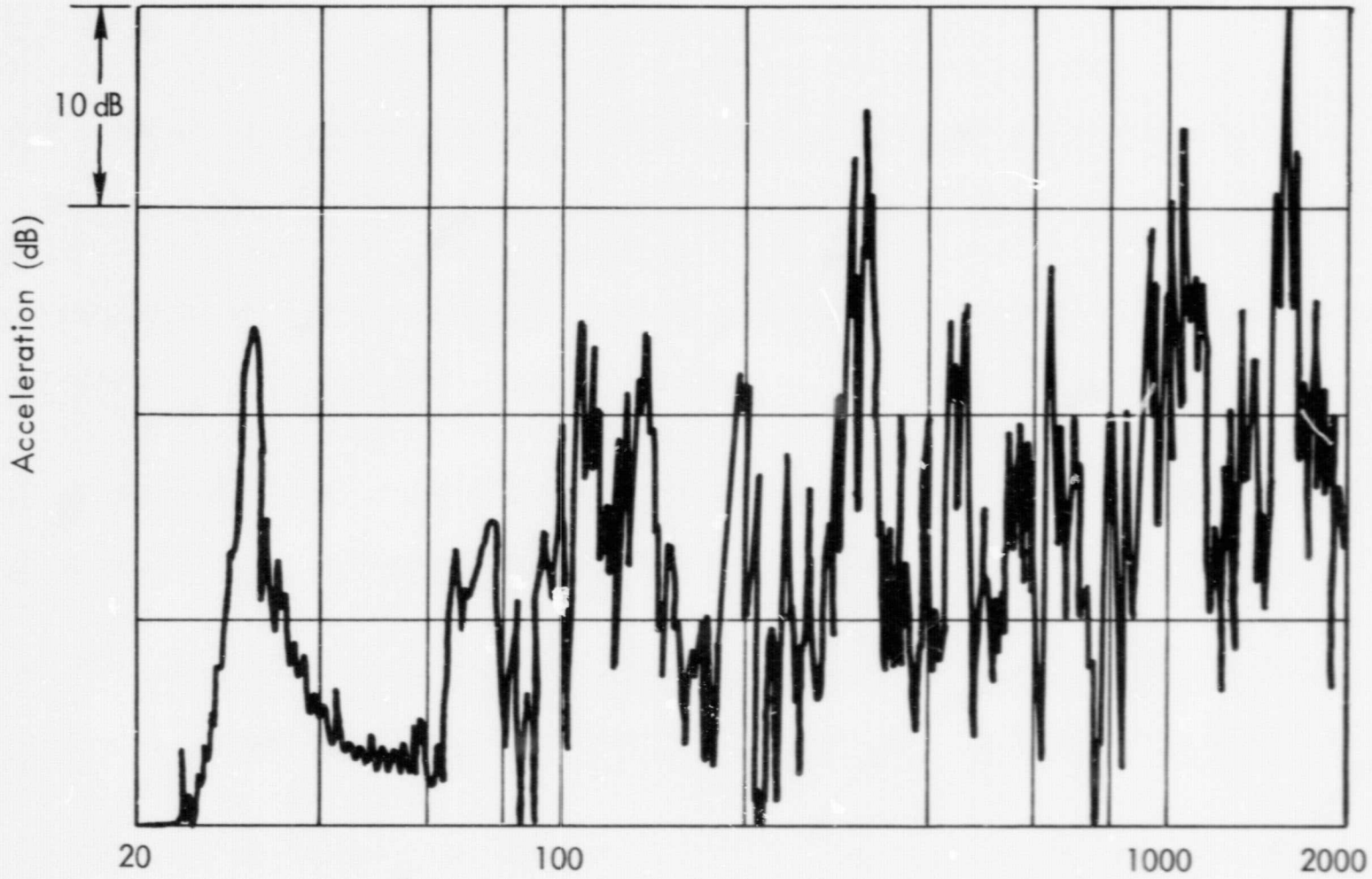


FIGURE 9. RESPONSE OF SPACECRAFT ASSEMBLY BACKPLATE WITH MULTIMODAL FIXTURE EXCITED WITH SMALL SHAKERS ATTACHED TO FIXTURE BACKPLATES

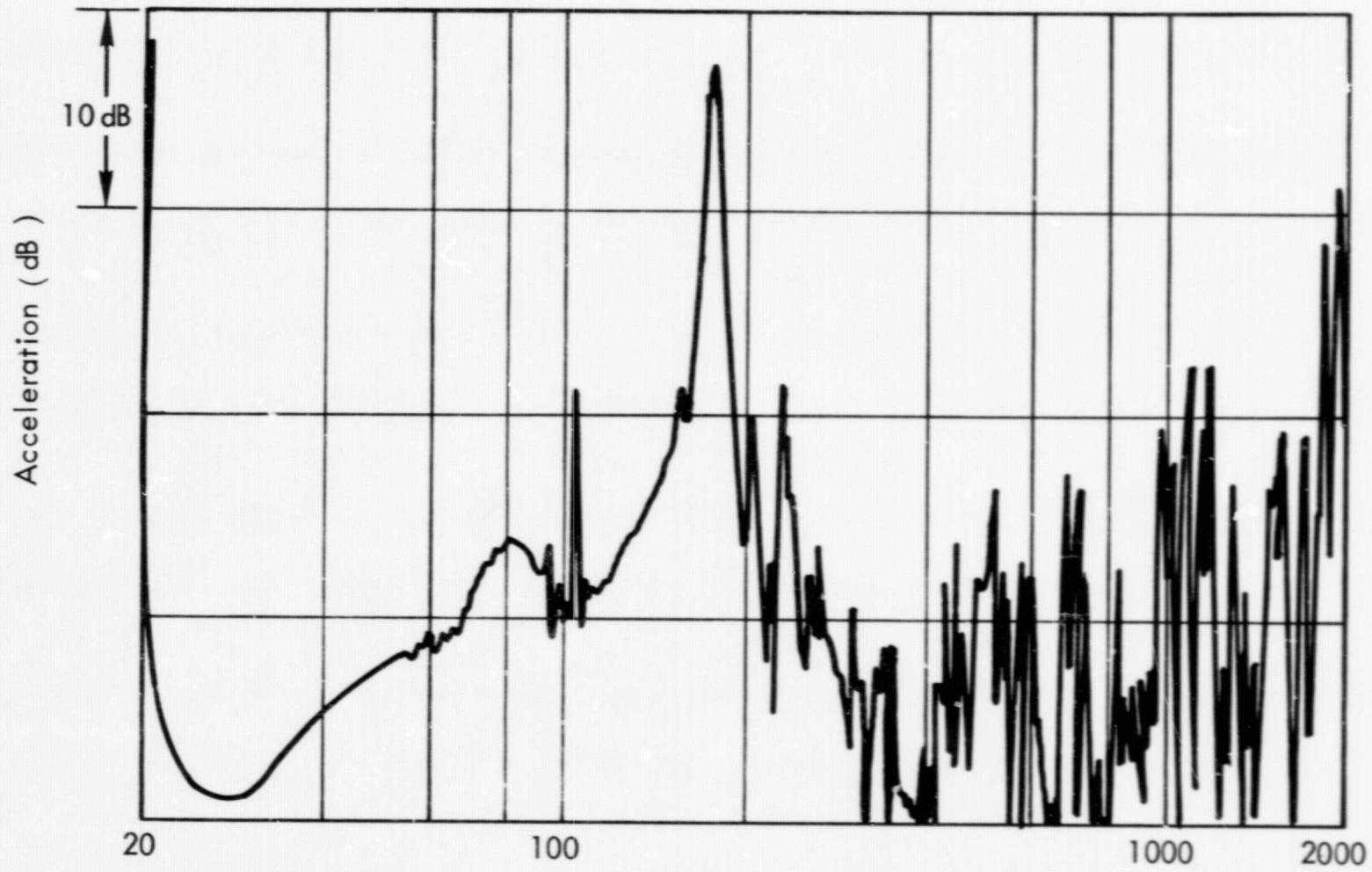


FIGURE 10. RESPONSE OF SPACECRAFT ASSEMBLY BACKPLATE WITH MULTIMODAL FIGURE EXCITED WITH CONVENTIONAL SLIP-TABLE AT MOUNTING BLOCKS

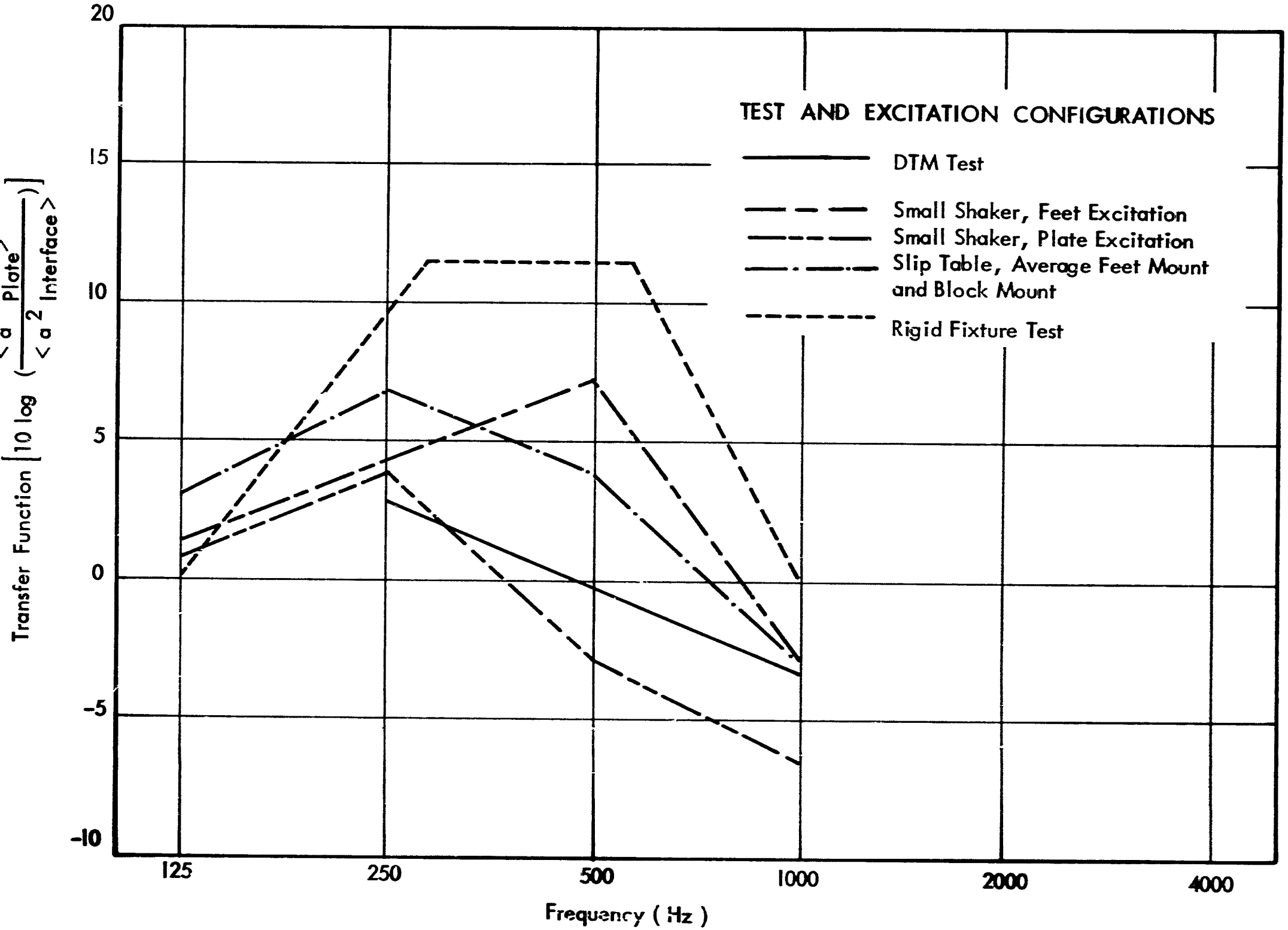


FIGURE 11. TRANSFER FUNCTIONS FROM FIXTURE-ASSEMBLY INTERFACE TO ASSEMBLY BACKPLATE

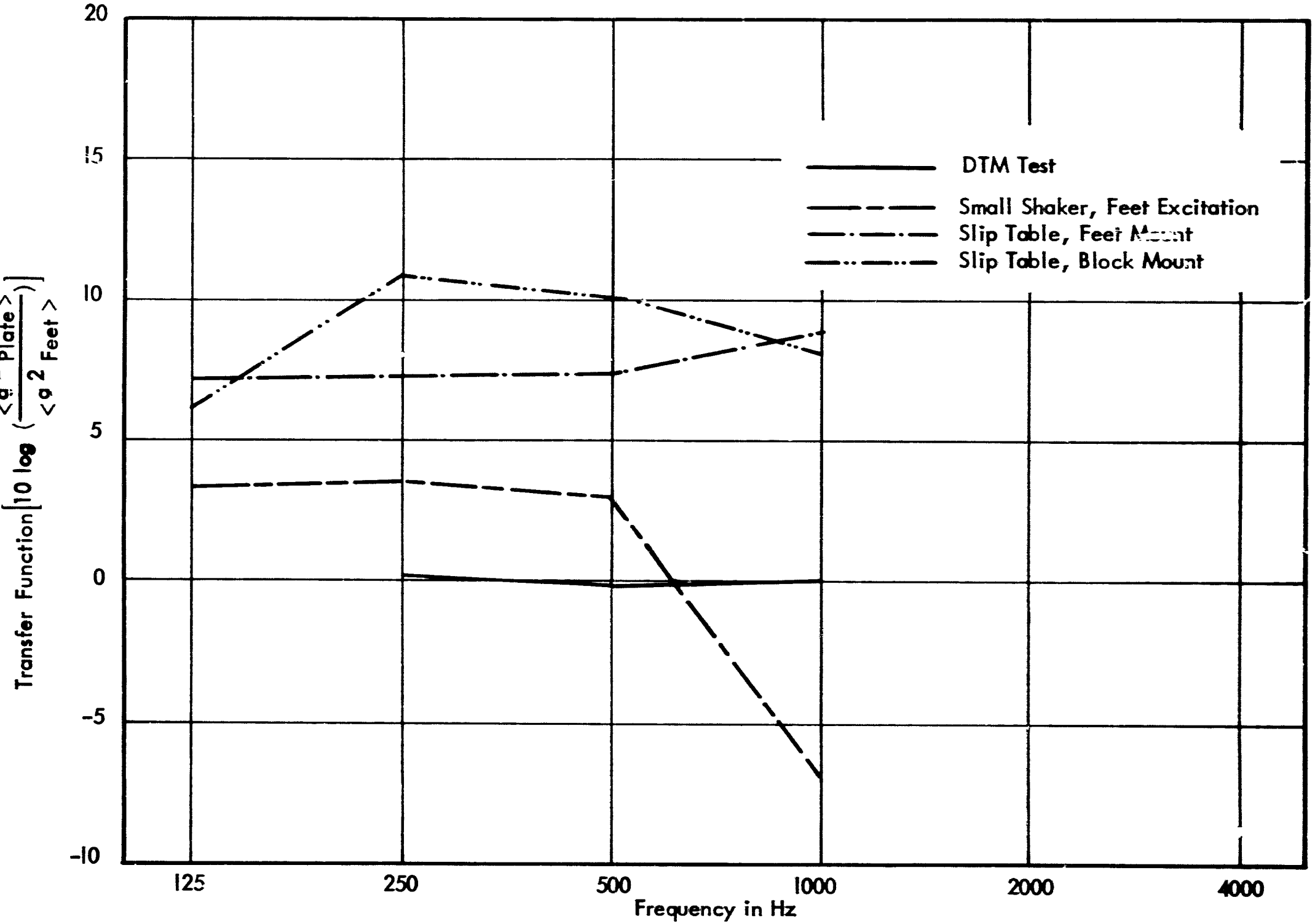


FIGURE 12 . TRANSFER FUNCTIONS FROM FIXTURE FEET TO ASSEMBLY BACKPLATE

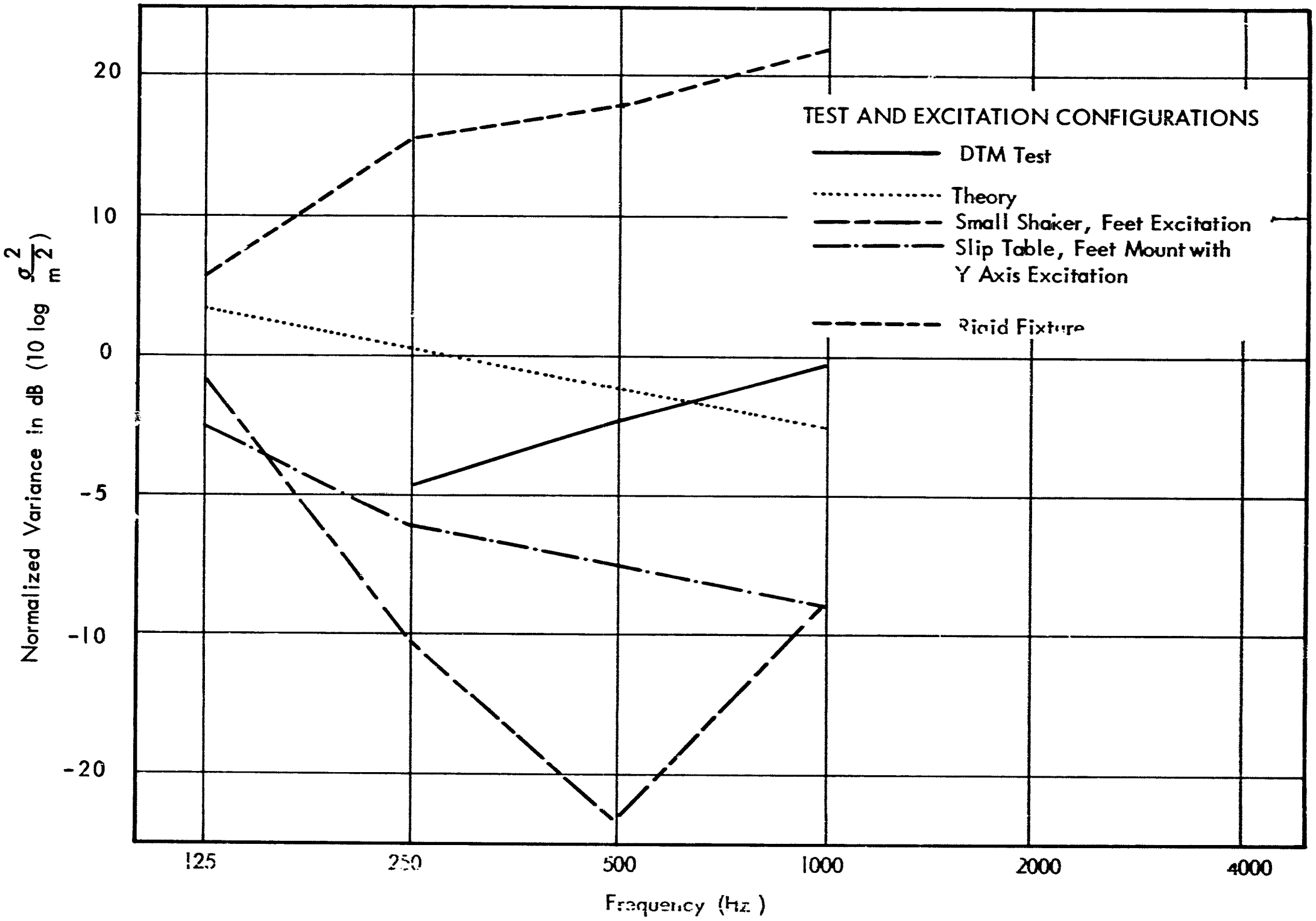


FIGURE 13. SPATIAL VARIANCES OF ASSEMBLY BACKPLATE MEAN-SQUARE ACCELERATION RESPONSES