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Final Report

SHOCK AND VIBRATION TRANSPORTATION ENVIRONMENTAL CRITERIA

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ABSTRACT

A comprehensive literature search was conducted for existing information and data describing shock and vibration environments in the four major modes of transportation (rail, truck, air, and water). Data specifically concerned with the transportation of rockets and allied equipment was sought.

Over 300 reports were reviewed and contacts were made with 55 agencies and organizations active in the transportation field. Based upon the data collected, acceleration versus frequency diagrams were constructed which describe the inputs to the transported package for each of the four major transportation modes. These diagrams facilitate the comparison of environments in the different modes of transportation. A number of acceleration versus frequency diagrams within the four major mode classifications were also constructed. These curves show the effects of various operating parameters on the transportation environment.

Shock spectra are presented to describe the railroad coupling environment. Spectra are given for various impact speeds for both standard and high capacity draft gears. A curve showing the variation of railroad car speeds during coupling operations is also presented.

Statistical techniques were not employed to describe the transportation shock and vibration environments because of the paucity of data.

The problems which arise in the interpretation of the various forms of data are discussed together with the applicability of the data to package design.

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SECTION 1

INTRODUCTION

The transportation shock and vibration environment is defined as the motions at the points where the cargo is supported in a transport vehicle. The severity of this environment will be influenced by operating factors such as speed, road conditions, sea conditions, air turbulence, maneuvers, etc.

This criteria describes the environment of the four major modes of transportation: railroad, truck, ship and aircraft. The descriptions apply to commercial vehicles traveling normal routes.

Packaging and design engineers require an accurate description of the transportation shock and vibration environment. The information enables packaging engineers to determine if an item being shipped will be subjected to a shock and vibration environment severe enough to cause damage. If protection is required the information can be used in the design of shock isolation systems. Further, a knowledge of the transportation shock and vibration environment in the early stages of development of an item may enable it to be designed to withstand the transportation environment without the need of expensive packaging.

SECTION 2

STATE OF THE ART

Extensive programs have been conducted to measure the shock and vibration environment on vehicles. However, only a small amount of the available data may be used to describe the commercial transportation environment. The reasons for this are: (1) the measurements were made to determine the structural integrity of a vehicle or package and have no general application, (2) the measurements were made on obsolete, or special purpose vehicles, and (3) the measurements were not adequately defined or other factors of importance were not reported.

Field measurements include both shock and vibration. Shock is defined as transient motion of short duration. Vibration is defined as an oscillatory motion of relatively long duration. Since there is no precise distinction between these two types of motions, they have been described in different ways by different investigators.

2.1 Vibrations

Methods for recording and presenting vibration data vary from one program to another depending on the available instrumentation and on the desired degree of sophistication. In early programs, vibrations were generally recorded with an oscillograph and visually analyzed. The range of frequencies which could be recorded was limited either by the recording system or the transducers. These results, therefore, should not be interpreted as implying that vibration levels beyond the reported frequencies do not exist at significant levels. (In some programs, the frequency range was limited at the discretion of the investigator.)

The investigator analyzing a record would normally determine a maximum amplitude of vibration (acceleration, velocity, or displacement) determine a frequency associated with it, and record them as an amplitude and frequency. If the record consisted of two widely different frequencies, it sometimes would be possible to accurately estimate the amplitude and frequency of each component.

This method of analysis has the advantage that it provides immediate results and it requires minimum data reduction equipment. A disadvantage, however, is that it tends to overestimate amplitudes because amplitudes corresponding to transients or "beats" may be selected. Thus, the severity of the environment may be overemphasized.

With the development of the tape recorder, newer techniques were employed in analyzing vibration data. The data tape was generally processed by playing it back through sound or vibration analyzers. This provided a continuous vibration spectrum. Peaks on the spectrum were then assumed to represent sinusoidal components of the vibratory motion. The motion was described by assuming that it was a steady-state vibration composed of these discrete sinusoidal components.

A disadvantage of this method of analysis is the tendency to underestimate the severity of the environment because the occasional high levels of vibration are disregarded. Furthermore, the amplitude levels are of questionable accuracy especially when the spectrum peaks are not pronounced. A steady-state condition is implied even though this is usually not the case.

In more recent programs data has been analyzed with vibration analyzers employing narrow band (1/2 CPS) variable frequency filters. In addition to providing continuous spectra for the data, a distribution of peaks is determined within selected frequency bands. This method of analysis and presentation represents the most descriptive and accurate definition of the transportation environment currently available.

2.2 Shock

Descriptions of the vibration environment have been supplemented in many studies by specifying the occasional extreme values of excitation. These transient or shock motions have been described by either of two methods. In one method the actual motion is described. In the other method the peak responses are given for a series of single-degree-of-freedom systems, whose resonant frequencies cover the range of the dominant shock-excited frequencies.

The first method describes what the shock motion is. Usually the actual motion of a complicated shock disturbance requires graphical representation. This is unsatisfactory for ordinary use. Generally a simplified representation of the motion is given instead.

The second method, called the shock spectrum method, gives the effect of the shock motion on a series of single-degree-of-freedom spring-mass systems. A curve relating the responses of the systems as a function of their natural frequencies, for a given shock motion, is called a shock spectrum.

SECTION 3

SUMMARY OF AVAILABLE DATA

Most vibration data has been reported in terms of magnitudes of peak or RMS values of the filtered vibrations and the corresponding frequencies. This data may be summarized by enveloping the maximum vibration levels, either peak or RMS, for all reported frequencies. Any point on the envelope represents the maximum level of vibration which occurs at that frequency, although it is possible that the actual level may be much less. (If the vibration is random, the envelope of peak values gives the maximum levels with a confidence of at least 99%.) This is considered to be the best format for presenting the existing data and it is the one used in this report.

The only shock motions which are described independently of the normal transportation environment are those encountered in railroad car coupling. For this environment, with its correspondingly complex short-duration time histories, the shock spectrum is considered the most concise and descriptive format for presenting the data.

Summaries of the shock and vibration data for the four major modes of transportation are presented in the following paragraphs.

3.1 Aircraft

Extensive shock and vibration measurements on aircraft have been performed by the Wright Air Development Division (WADD). Their most recent test programs cover the following aircraft:

- C-123 Medium assault cargo airplane, high-wing, twin engine (reciprocating), 3 bladed propellers.
- C-130 Medium range cargo airplane, high-wing, four engine (turbo prop), 3 bladed propellers.
- C-133 Long range cargo airplane, high-wing, four engine (turbo prop), 3 bladed propellers.
- H-37 Cargo helicopter, single main rotor plus torque compensating tail rotor, twin engine (reciprocating), five main rotor blades and four tail rotor blades.

The measurements on these aircraft were taken during all of the normal service conditions such as taxi, ground run-up, take-off, straight and level flight (at various altitudes, speeds, and power settings), turn, descent, landing, and landing roll. The effects of cargo load, speed brakes, and other control surfaces were also investigated on some of the aircraft.

The vibrations in these tests were monitored with velocity pick-ups and recorded on magnetic tape. Data reduction was performed with a Davies Automatic Analyzer employing a variable-frequency, narrow-band (10 cps) filter. An unspecified number of peak values were then taken from each of the continuous spectrum curves and plotted as individual data points. The data for each aircraft has been summarized for presentation in this report by enveloping all of the data points obtained on the cargo floor under all operating conditions. The curves, therefore, should not be construed as continuous spectrum curves, but rather as envelopes encompassing the highest recorded vibration levels for all flight conditions. A point on any one of these curves represents the maximum rms level of vibration that has occurred at that frequency. That is, if a discrete sinusoid occurs at a given frequency, its maximum rms amplitude is given by the ordinate on the appropriate envelope corresponding to that frequency. Diagrams of the above data are plotted in terms of "g" (rms) versus frequency. They are shown in Figures 3, 5, 6, and 7 for the C-123, C-130, C-133 and H-37 respectively.

Vibration data for the KC-135, a military version of the Boeing 707 jet aircraft, has been obtained from tests performed by Boeing. The data covers ground run-up, taxi, take-off, and cruise conditions. The original Boeing report presents the vibration data in power spectral density (g^2/cps vs frequency). For presentation in this report the data has been converted to "g" (rms) versus frequency. In the Boeing tests the vibrations were monitored with accelerometers and recorded on magnetic tape. The data was analyzed with a Davies Analyzer using the following filters for the different frequency ranges:

Frequency Range (CPS)	Filter Bandwidth (CPS)
0-30	.80
30-50	1.33
50-100	2.64
100-200	5.41
200-400	10.1
400-800	18.7
800-1000	35.3
1000-2000	43.5

The summary diagram for this aircraft (Figure 8) envelopes the highest reported acceleration points for all flight conditions. This diagram, therefore, is an envelope curve of the maximum vibration levels, not a continuous spectrum plot.

Vibration data for the Pregnant Guppy (377PG), a low-wing, four engine (reciprocating), 4 bladed propeller cargo airplane is also included. Although a special aircraft, it is included because of its extensive use in transporting rockets and missile components. Measurements of the vibration environment on the cargo floor and on the cargo itself have been made during all shipments of rocket and missile components in this aircraft.

The vibration environment in these tests was monitored with accelerometers and the data recorded on magnetic tape. A variable frequency, one-half cycle per second bandwidth filter was used in the analysis of the data. Frequency bands having relatively high vibration levels were analyzed further. At these selected frequencies a distribution of the accelerations was determined. Data currently is available for a number of locations on the cargo floor, for a number of flight conditions, and for a number of loads. This data represents one of the most complete descriptions of the shock and vibration environment of any transport vehicle. Summary diagrams of the Pregnant Guppy in this report (Figure 4) envelop all data points for all flight conditions and loads. The plots are presented in terms of "g" (rms) versus frequency.

Envelopes depicting the environment for propeller aircraft, jet aircraft and helicopters were obtained by encompassing the maximum vibration levels for each classification. These diagrams are shown in Figure 1. The data for the C-123, C-130, C-133 and 377PG was used in developing the summary diagram for propeller aircraft while the data for the H-37 Helicopter and KC-135 jet aircraft was used to describe the environment for the helicopter and jet categories respectively. The diagrams show that the vibration levels are highest for the helicopter and lowest for the jet.

Directional composite diagrams for the individual aircraft were made by enveloping the data for each aircraft for all reported flight conditions, directions and loads. These diagrams are shown in Figure 2.

The individual directional aircraft diagrams were made by segregating the data with respect to direction of measurement. These results are shown in Figures 3, 4, 5, 6, 7, and 8 for the C-123, 377PG, C-130, C-133, H-37 and KC-135 aircraft. Examination of these diagrams show that the only aircraft demonstrating a significant effect of orientation on the environment is the Pregnant Guppy (Figure 4). For this aircraft the vertical vibrations are highest and the fore-and-aft vibrations are lowest.

3.2 Railroad

Data descriptive of the railroad shock and vibration environment has been categorized into two major classifications: over-the-road operation and coupling. The over-the-road environment includes all data except the shock motions associated with coupling or humping operations.

3.2.1 Railroad (Over-the-Road)

Acceleration versus frequency envelopes of the shock and vibration environment of railroad cars have been compiled from many sources. Because of the high-amplitude transient vibrations which occur during starts, stops, slack run-outs and run-ins, this data has been segregated, when specified, from the vibration data describing normal running conditions. Composite diagrams of this data are shown in Figure 9.

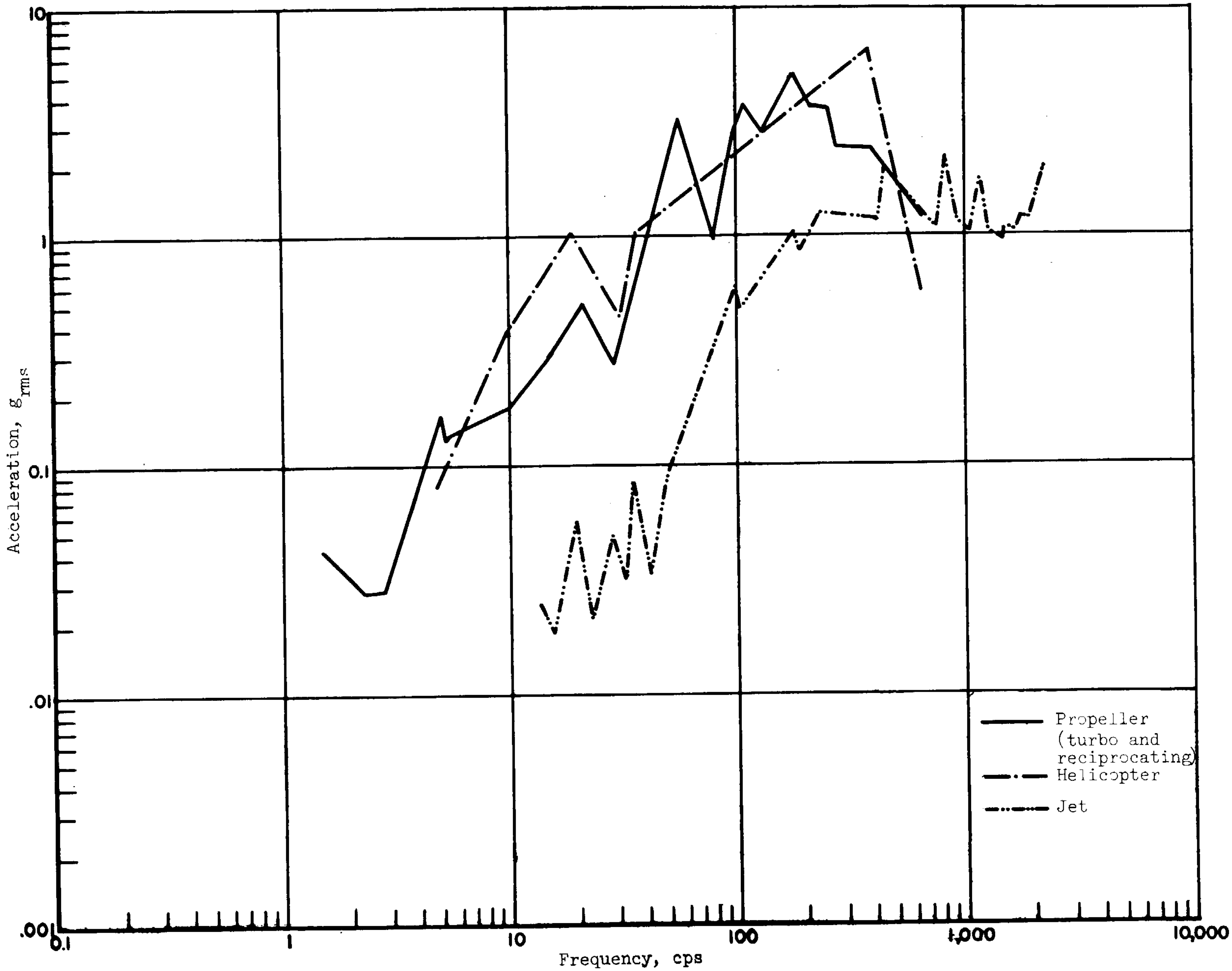


Figure 1. AIRCRAFT ACCELERATION ENVELOPES - COMPOSITE (Propeller, Helicopter, Jet)

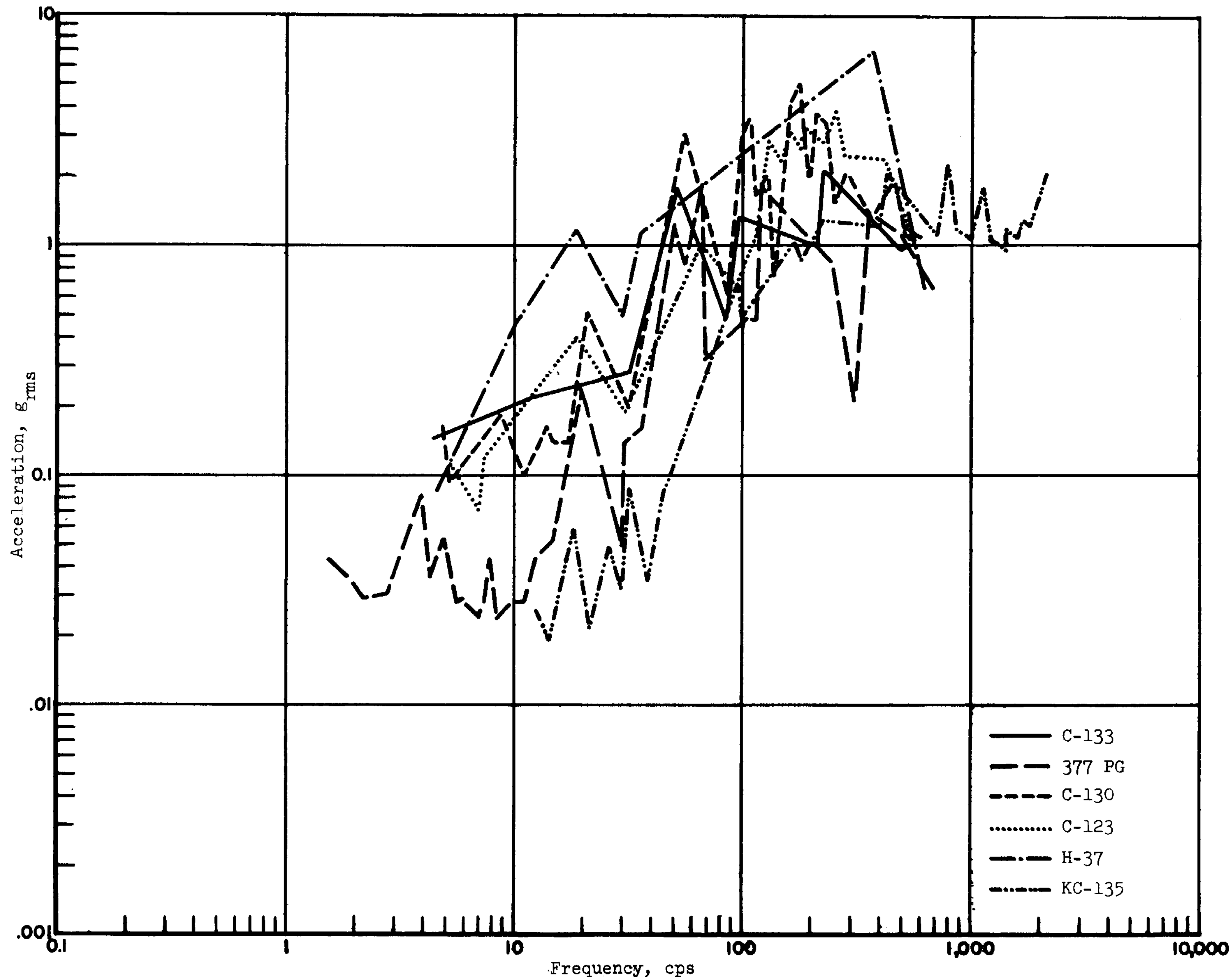


Figure 2. AIRCRAFT ACCELERATION ENVELOPES - DIRECTIONAL COMPOSITES (C-133, 377 PG, C-130, C-123, H-37, KC-135)

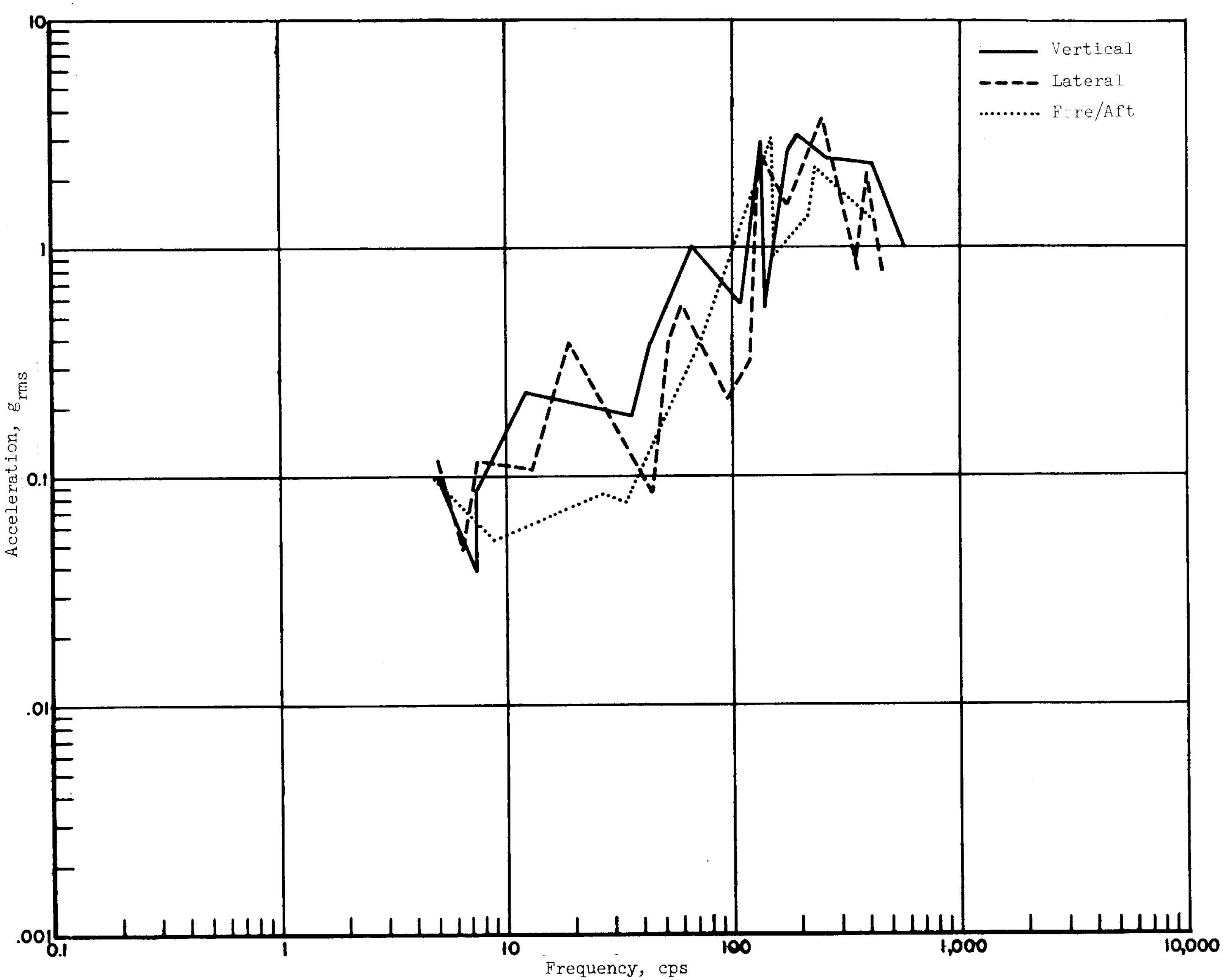


Figure 3. AIRCRAFT ACCELERATION ENVELOPES - C-123 PROPELLER, DIRECTIONAL (Vertical, Lateral, Fore/Aft)

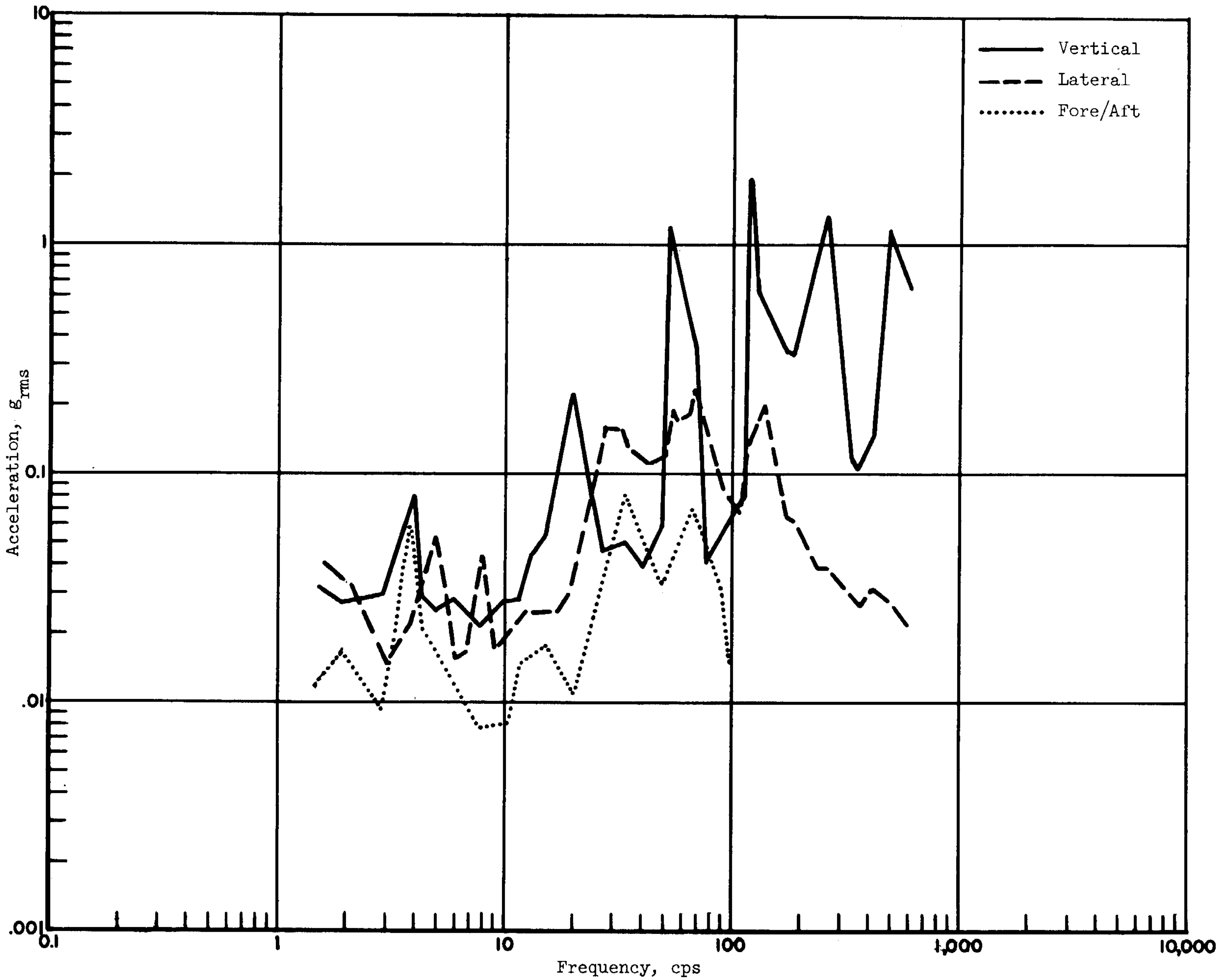


Figure 4. AIRCRAFT ACCELERATION ENVELOPES - 377 PG PROPELLER, DIRECTIONAL ((Vertical, Lateral, Fore/Aft)

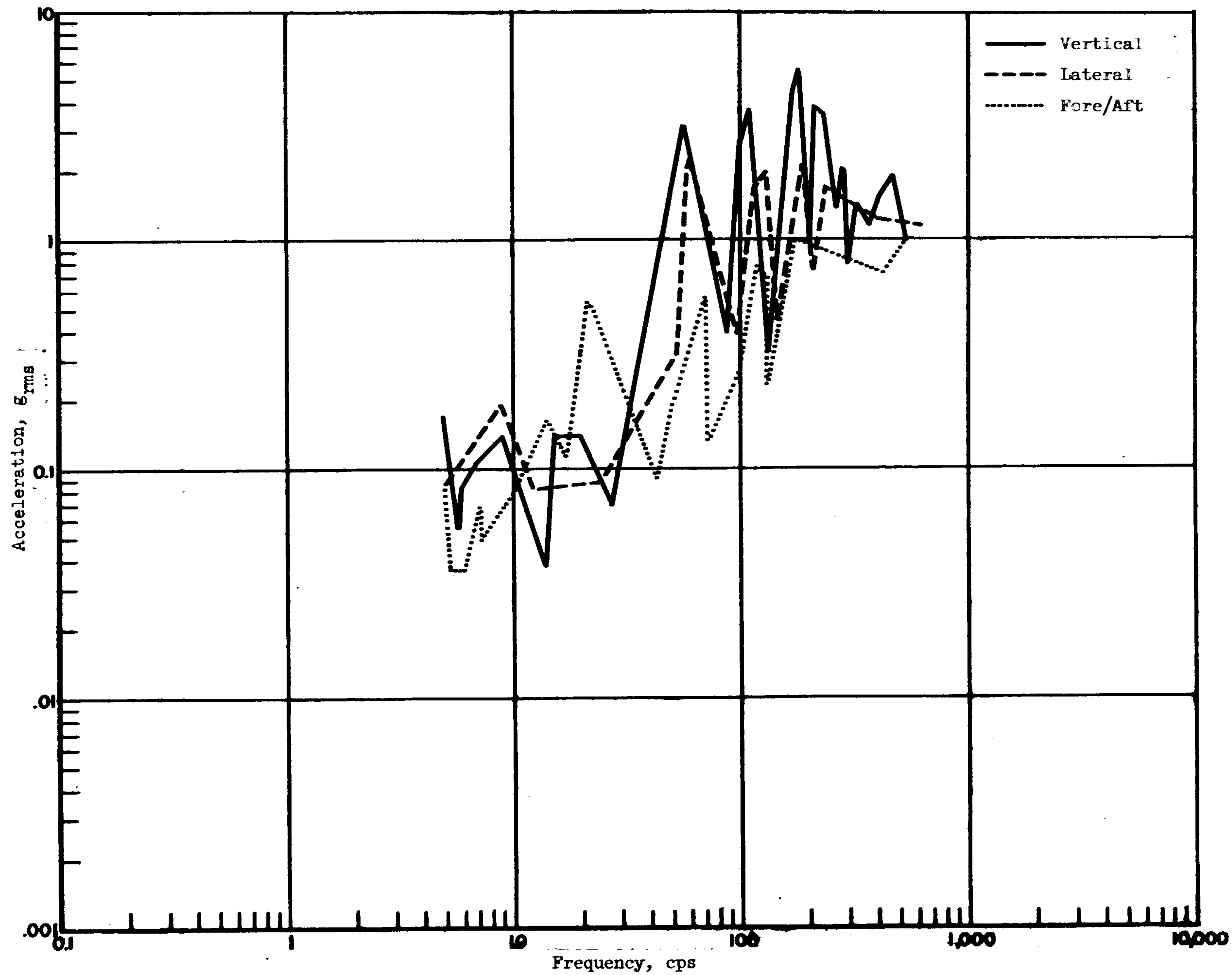


Figure 5. AIRCRAFT ACCELERATION ENVELOPES - C-130 TURBOPROP, DIRECTIONAL (Vertical, Lateral, Fore/Aft)

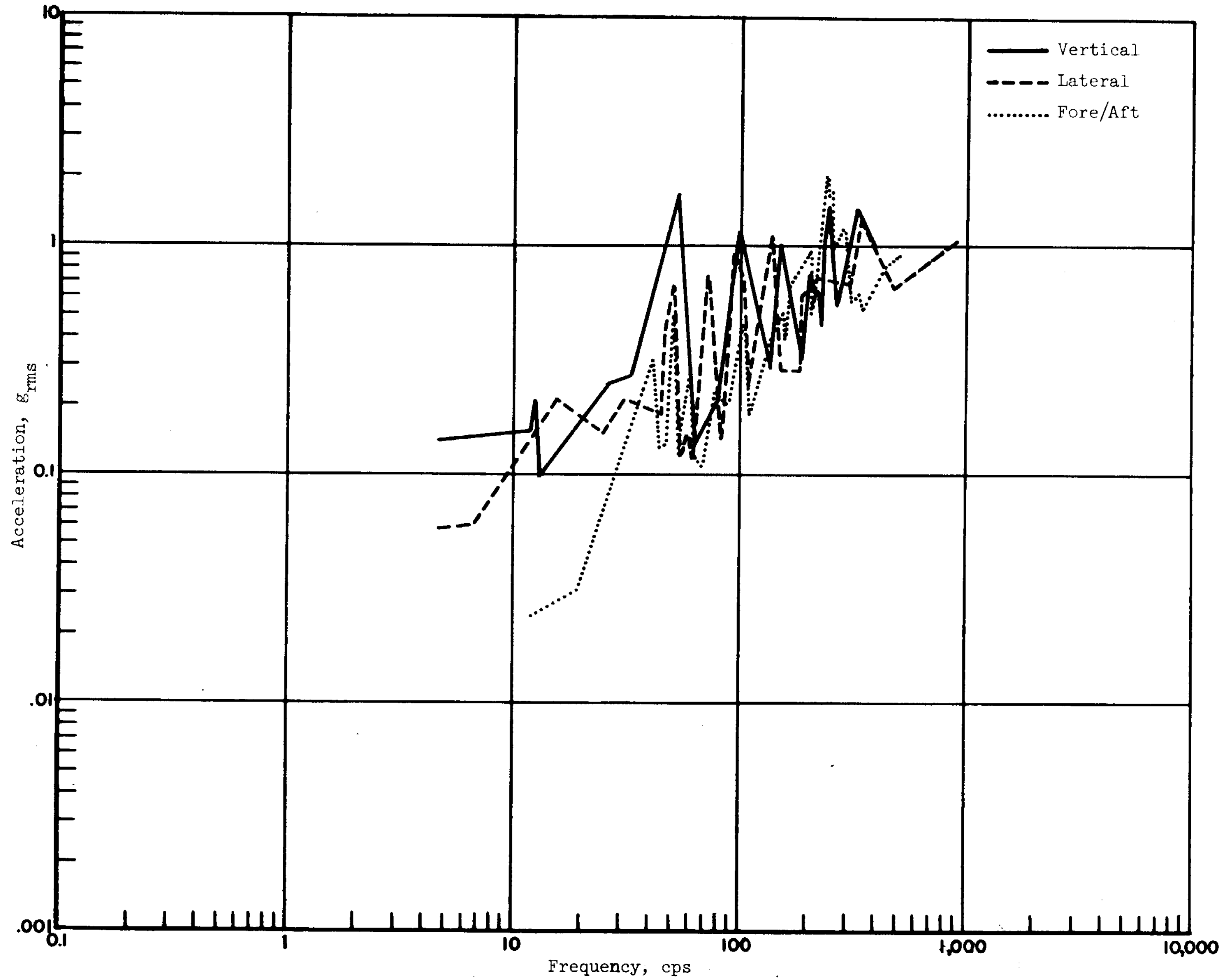


Figure 6. AIRCRAFT ACCELERATION ENVELOPES - C-133 TURBOPROP, DIRECTIONAL (Vertical, Lateral, Fore/Aft)

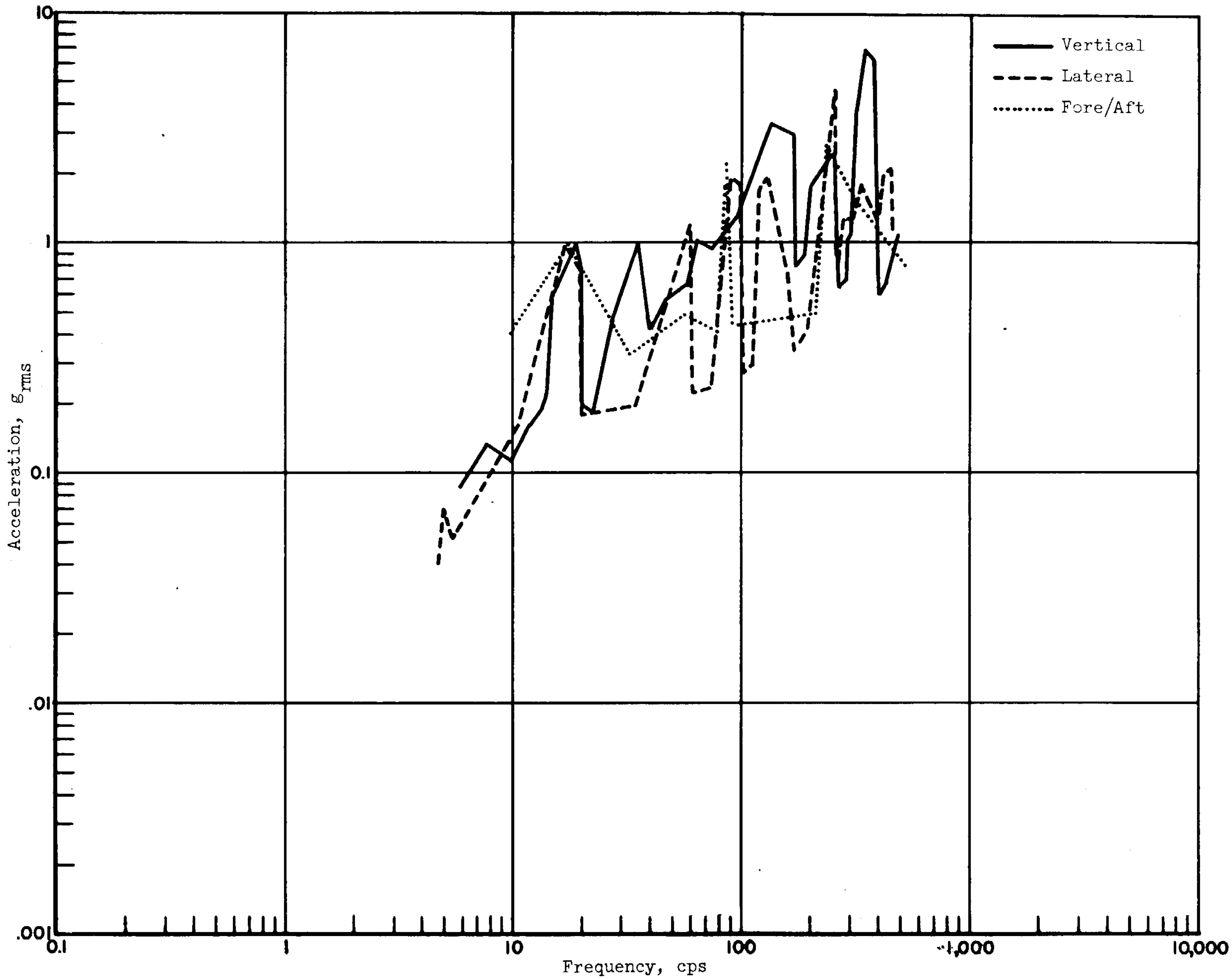


Figure 7. AIRCRAFT ACCELERATION ENVELOPES - H-37 HELICOPTER, DIRECTIONAL (Vertical, Lateral, Fore/Aft)

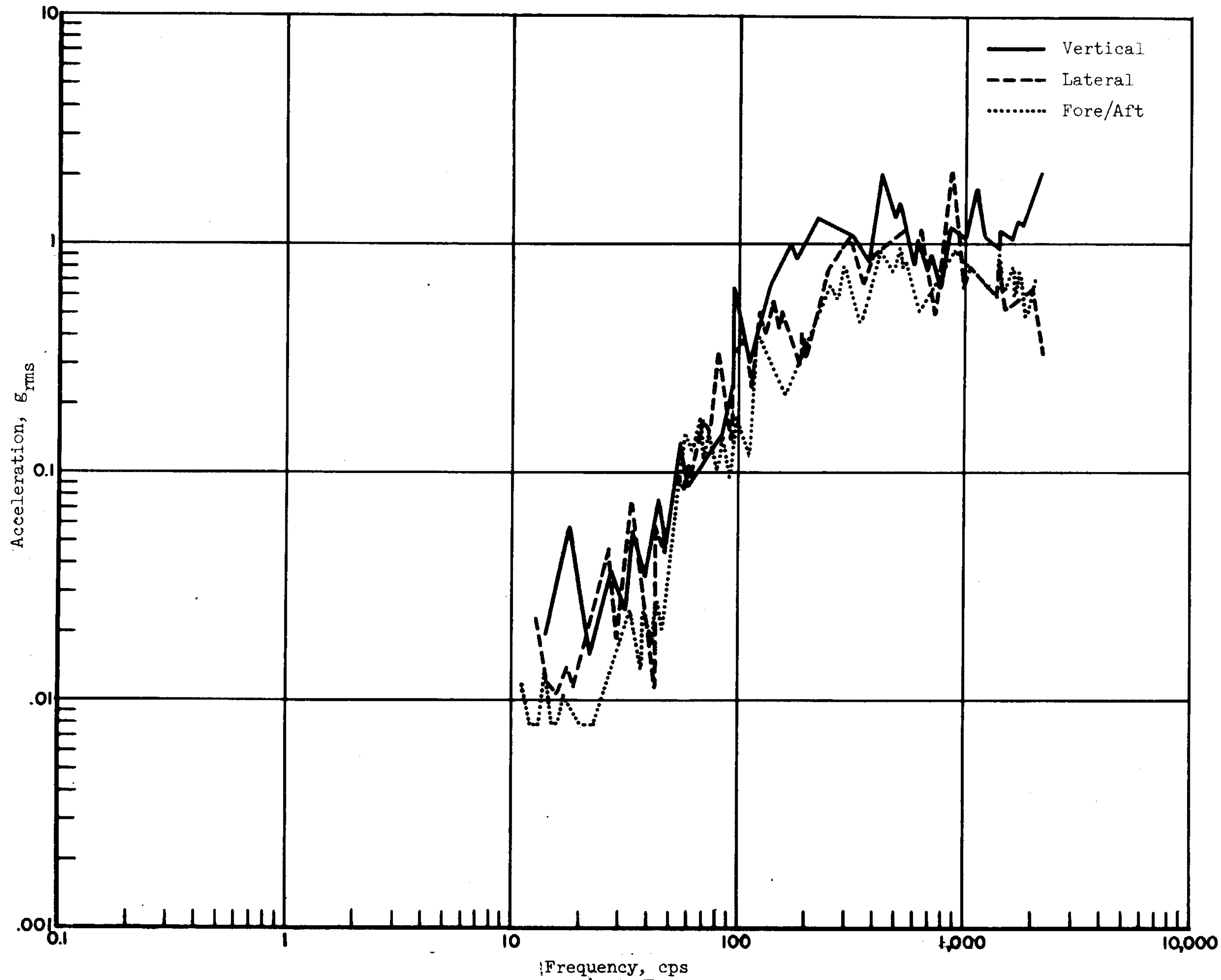


Figure 8. AIRCRAFT ACCELERATION ENVELOPES - KC-135 JET, DIRECTIONAL (Vertical, Lateral, Fore/Aft)

The railroad acceleration diagrams have been formed by enveloping all data for all types of suspension systems, road conditions, directions, and speeds. These diagrams, therefore, reflect the highest levels of vibration which would normally be experienced. The diagrams are presented in terms of zero-to-peak acceleration versus frequency. It should be noted that these diagrams are not continuous curves, but rather represent envelopes of discrete peak acceleration measurements.

The directional composite diagrams (Figure 10) showing the effect of direction have been formed in a similar manner. However, the data has been segregated to show the effect of orientation (direction) during normal running conditions (i.e., data for vibrations during stops and starts has been omitted). These diagrams show that the most severe environment over wide frequency ranges is caused by vertical vibrations followed by that caused from vibration in the lateral direction. The fore-and-aft direction experiences the lowest level of continuous vibration. The vibration levels are presented in terms of zero-to-peak acceleration versus frequency.

The effect of train speed on vibration levels is shown on the velocity composite diagram (Figure 11) for a number of train speeds (20, 40, 73, 80 MPH). The curves depicting the environment at 40, 73, and 80 MPH are based on tests conducted by the U. S. Army Signal Corps. The data includes measurements on cars with various types of suspension systems. The vibrations on the cargo floor were monitored with barium-titanate accelerometers and recorded on magnetic tape. The frequency range of interest in these tests was 20-10,000 cps. The recorded data was analyzed by passing it through various filters. The filter pass-bands used in the analyses are listed as follows:

20-50 cps
50-100
100-200
200-400
300-600
600-1200
1200-2400
2400-4800
> 4800

The figure summarizing the results of the above analyses is presented in terms of average acceleration versus frequency (CPS).

The data presented for the 20 mile per hour speed may be misleading because of the difference in frequency range. This data is a result of a separate program in which frequencies greater than 25 cps were not of interest. This data was analyzed with a frequency analyzer having a continuous band-scan device with a frequency resolution less than one-half cycle. The data from this investigation was presented in terms of average acceleration versus frequency.

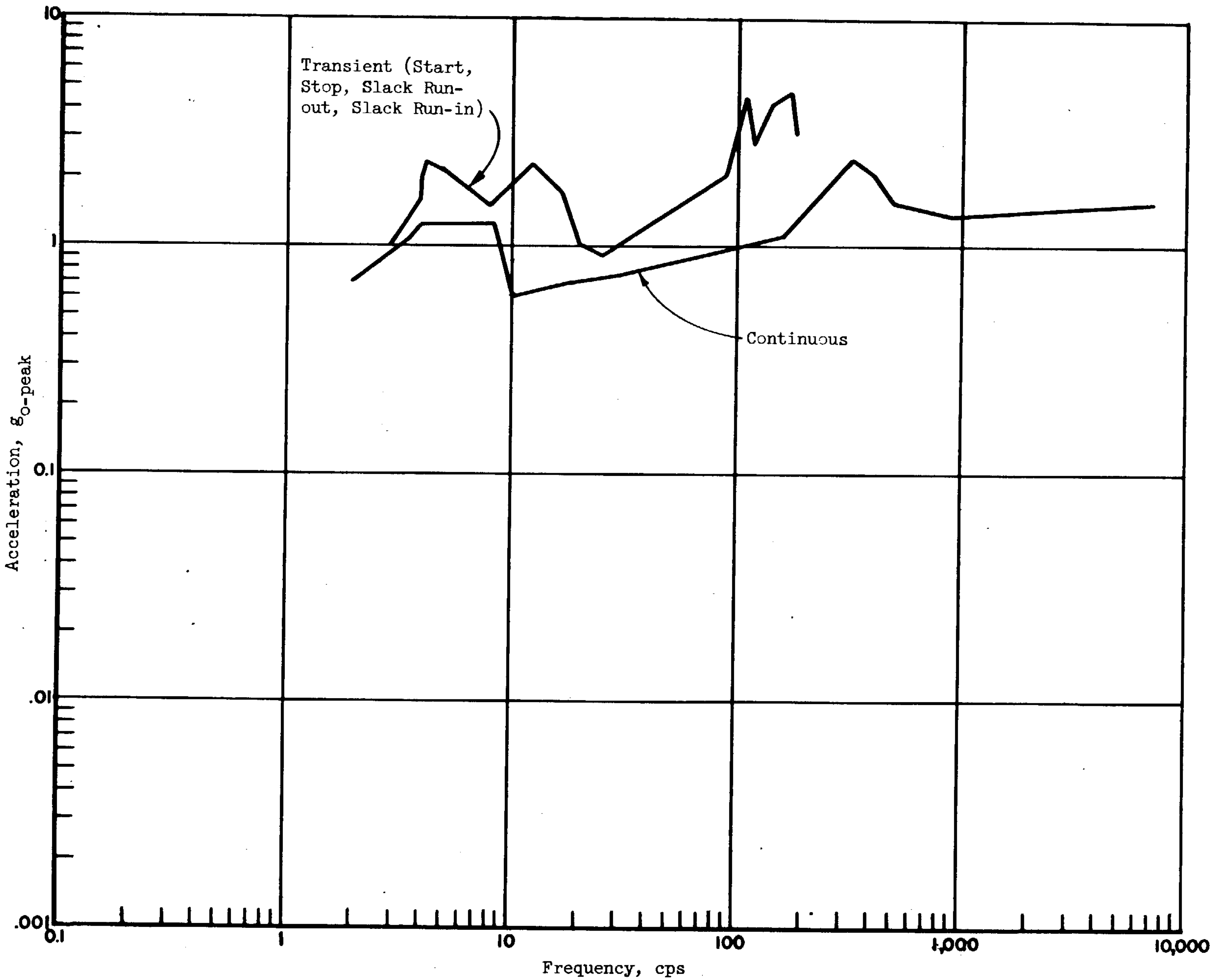


Figure 9. RAILROAD ACCELERATION ENVELOPES - COMPOSITE (Over the road)

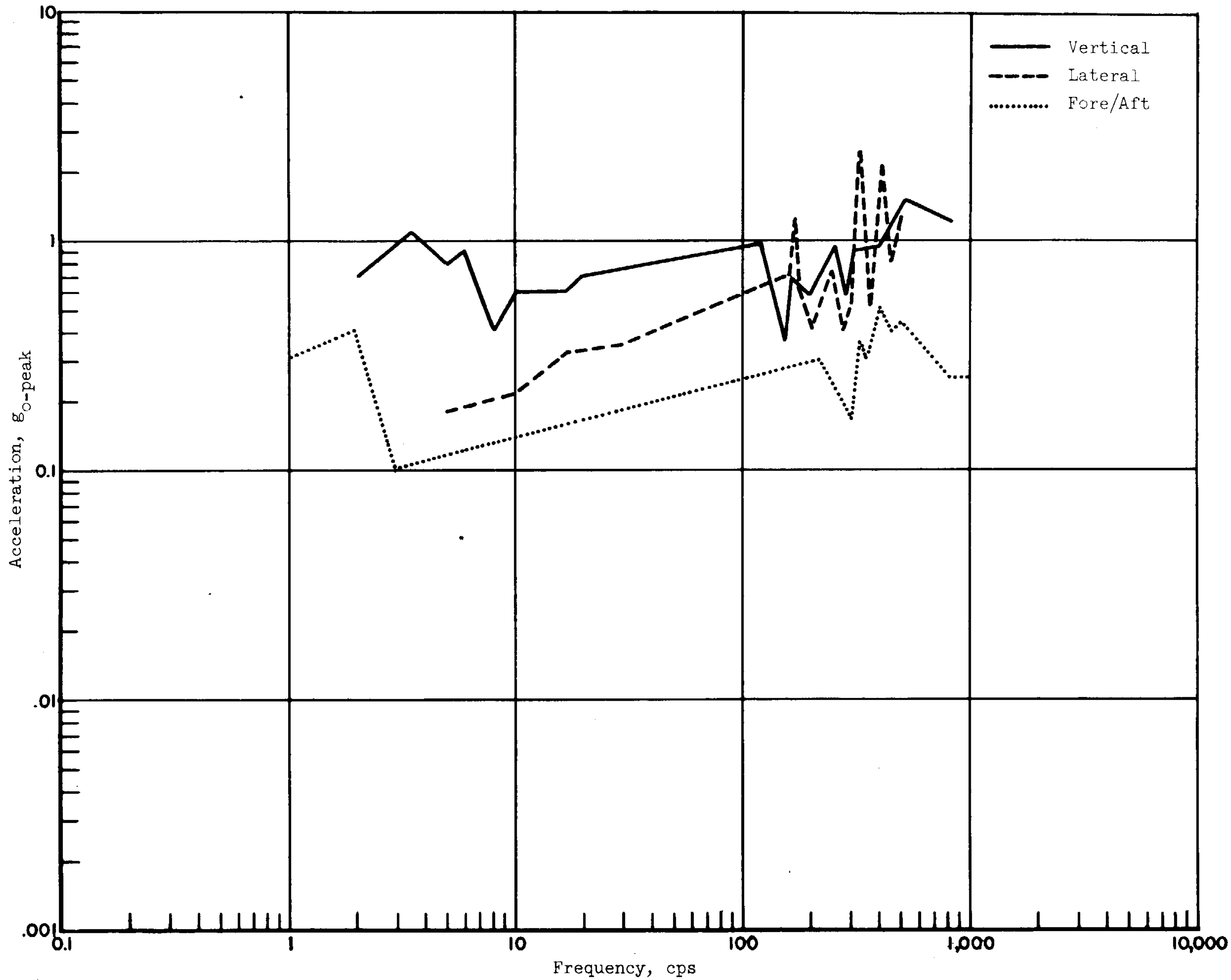


Figure 10 RAILROAD ACCELERATION ENVELOPES - DIRECTIONAL COMPOSITES (Vertical, Lateral, Fore/Aft)

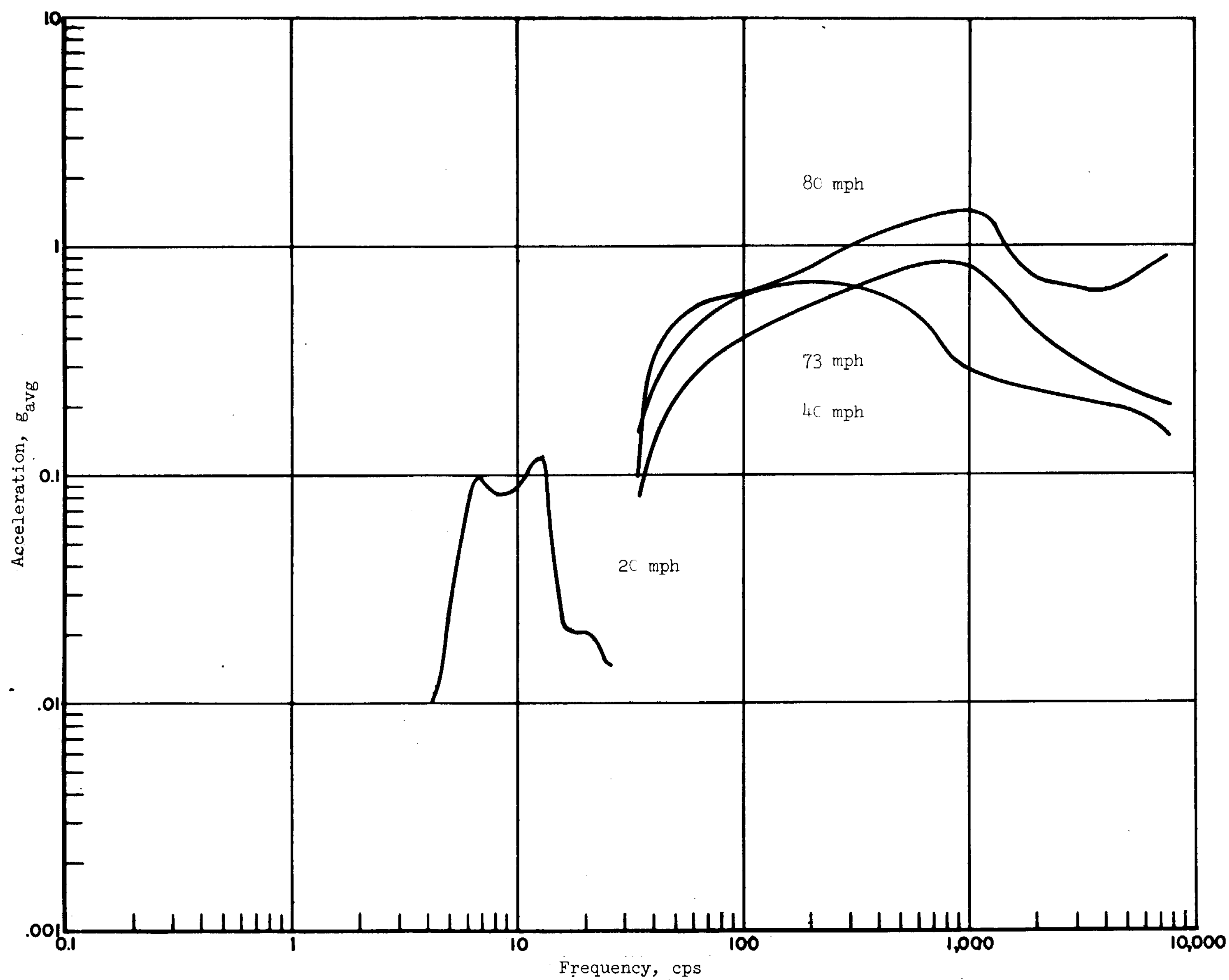


Figure 11. RAILROAD ACCELERATION ENVELOPES - VELOCITY COMPOSITES (Over the road: 20, 40, 73, 80 mph)

The diagrams describing the environment on soft-ride cars (Figure 12) were obtained from tests conducted on the Minuteman transporter railroad car. These diagrams were formed by enveloping the maximum accelerations recorded on the floor of the missile car during cross-country operation. The data was reduced by averaging the 4-6 cycles of peak acceleration. The longitudinal levels for the short Minuteman train were so low (approximately 0.10 g) that humping was considered the only significant longitudinal environment. The truck suspension system for this missile car consisted of a combination air and coil spring system in the vertical direction and a pendulum system with snubbers in the lateral direction. Damping was provided in both directions of motion. In the longitudinal direction, isolation was provided by a sliding center sill and a hydraulic cushioning device. Vibration levels on the diagrams are given in terms of "g" (0-peak) versus frequency.

3.2.2 Railroad - Distribution of Coupling Speeds

The curve showing the distribution of coupling speeds was constructed by averaging the results of a number of independent investigations. This curve (Figure 13) shows the variations in observed impact speeds which occur during coupling and humping operations. Data is presented in terms of percent of total couplings versus coupling speed. The diagram shows that an average of 50 percent of all observed coupling speeds have been less than 5.2 MPH.

3.2.3 Railroad (Coupling)

Curves describing the shock and vibration environment for railroad coupling and humping operations are presented as shock spectra. A shock spectrum is defined as the maximum acceleration of a series of single degree of freedom systems resulting from a specific shock excitation.

Shock spectra are presented for both standard and high-capacity draft gears. Shock spectra corresponding to coupling speeds of 3.4 MPH (Figures 14, 15, 16), 6.0 MPH (Figures 17, 18), 8.0 MPH (Figures 19, 20), and 10.0 MPH (Figures 21, 22, 23) are presented for a standard draft gear.

Spectra for a high-capacity shock-absorbing draft gear are presented for impact velocities of 3.7 MPH (Figure 24), 6.8 MPH (Figure 25), 9.8 MPH (Figure 26), and 12.0 MPH (Figure 27). A Miner draft gear (an end-of-car device combining friction and hydraulic shock absorbers) was used in these tests.

The above spectra were constructed by enveloping the peaks of the shock spectra given in the original report. This procedure has eliminated many of the valleys between the reported peaks. Details concerning the instrumentation locations, car weights, testing methods, or shock spectra for other impact velocities may be found in the original reports.

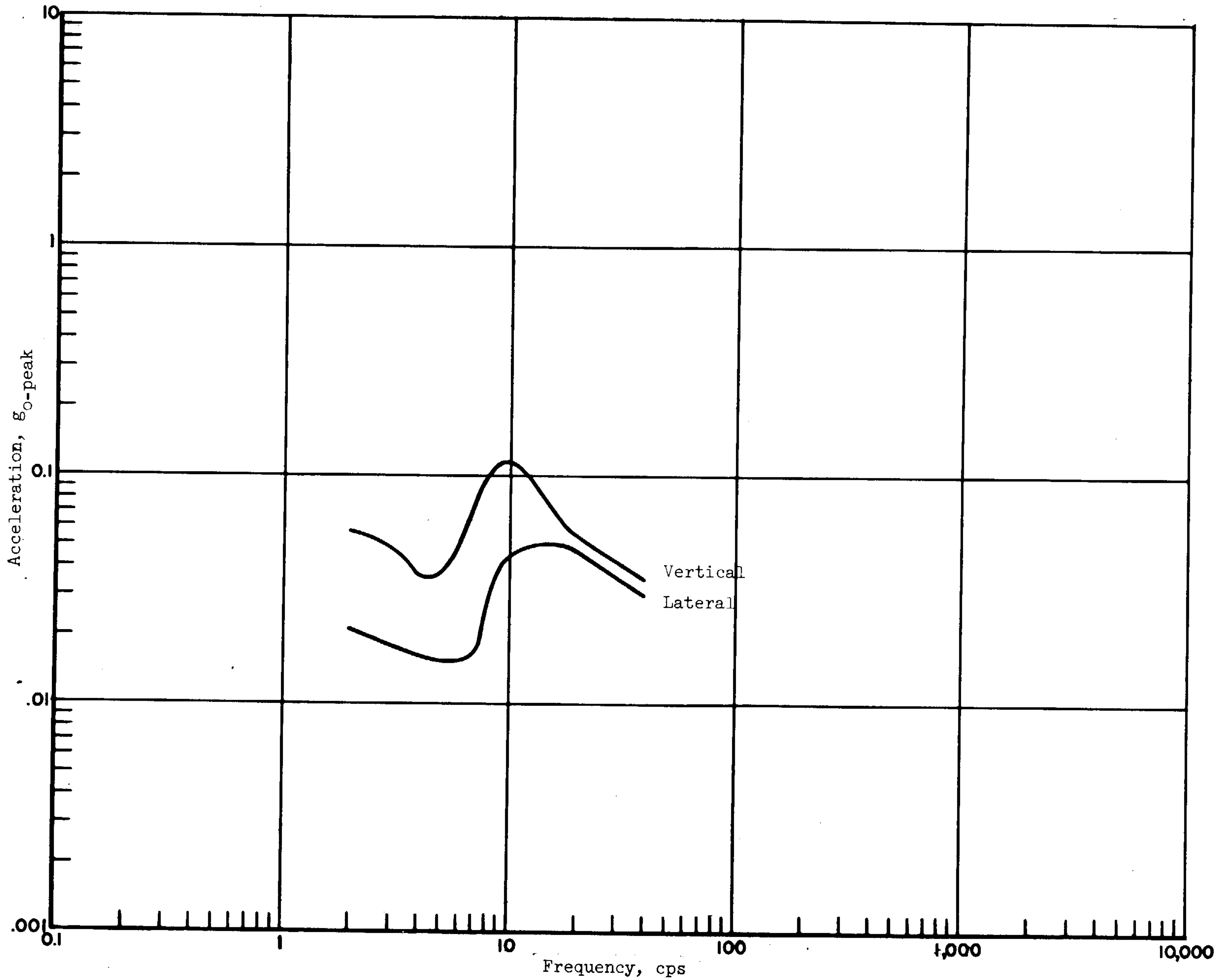


Figure 12. RAILROAD ACCELERATION ENVELOPES - SOFT RIDE CAR, DIRECTIONAL (Vertical and lateral)

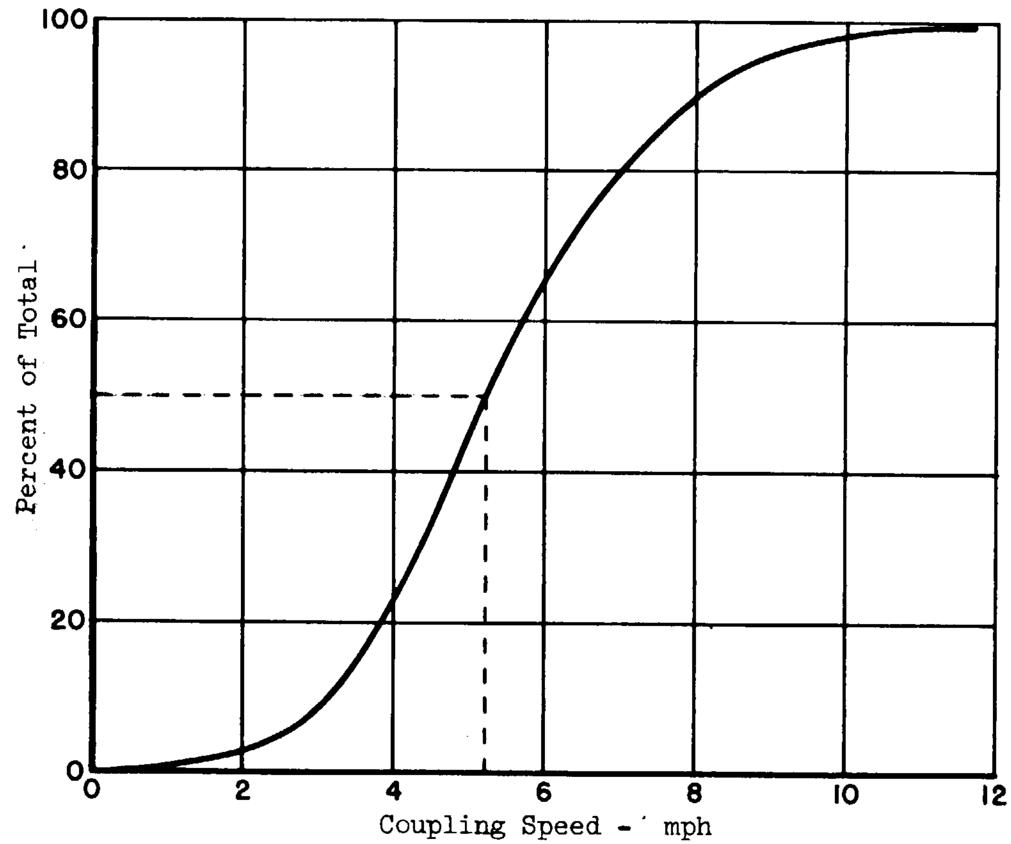


Figure 13. RAILROAD - DISTRIBUTION OF COUPLING SPEEDS

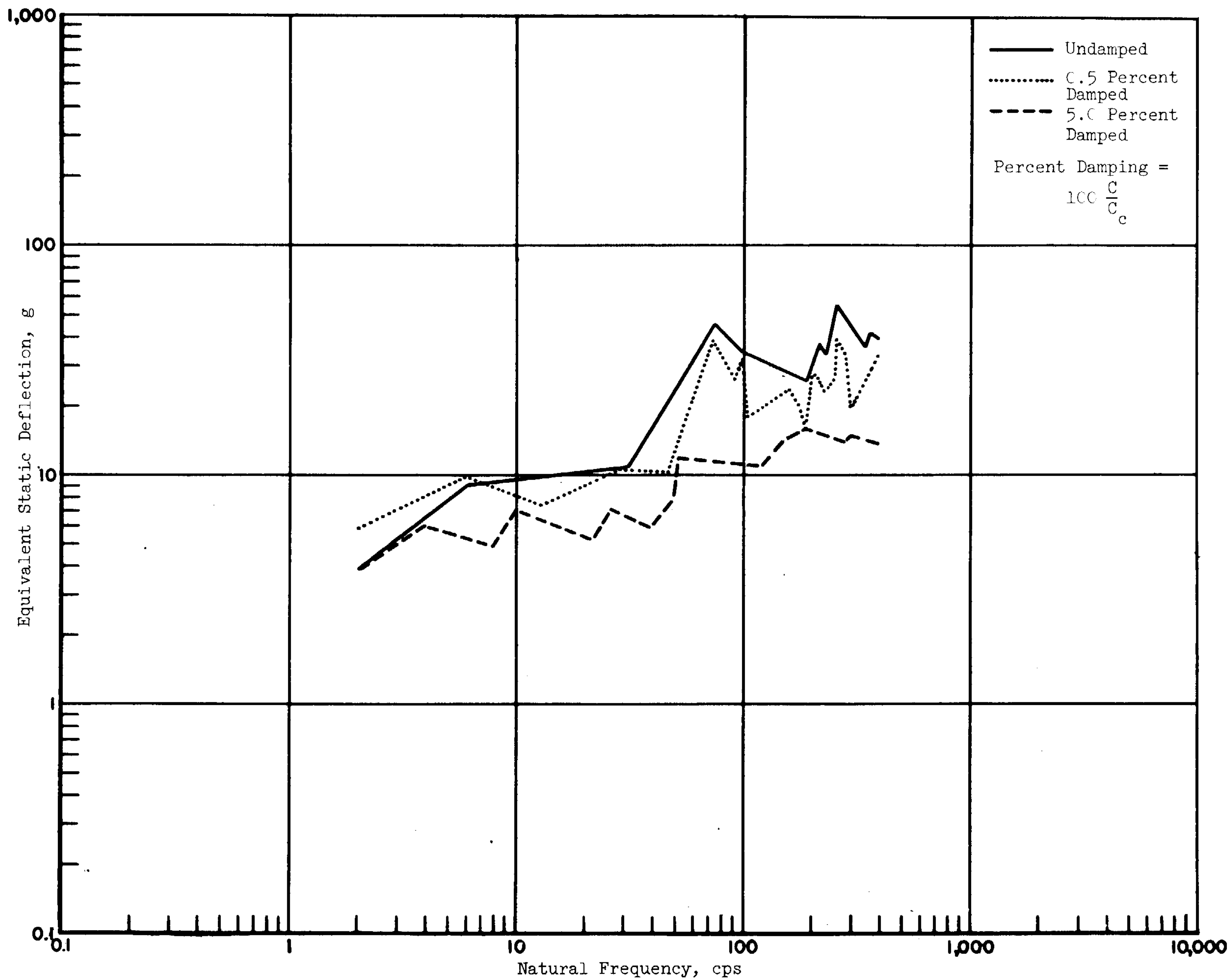


Figure 14. RAILROAD COUPLING SHOCK SPECTRUM - STANDARD DRAFT GEAR (3.4 mph, Fore/Aft)

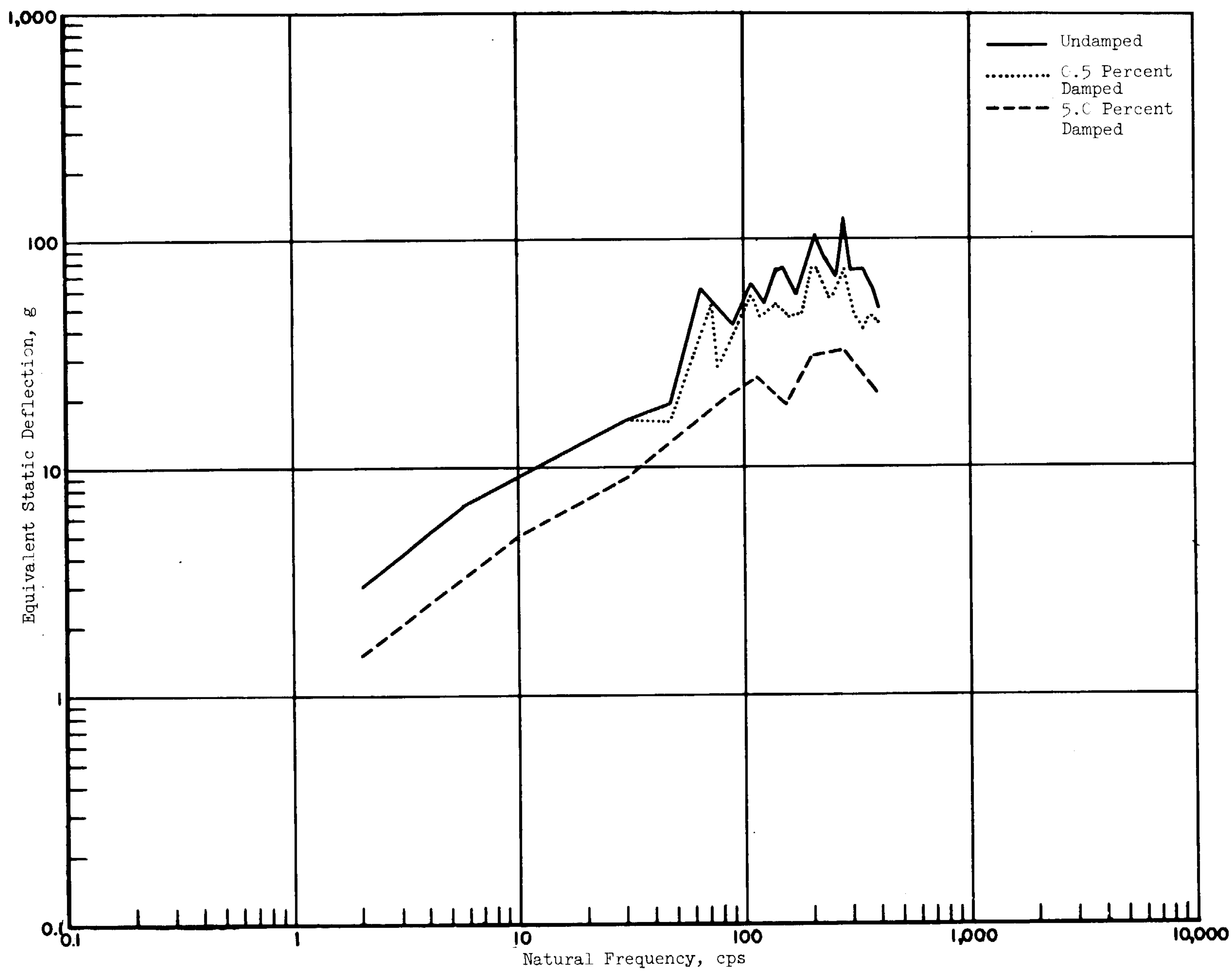


Figure 15. RAILROAD COUPLING SHOCK SPECTRUM - STANDARD DRAFT GEAR (3.4 mph, Vertical)

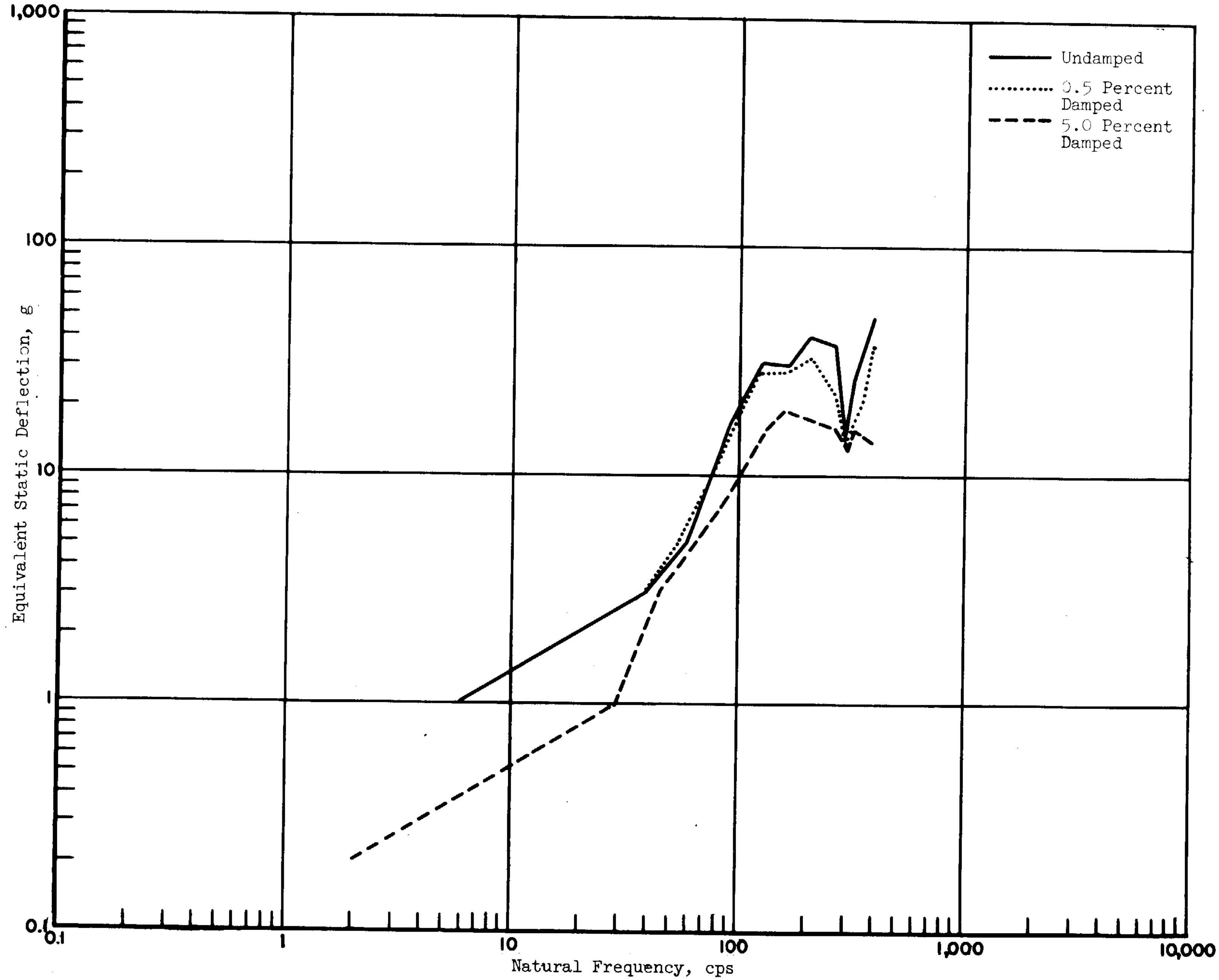


Figure 16. RAILROAD COUPLING SHOCK SPECTRUM - STANDARD DRAFT GEAR (3.4 mph, Lateral)

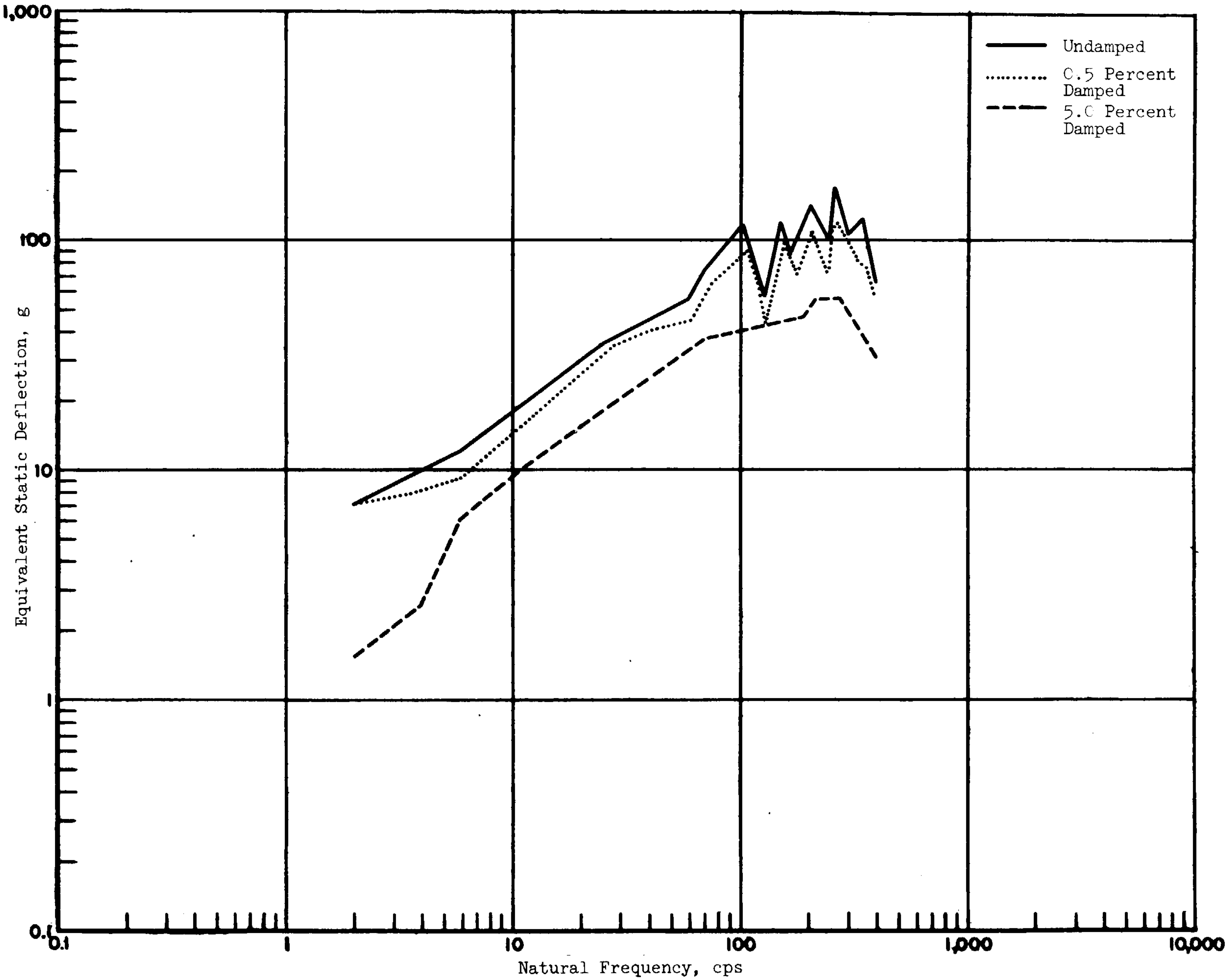


Figure 17. RAILROAD COUPLING SHOCK SPECTRUM - STANDARD DRAFT GEAR (6.0 mph, Fore/Aft)

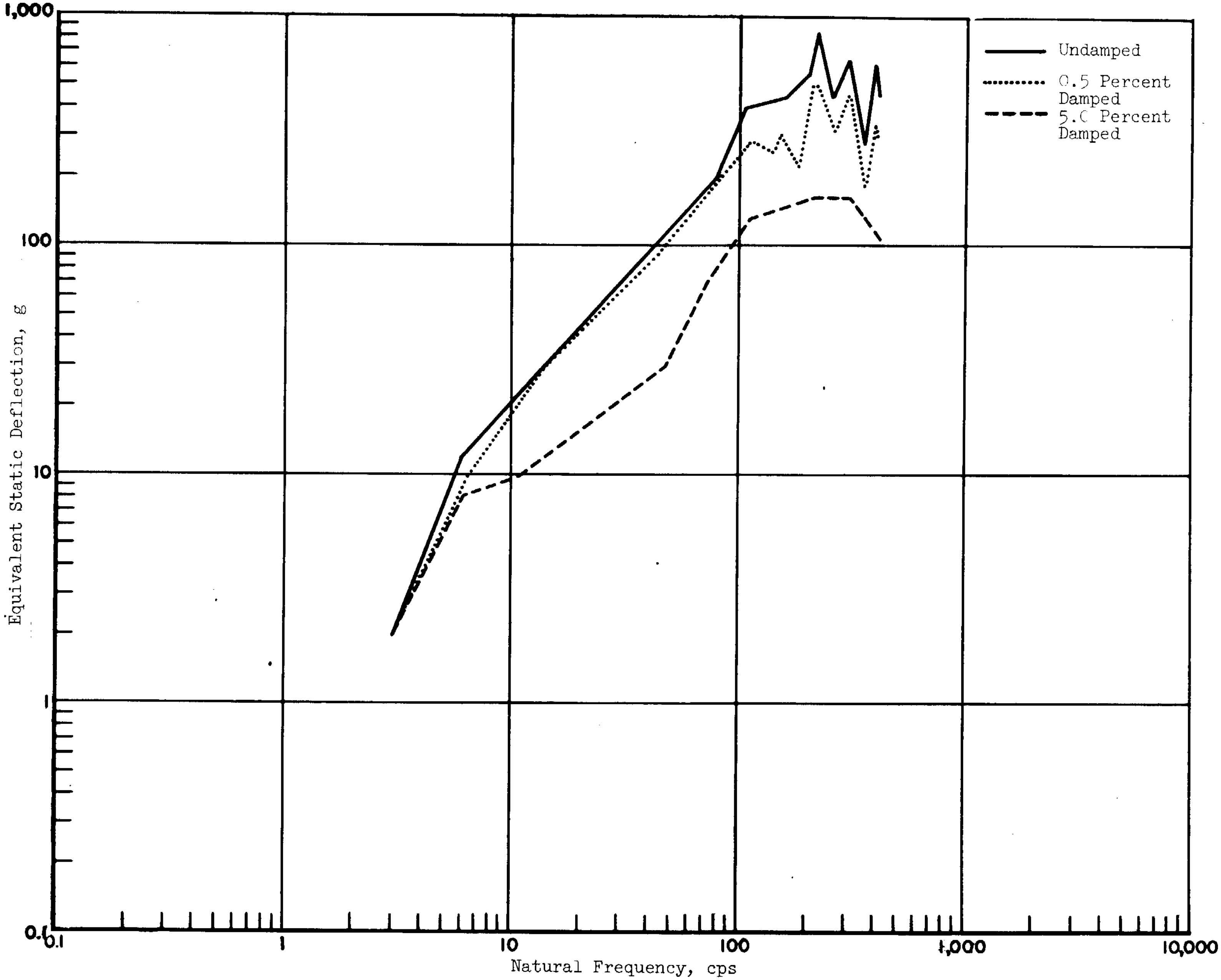


Figure 18. RAILROAD COUPLING SHOCK SPECTRUM - STANDARD DRAFT GEAR (6.0 mph, Vertical)

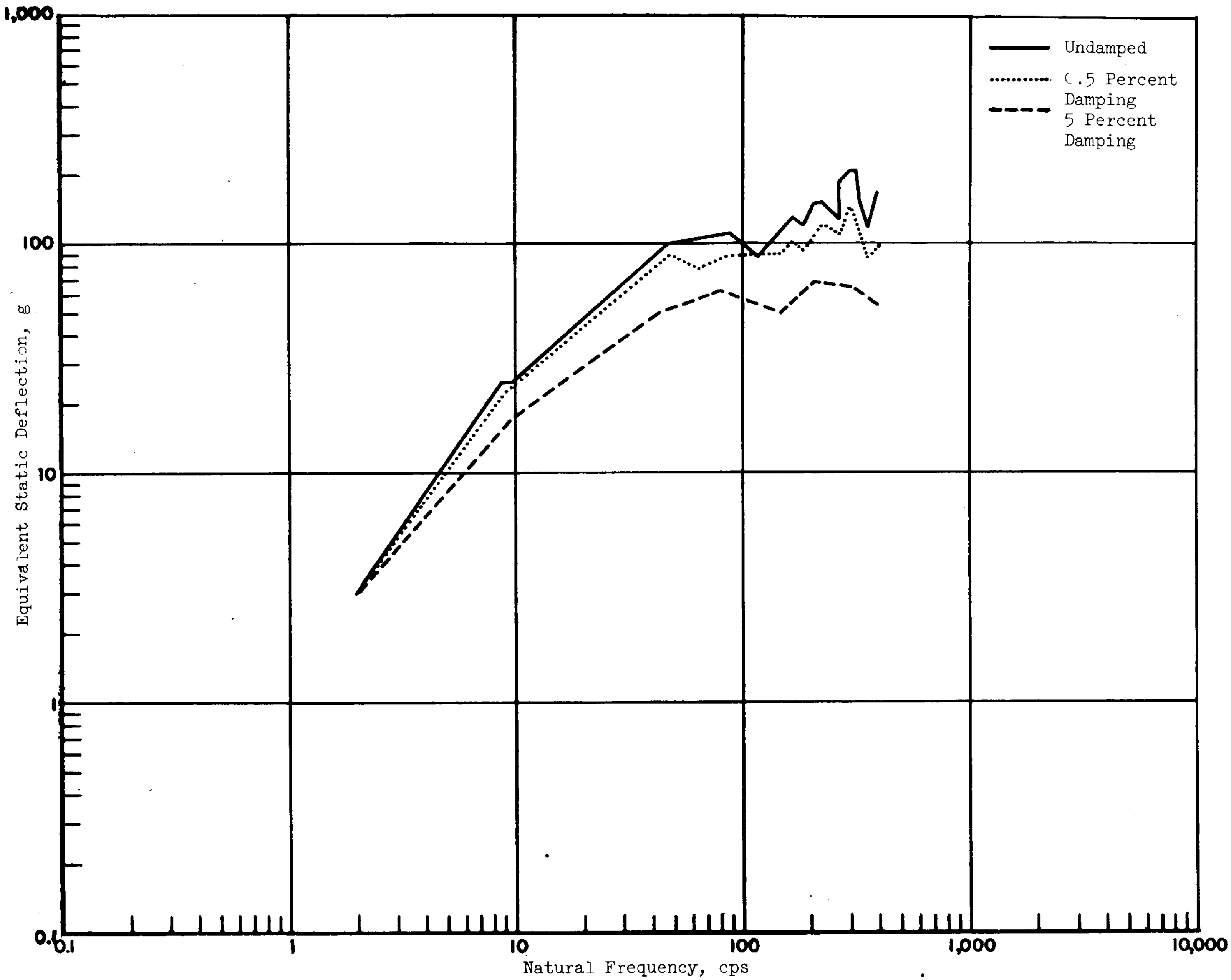


Figure 19. RAILROAD COUPLING SHOCK SPECTRUM - STANDARD DRAFT GEAR (8.0 mph, Fore/Aft)

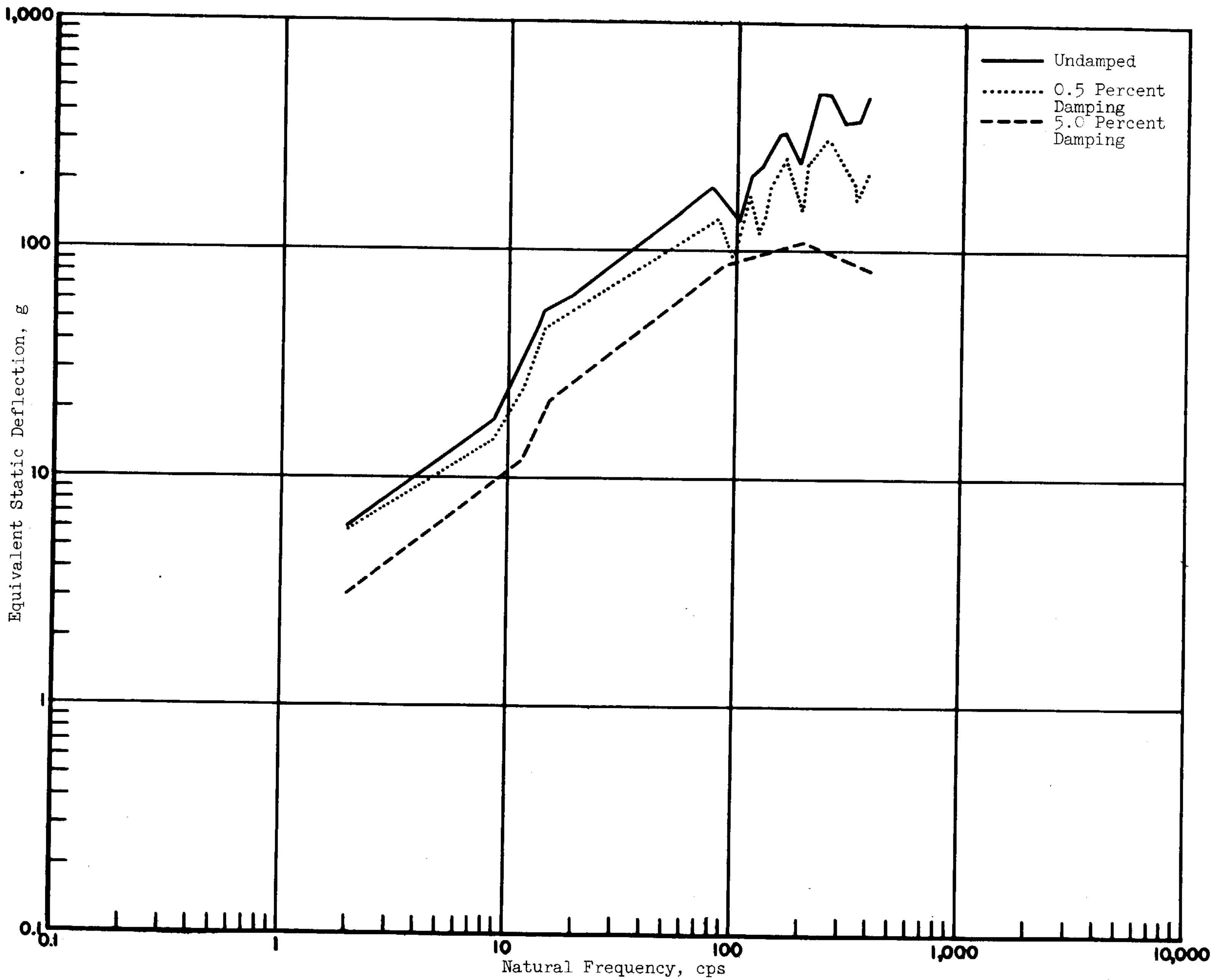


Figure 20. RAILROAD COUPLING SHOCK SPECTRUM - STANDARD DRAFT GEAR (8.0 mph, Vertical)

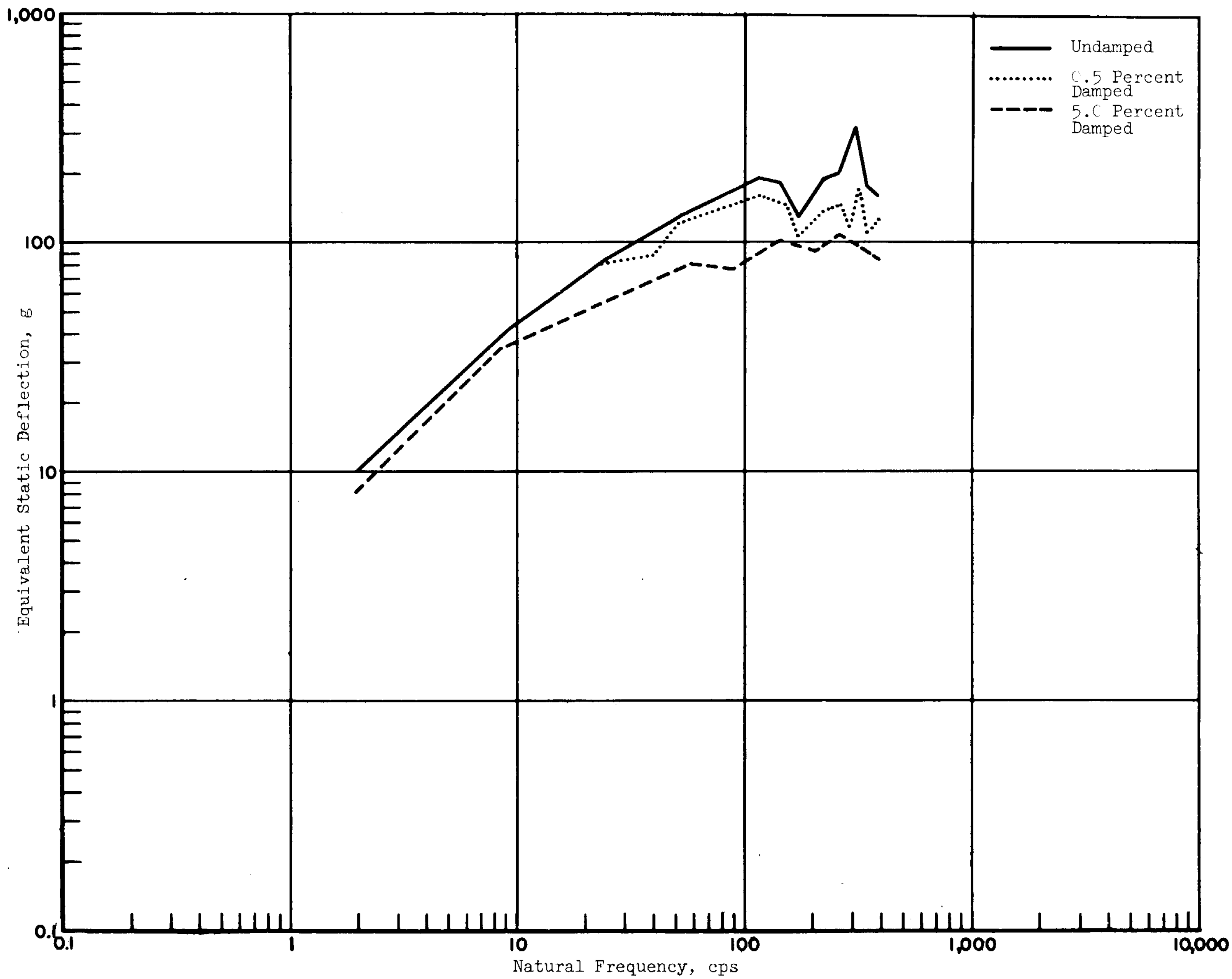


Figure 21. RAILROAD COUPLING SHOCK SPECTRUM - STANDARD DRAFT GEAR (10.0 mph, Fore/Aft)

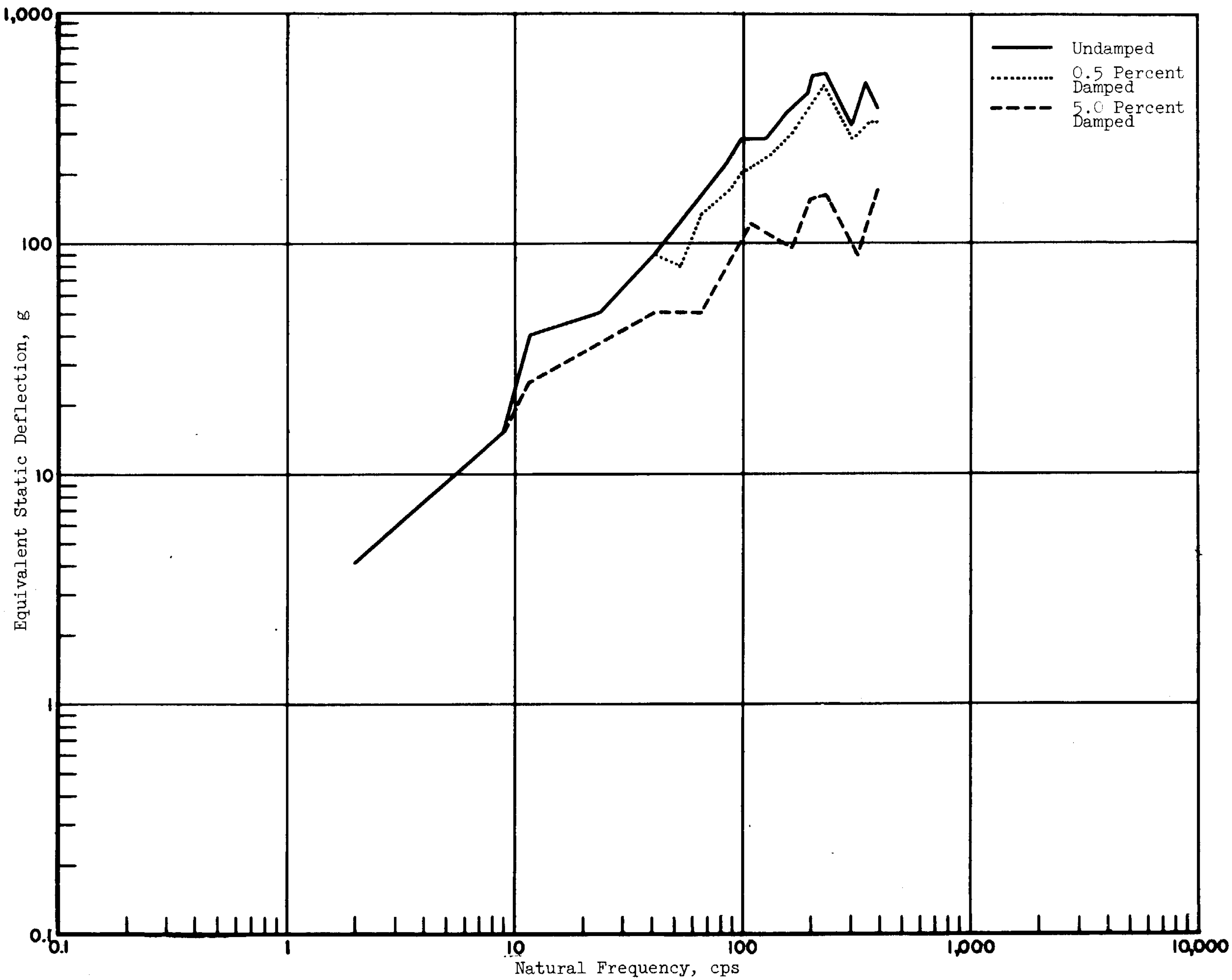


Figure 22. RAILROAD COUPLING SHOCK SPECTRUM - STANDARD DRAFT GEAR (10.0 mph, Vertical)

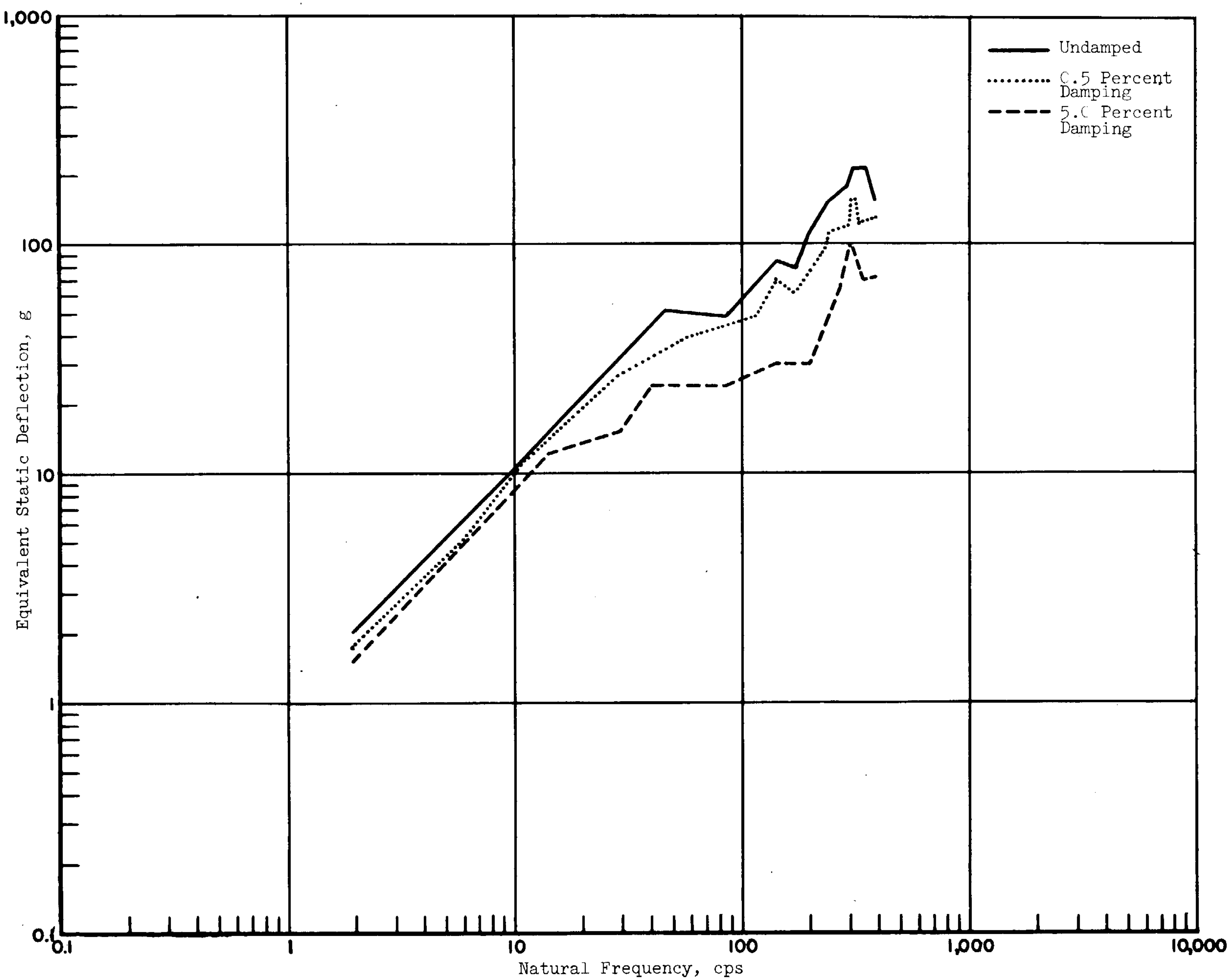


Figure 23. RAILROAD COUPLING SHOCK SPECTRUM - STANDARD DRAFT GEAR (10.0 mph, Lateral)

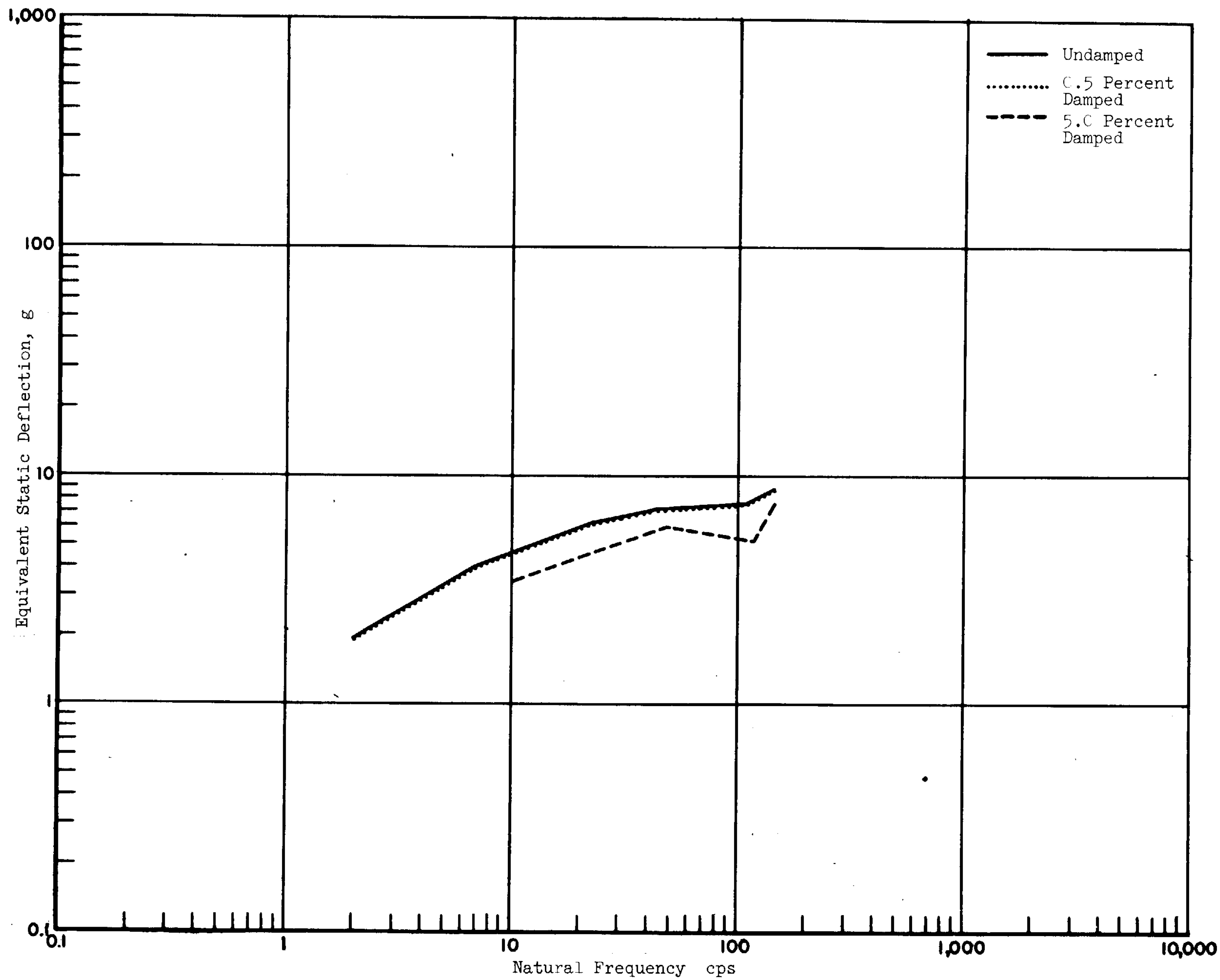


Figure 24. RAILROAD COUPLING SHOCK SPECTRUM - HIGH CAPACITY DRAFT GEAR (3.7 mph, Fore/Aft)

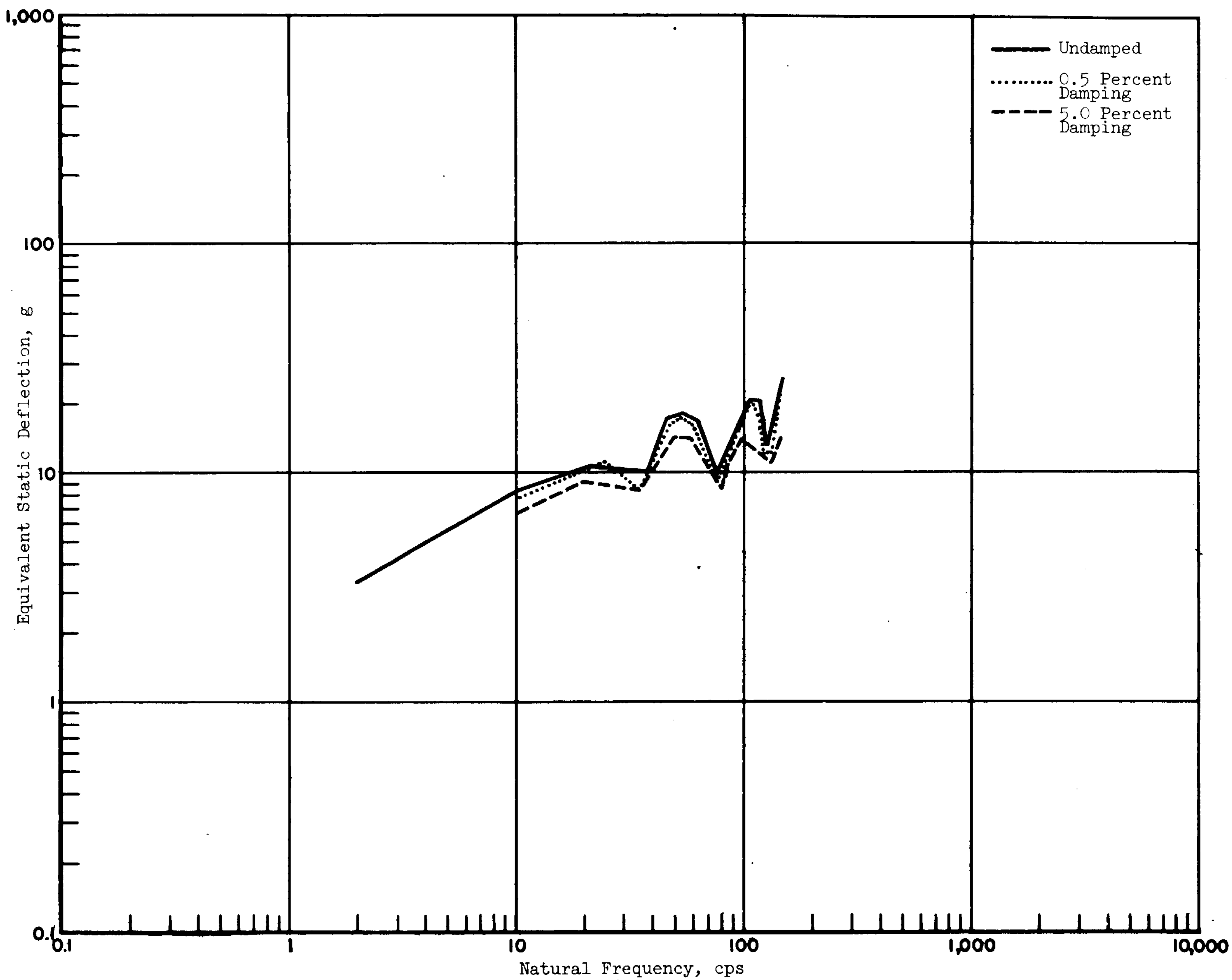


Figure 25. RAILROAD COUPLING SHOCK SPECTRUM - HIGH CAPACITY DRAFT GEAR (6.8 mph, Fore/Aft)

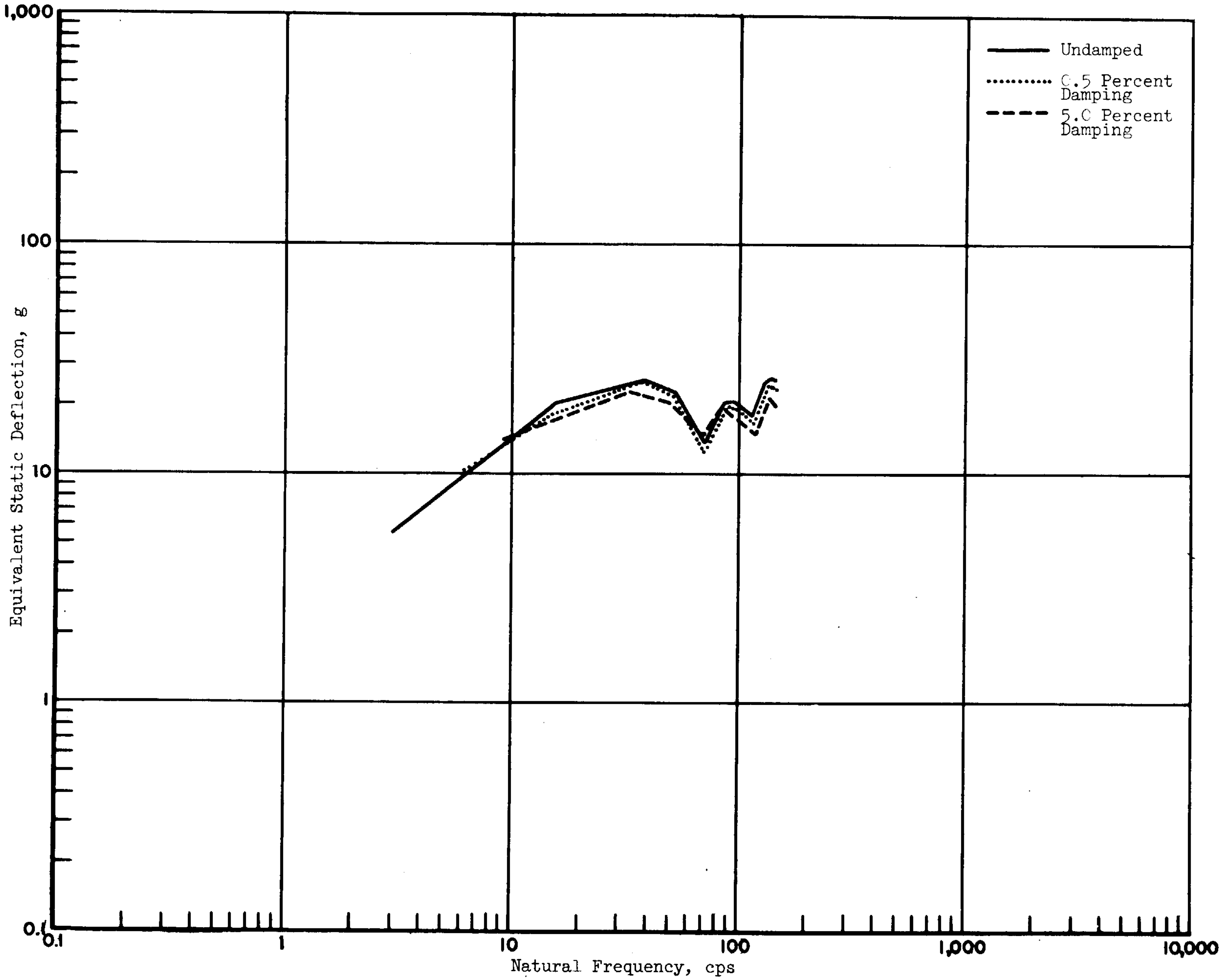


Figure 26. RAILROAD COUPLING SHOCK SPECTRUM - HIGH CAPACITY DRAFT GEAR (9.8 mph, Fore/Aft)

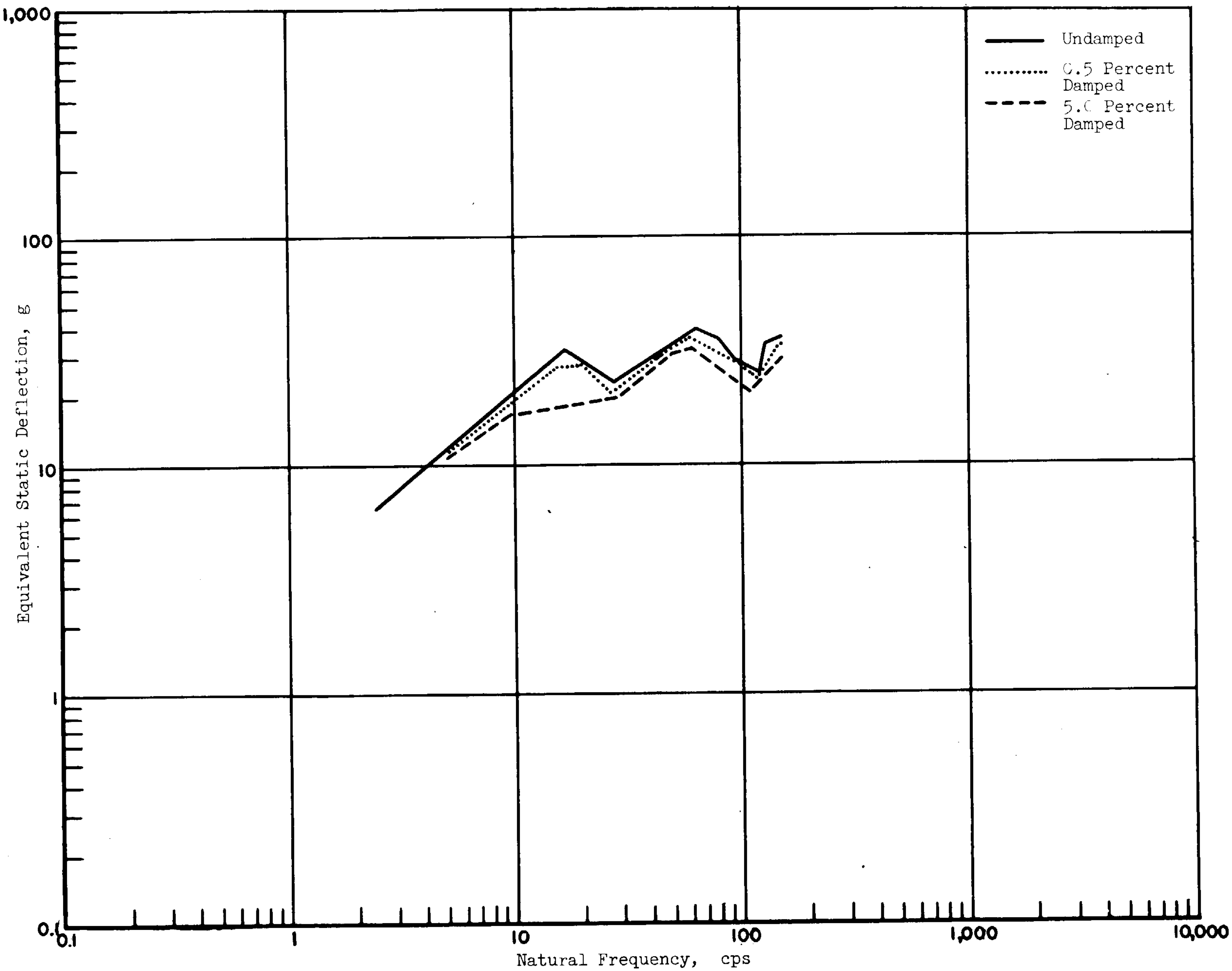


Figure 27. RAILROAD COUPLING SHOCK SPECTRUM - HIGH CAPACITY DRAFT GEAR (12.0 mph, Fore/Aft)

3.3 Trucks

The composite diagram describing the truck shock and vibration environment (Figure 28) has been obtained by enveloping data from a number of individual test programs. This diagram includes data representative of the environment experienced in traversing rough roads, ditches, potholes, railroad crossings and bridges. Data reduction procedures employed in determining the vibration amplitudes vary from one report to another, but in most cases the method used was to record the data with oscillographs and visually determine the peak (zero-to-peak) acceleration and dominant frequency. This method has been used extensively in transportation studies since it requires little auxiliary equipment and since the magnitude of the significant dominant frequencies can be conveniently and immediately determined. Continuous spectral analyses of data applicable to commercial trucks traveling normal routes could not be found.

The diagrams showing the effects of cargo load (Figure 29) are the result of a single measurement program. In this program, tests were conducted with three standard commercial semi-trailers, each having one of three basic suspension systems (air-ride tandem suspension, stable-ride single suspension, and single-axle spring suspension). The tests were run at two load conditions, empty and full, over a first-class asphalt road. Vertical accelerations were monitored at three locations on the cargo floor (over the fifth wheel, the center of the van floor, and over the rear axle). Frequencies were determined by visually examining the oscillograph records. Examination of Figure 29 shows that the vibration levels are almost unaffected by load in the lower frequency ranges. Higher frequency components, however, are reduced on loaded trucks.

Data recorded on a 2-1/2 ton military truck (Figure 30) has been included to show the effect of orientation because no data of this type is available for a commercial vehicle. Although it is recognized that the vibration levels for this type of vehicle will be higher than for a commercial vehicle, because of the more rigid suspension system, it is believed that the relative severity of the vibration levels as a function of direction are demonstrated by these results. The curves show that the vibration levels are highest in the vertical direction over wide ranges of frequency. The lateral and fore-and-aft directions experience almost equal vibration levels.

The diagram showing the effect of road condition on the vibration environment (Figure 31) was formed by combining the plot of the composite truck acceleration envelope, (which included rough road) with paved road environmental data. This diagram shows the difference in levels between the vibrations which occur while traversing rough roads, potholes, ditches, railroad crossings, etc., and the maximum vibration environment occurring during operation on paved roads.

3.4 Ships

The data used to construct the composite diagram for ships was obtained from many sources. For purposes of presentation, the data was separated into continuous and transient vibrations. Transient vibrations are defined as those which occur

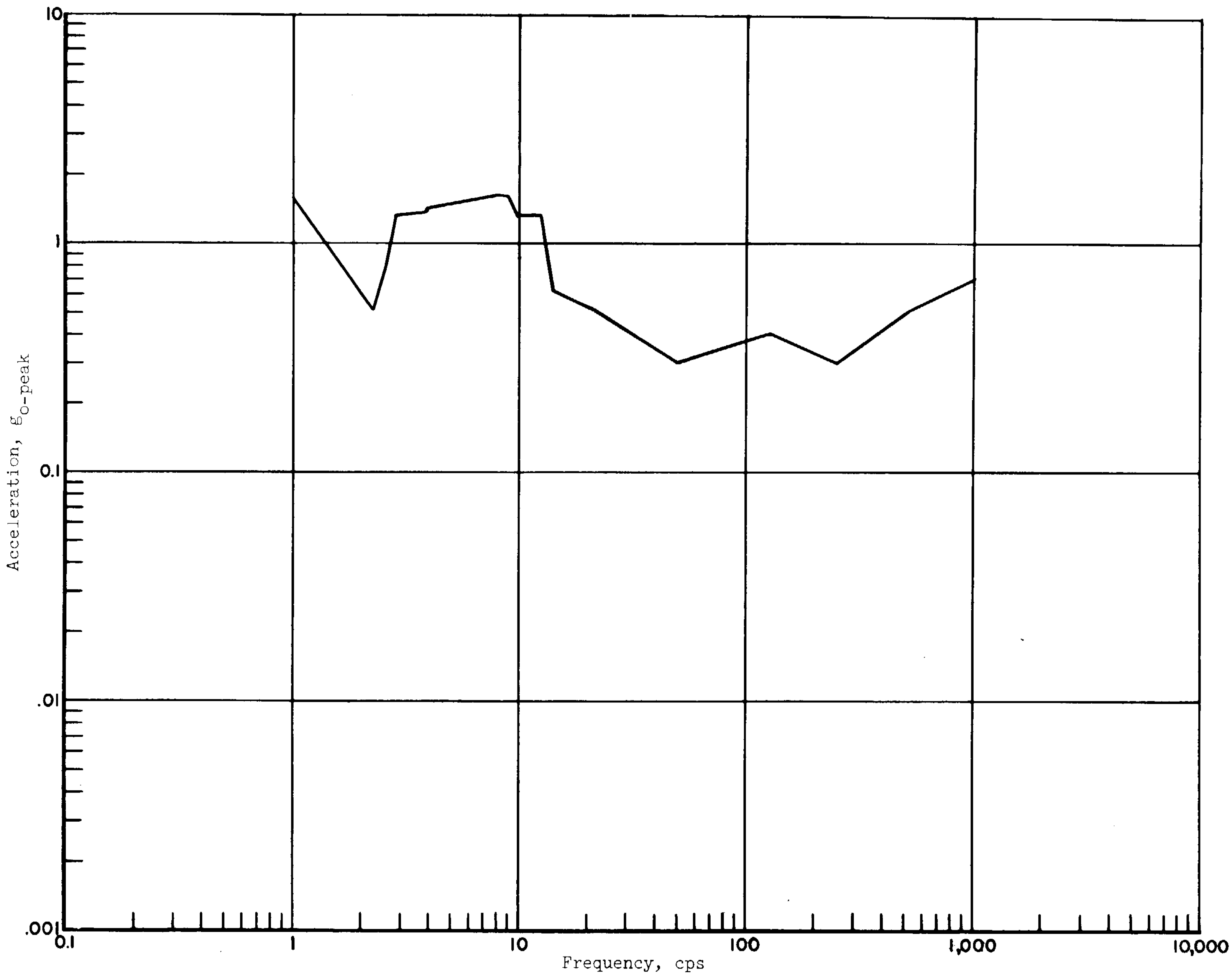


Figure 28. TRUCK ACCELERATION ENVELOPE - COMPOSITE

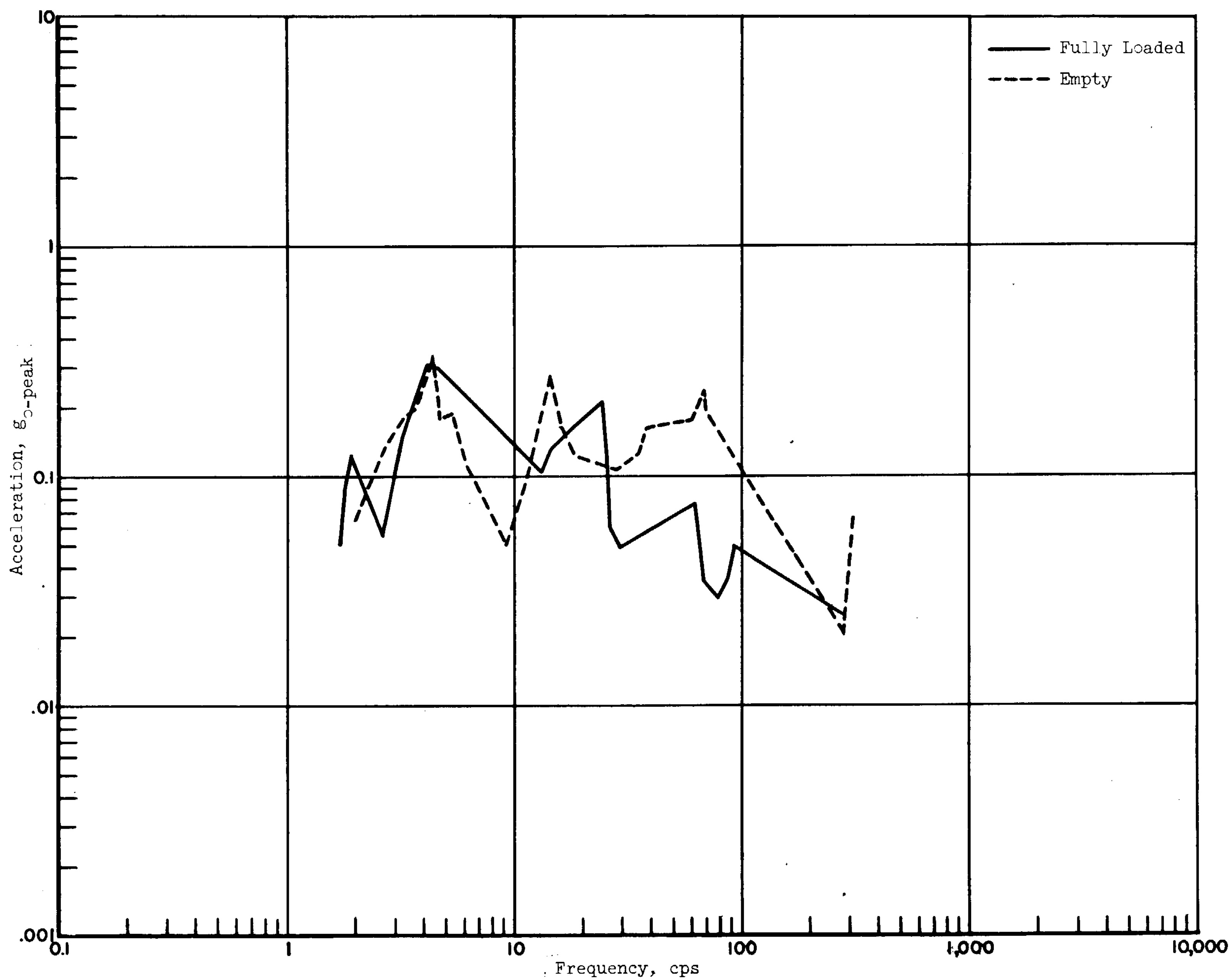


Figure 29. TRUCK ACCELERATION ENVELOPES - CARGO LOAD COMPOSITES (Empty and fully loaded)

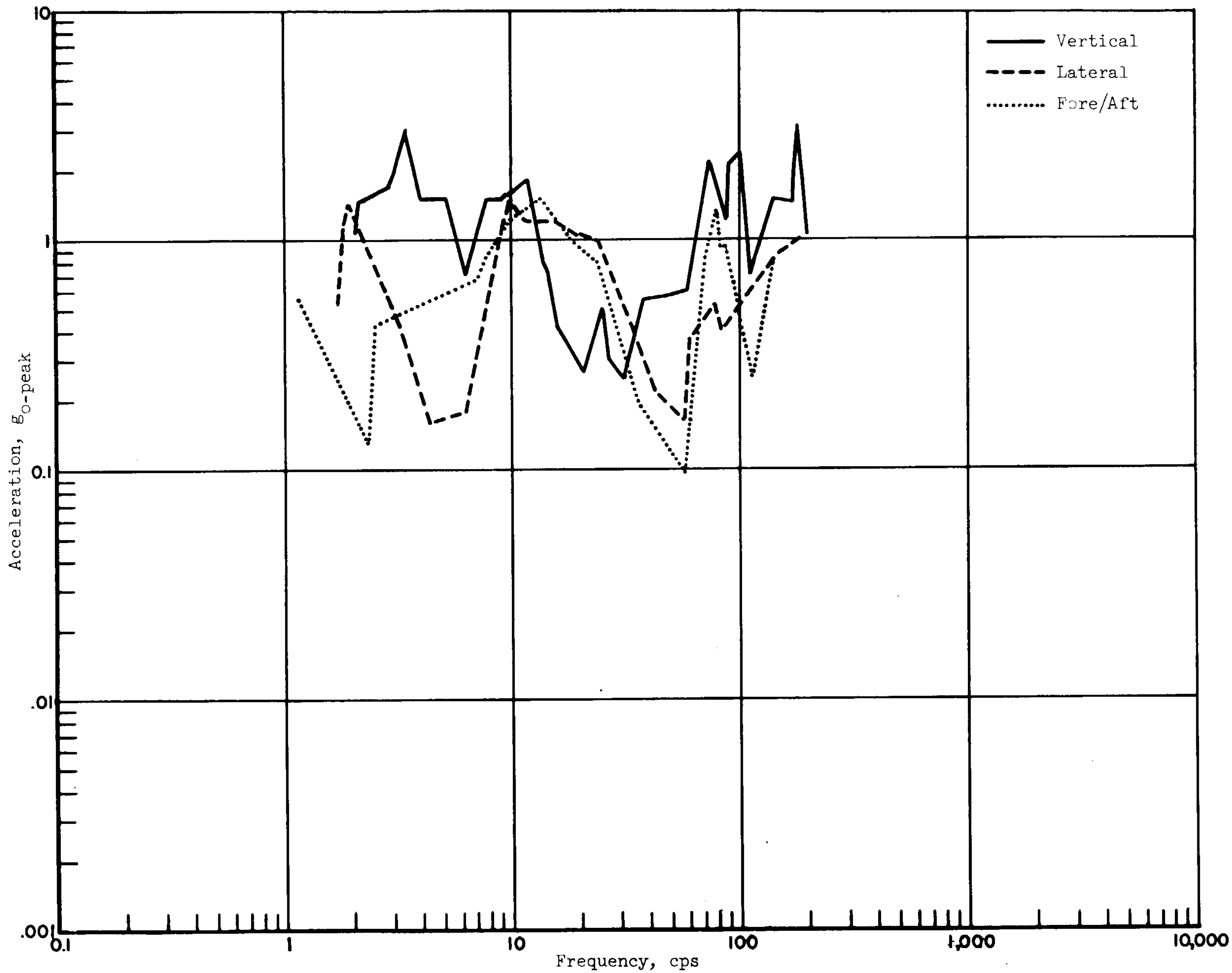


Figure 30. TRUCK ACCELERATION ENVELOPES - $2\frac{1}{2}$ TON DIRECTIONAL (Vertical, Lateral, Fore/Aft)

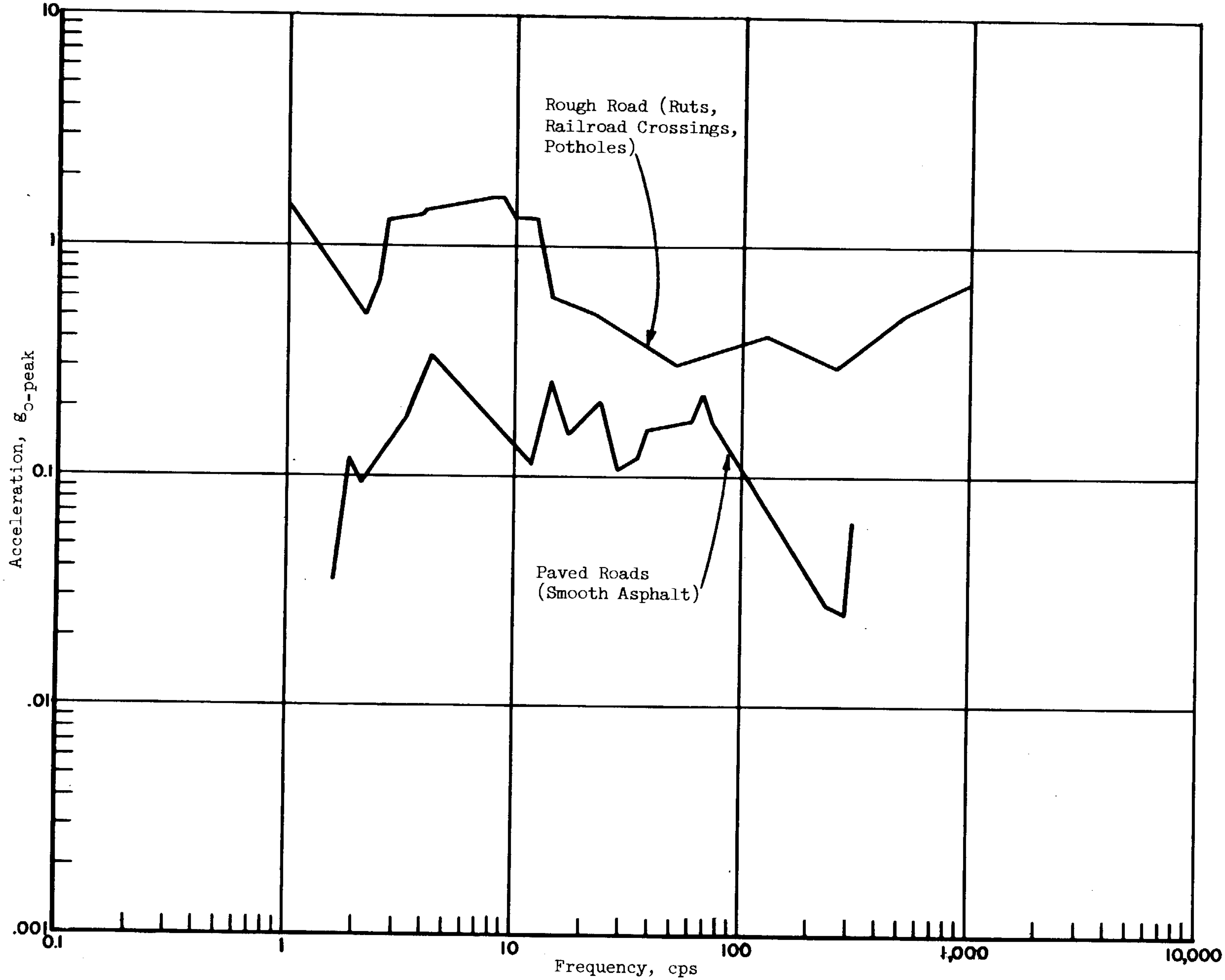


Figure 31. TRUCK ACCELERATION ENVELOPES - ROAD CONDITION COMPOSITES (Paved and rough roads)

during emergency maneuvers and slamming (the impacting of the ship on the water after the bow has left the water). This separation of data was made because the transient vibrations can usually be eliminated from the environment. Slamming for example, can be avoided if the ship avoids storm areas. Continuous vibrations are defined as those which occur during normal operations, including operations in rough seas.

Most of the collected data has been obtained from tests conducted by the David Taylor Model Basin. In their investigations, the area near the aft perpendicular (a line perpendicular to the water line at the stern) is usually monitored because it experiences the highest vibrational levels. The levels of vibration for the cargo area will be lower. However, since meager information is available for the cargo area, data for the stern location can serve as an upper bound of the environment.

The ship composite acceleration envelopes have been constructed (Figure 32) by enveloping all data referring to slamming and emergency maneuvers on one diagram and all data applying to continuous vibrations on another. The continuous vibration composite envelope includes data recorded under extremely rough sea conditions. This data is presented in terms of "g" (zero-to-peak) vs frequency (cps).

Diagrams showing the effect of sea state for two different ship lengths (L = 820 ft. and L = 380 ft.) are presented in Figures 33 and 34. In these investigations the sea states have been arbitrarly defined as shown on the diagram. Examination of these curves shows that the acceleration levels increase with increasing frequency from 4 to 10 cps and are constant at higher frequencies. The accelerations for the smaller ship (L = 380 ft., a destroyer) are almost twice as large as those for the large ship (L = 820 ft., an aircraft carrier). For each class of ship, the acceleration increases by a factor of two when the ship operates in a rough rather than in a smooth sea.

The higher vibration frequencies (> 10 cps) are due to machinery vibration and are seen to be less a function of sea state than the lower frequency rigid body motions.

Diagrams showing the effect of orientation (i.e. the effect of direction, Figure 35) and maneuvers (Figure 36) are the result of a single measurement program on a 572-ft. single screw cargo ship. In this program the vibrations were monitored with velocity pick-ups mounted at the main thrust bearing foundation and to an angle welded to the deck over the main transverse member at the aft perpendicular. The data was recorded with an oscillograph during operation in calm seas. Data was recorded during straight runs and standard maneuvers for various propeller speeds. The data was reduced by recording the highest ten percent of all amplitudes at a given frequency. Examination cf these curves show that the vertical vibration environment is the highest, followed by the lateral and fore-and-aft directions. Maneuvers generate higher levels of vibration than straight runs.

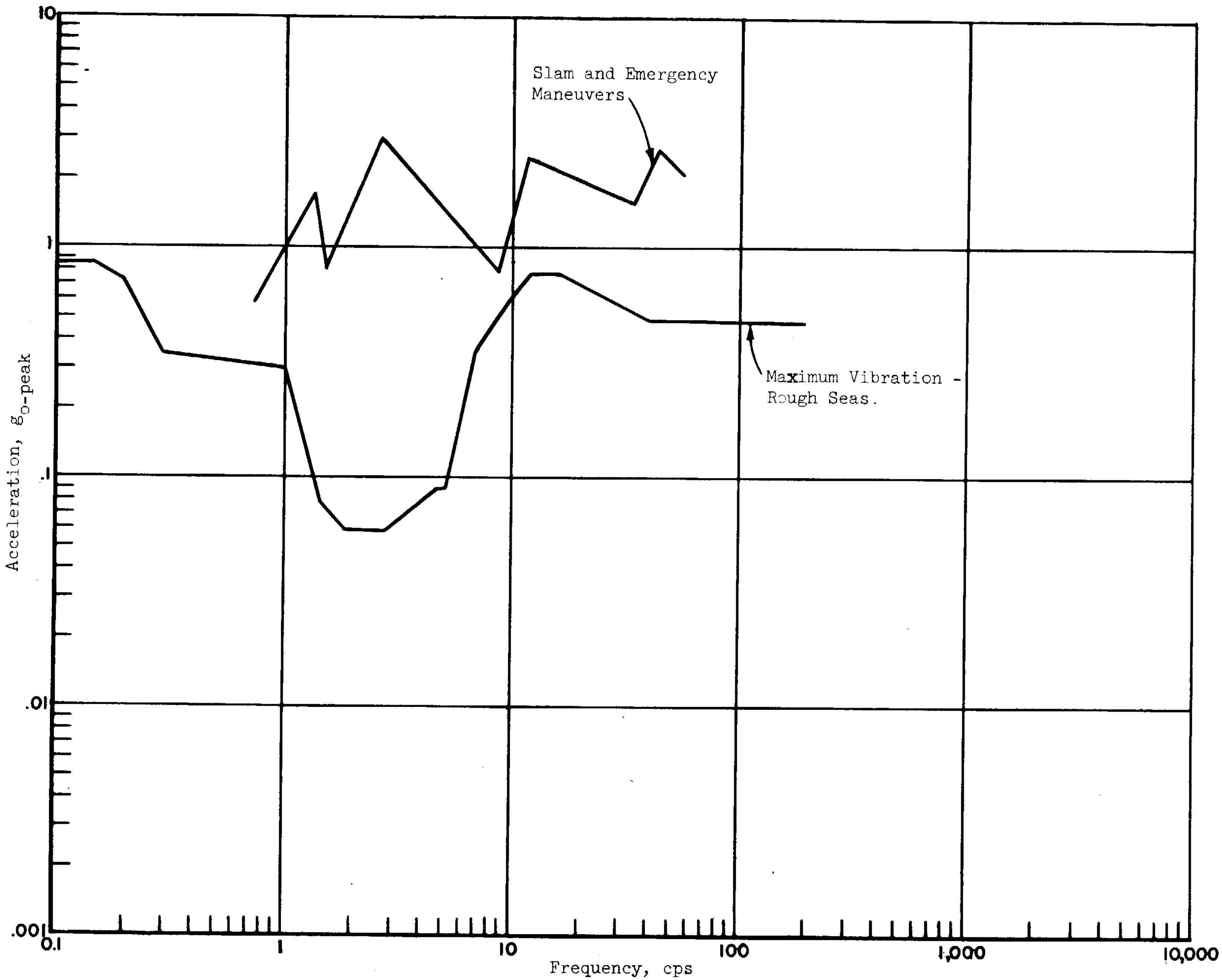


Figure 32. SHIP ACCELERATION ENVELOPES - COMPOSITES (Slam and emergency maneuvers; maximum vibration)

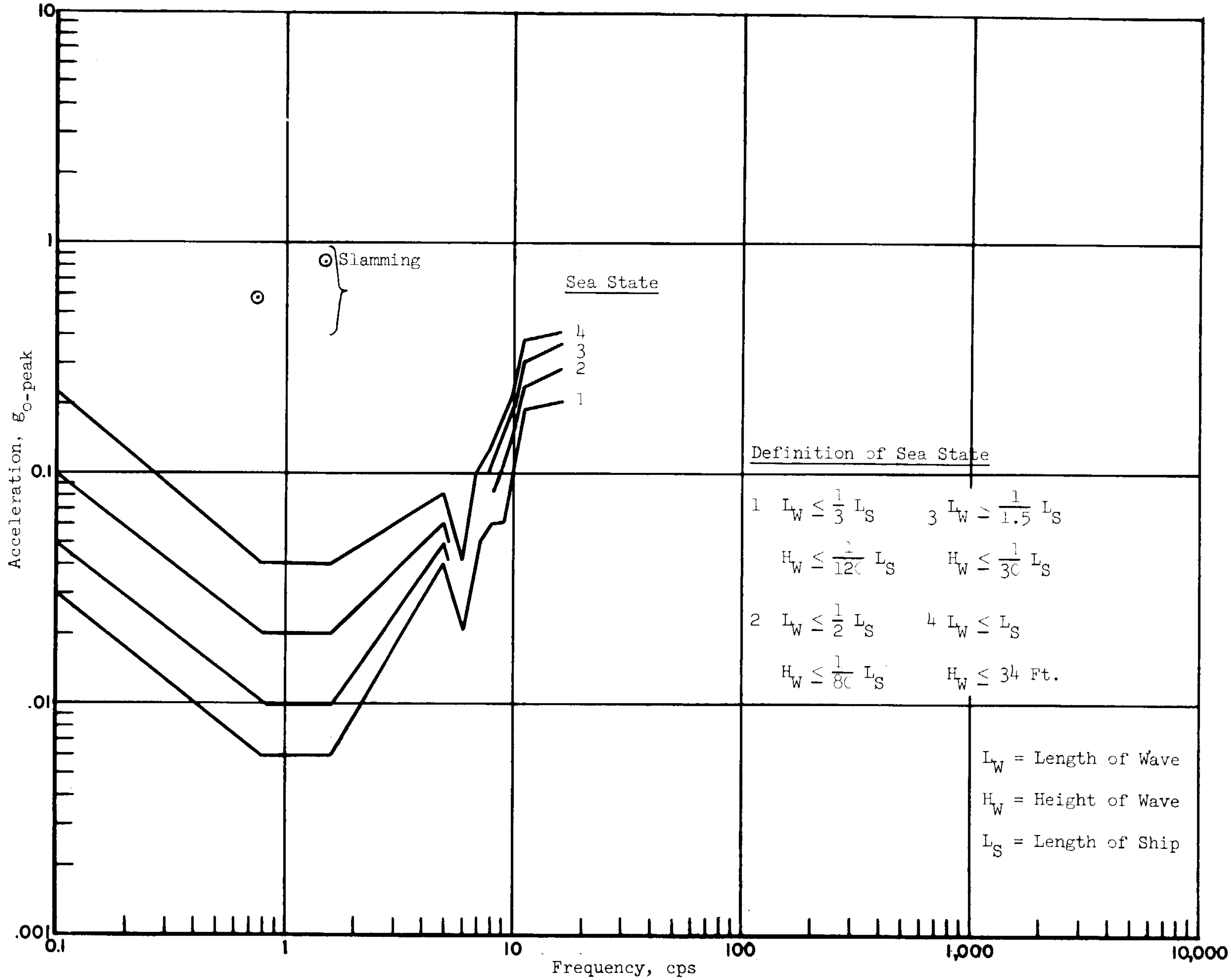


Figure 33. SHIP ACCELERATION ENVELOPES - SEA STATE (820 ft ship length)

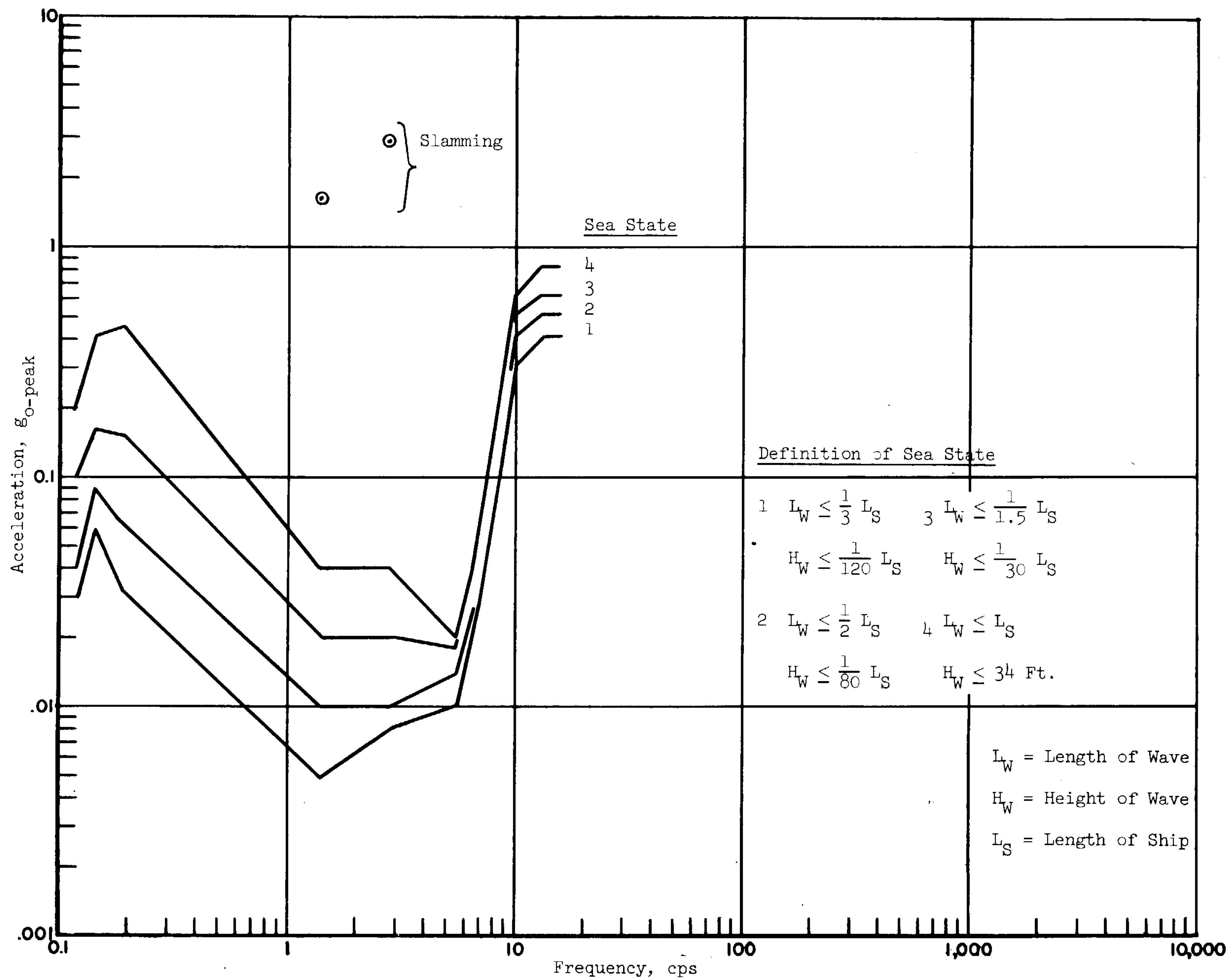


Figure 34. SHIP ACCELERATION ENVELOPES - SEA STATE (380 ft ship length)

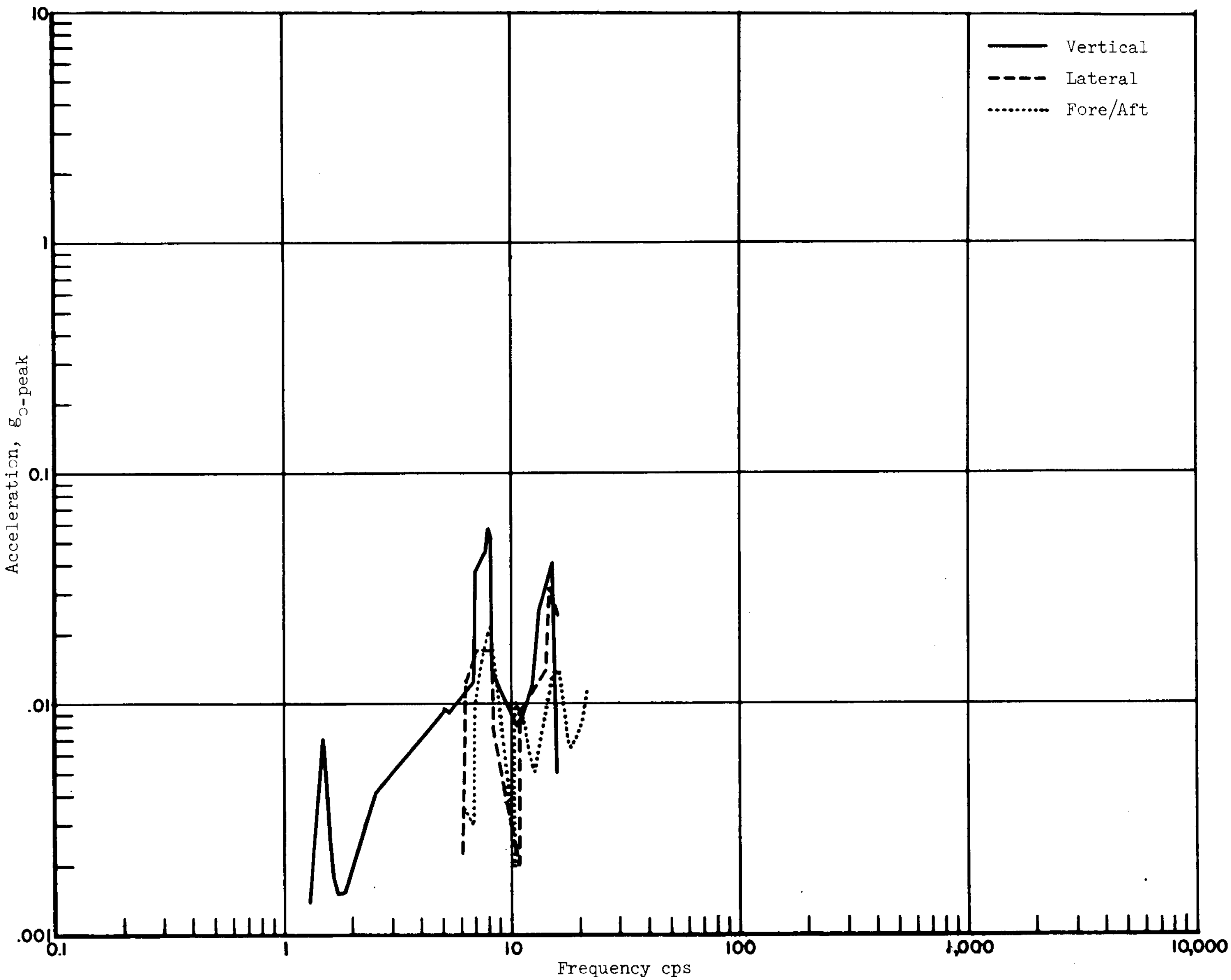


Figure 35. SHIP ACCELERATION ENVELOPES - DIRECTIONAL (Straight runs: Vertical, Lateral, Fore/Aft)

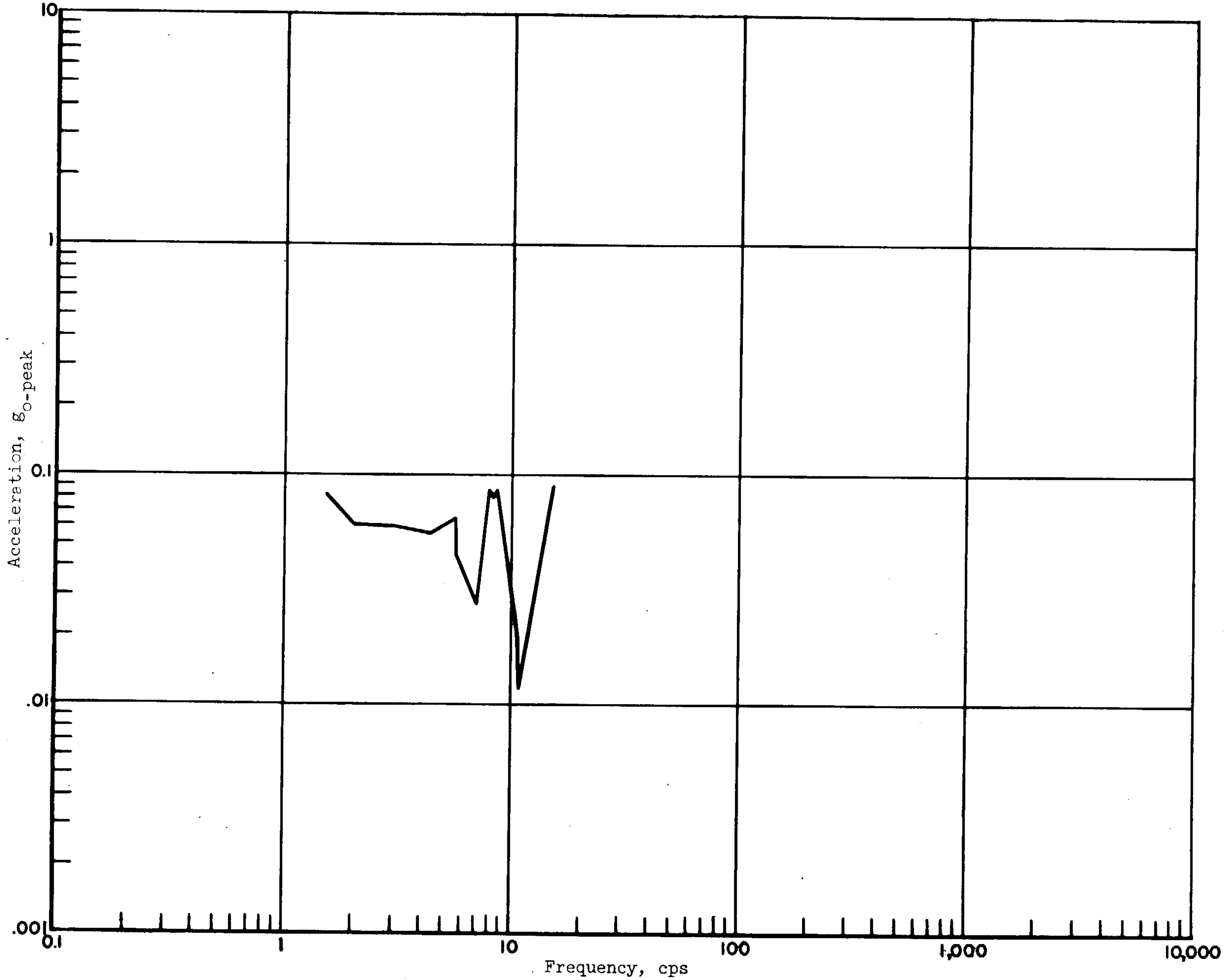


Figure 36. SHIP ACCELERATION ENVELOPES - NORMAL MANEUVERS

SECTION 4

CONCLUSIONS

The following order can be assigned to the quantity and quality of available data.

1. Aircraft
2. Ship
3. Railroad
4. Truck

The aircraft environment has been monitored extensively and the data is well documented. Shipboard environments are continuously monitored, but the measurements usually concern the fantail area. Little data could be found of measurements recorded in the cargo hold. Railroad data is meager. However, some continuous spectral analyses of data have been performed for this mode. Truck data is very sparse and none of the data located had been analyzed by continuous spectral methods.

The severest shock and vibration environment associated with transport vehicles occurs during railroad car coupling or humping operations.

The aircraft transportation environment exposes the transported item to the highest levels of continuous vibration. Railroad, truck, and ship follow aircraft in order of decreasing levels of vibration.

Data is sufficient at the present time to adequately define the transportation environment, that is, the input to the transported package, in terms of envelopes of maximum acceleration versus frequency. These diagrams facilitate a comparison of the environments for the various modes. Envelopes of acceleration versus frequency can also be constructed for categories within the major mode classifications. These diagrams describe the effect of various operating parameters on the environment.

Shock spectra are available to describe the environment which occurs during railroad car coupling operations. Spectra are available for a number of impact speeds for both standard and high capacity draft gear.

Statistical techniques could not be employed in a description of the transportation shock and vibration environment because of the paucity of data. However, a number of organizations have programs in progress which may provide the required data. Among these are the Sandia Corporation (trucks), United Technology Center (railroads), and David Taylor Model Basin (ships).

The levels of vibration presented on the various acceleration versus frequency diagrams are considered reliable even though they are derived from some data analyzed by rather crude data reduction techniques. This conclusion is based in part on the close correlation between data obtained from various sources (for similar test vehicles and test conditions). The aircraft data, although derived from individual test programs is considered reliable because these programs have employed the latest in instrumentation and data reduction techniques.

The acceleration levels presented on the diagrams represent the maximum environment that can normally be anticipated. When it is of particular interest that an item be neither over-designed nor under-designed, tests should be conducted for the particular vehicle and cargo. For the aircraft, ship and railroad modes of transportation, testing methods and procedures have been established by cognizant agencies. The truck mode of transportation has not been so controlled and is further complicated by the many variables which influence the environment (terrain, driver, RR crossings, speeds, etc.). Although a test course and test procedures have been standardized for military vehicles, no correlation has been established between the environment on this standard course (the Munson course) and the environment that would exist on normal commercial routes. Current practice by organizations investigating the shock and vibration environment on trucks is to subject the vehicle and cargo to very severe environments (e.g., 2 inch washboard, 4 x 4 timbers, buried logs, etc.). These test results are unrealistic and result in the over-design of packages or isolating equipment.

Other parameters than those presented in the report are known to influence the shock and vibration environment. However, data which would describe their effects are either very meager or unavailable. For example, in the railroad mode the end of a long train has a more severe environment than a car placed near the locomotive. Data could not be found which would substantiate this statement.

The frequency ranges examined by the various investigators were limited by the instrumentation and recording systems used and by the limits of interest of the particular programs. It should not be assumed, therefore, that vibration levels of significant amplitudes do not exist beyond the reported cut-off frequencies, either at the upper or lower ends.

SECTION 5

RECOMMENDATIONS

In view of the field measurement programs recently completed or in progress, a continued effort should be directed toward securing and incorporating this data into the transportation design manual. Updating of the design manual with this data is considered important because these studies are concerned with the modes of transportation having the least amount of data: railroad, ships, and trucks.

Since very sparse data is available which describes the environment in the cargo area of ships, an investigation should be conducted to determine if there is a correlation between the environment in this area and the environment at the fantail (the area which has been monitored extensively).

A study should be conducted to determine the possibility of prescribing a standard truck test course which would be representative of normal truck routes. Testing methods, instrumentation, and data reduction procedures should also be established.

Future effort should be directed toward processing some of the current data into a statistical format. Sufficient data, for example, is available for aircraft but this data has not been processed. Other modes of transportation, because of the random nature of the environment, should incorporate statistical techniques for data analysis and presentation. (Some of the more recent measurement programs are employing these newer techniques.) When sufficient data is available, the environments should be described by these methods.

A more extensive breakdown of the transportation modes should also be investigated. A description of the aircraft environment for various portions of flight, for example, would prescribe the environment occurring during the major portion of a flight (cruise). Other effects, such as location of railroad cars in a train, should be established with test data.

Additional effort should be expended in studies of methods to apply the data as currently presented and in recommending methods and techniques to employ in future measurement programs.

The terminal handling of cargo represents an important part of the transportation environment. This area should be investigated for descriptions of the environment for various handling operations (e.g., lift truck, dolly, man handling, docking, etc.).

SECTION 6

APPLICATION OF TRANSPORTATION SHOCK AND VIBRATION DATA TO PACKAGE DESIGN

6.1 Introduction

An item being transported will be subjected to a shock and vibration environment which may be severe enough to cause damage if steps are not taken to protect it. It is the purpose of this section to discuss the interpretation and use of the curves given in this report in the design of packages to mitigate the shock and vibration environment.

6.2 Damage Criteria

There are several damage criteria which may apply, singly or in combination, to an item being transported. These are:

1. first acceleration of the unit as a whole above its fragility rating

2. first displacement or stressing of a critical member of the unit above a certain value

3. first acceleration of a critical member of the unit above its own fragility rating

4. fatigue damage of critical members of the unit.

The first of these damage criteria can usually be applied by considering a one degree-of-freedom system, while the others require the analysis of at least a two degree-of-freedom system.

It is also possible that the isolator springs used to support the unit may bottom, thereby subjecting the equipment to a shock loading. Since it is difficult to determine the effect of such a shock upon the unit, it is best to consider the bottoming of the isolator as another source of damage and to design against such an occurrence. Thus, to the above list of damage criteria may be added:

5. bottoming of the isolator.

6.3 Nature of Transportation Shock and Vibration Environment

6.3.1 Vibration

Although the vibration found in all modes of transportation is to some extent random (because it cannot be specified precisely), the degree of randomness varies among the modes. In some instances, the original sources of vibration are random in that not only can't the magnitude be stated explicitly, but the spectral decomposition involves a wide band of frequencies. Such would be the case in road vehicles (trucks), where road irregularities are a primary source of excitation, and in jet aircraft, where the noise of the exhaust plays a leading role in producing the vibration environment. Other examples are atmospheric turbulence (all aircraft), and waves (ships).

In other instances, the source of vibration can be expected to involve discrete frequencies which are predictable. Typical examples are the excitation due to the unbalance in the engine of a vehicle, for which the frequencies are the operating frequency of the engine and its harmonics. For propeller aircraft and ships, an additional excitation is due to the interaction of the fluid pressure field and the vehicle body caused by the propeller. The dominant frequencies of this excitation are the blade passage frequency and its harmonics.

6.3.2 Shock

In addition to the steady excitation described above, additional excitation may result from impulsive (shock) loadings. These include ramp backing (trucks), landing (aircraft), and switching and coupling (trains). Only the data for railroad coupling shocks is considered separately in this report.

6.3.3 Package Environment

The excitations described above are imposed upon the vehicle structure. What is needed in the design of package protection is the response of the transport vehicle in the cargo area. It is this response to which the package is exposed. If the original excitation is of the discrete frequency type, then the structural vibration will occur at these frequencies. If the original excitation is wide-band, the response of the structure will occur primarily at its natural frequencies.

6.3.4 Random vs Deterministic Interpretation

As mentioned above, the vibration sources in trucks and jet aircraft are wide-band random processes, and, therefore, the environments in the cargo areas of these vehicles are random. There is, however, some question as to whether the periodic vibration sources, and resulting cargo space environment, in ships, propeller aircraft, and trains are random or deterministic. It would seem that these environments should be deterministic since the forces causing them are essentially constant as long as the conditions under which the vehicles are operated remain unchanged

Concerning this point, the findings of two teams of investigators are in conflict. When analyzing the structural vibration in reciprocating engine aircraft, the various time histories were passed through narrow-band filters. In one case, the output of each filter was constant, indicating that the vibration was deterministic, while in the other case, the amplitude of the output was distributed in a random fashion.

6.4 Use of Data in Design

6.4.1 Optimum Form for Data

If overdesign of package protection will lead to excessive size and weight, it is desirable to have the shock and vibration data in as much detail as possible, so that an economical design may be achieved. For example, if the mode of transportation being considered is air, the desired data would show the variation of the environment with the location in the aircraft (cargo deck forward, cargo dect aft, etc.), and with the direction of vibration (lateral, vertical, horizontal). With such information if would be possible to immediately limit the environment to which the unit is exposed by proper location and orientation of the package in the aircraft. Unfortunately, such information is not always available.

The ideal quantitative presentation of vibration data for a given mode of transportation consists of the following information:

1. the mean square spectral density for the operating condition which leads to the severest vibration environment

2. a listing of the dominant frequencies of the environment

3. a statement as to whether the environment is essentially deterministic or can be considered random and Gaussian.

With such a presentation, the task of the package designer is reduced to the following procedure:

1. comparison of allowable fragility curve (g vs frequency) with input

2. selection of a natural frequency based on input spectrum and allowable response spectrum (if an isolating system is required)

3. computation of the RMS response caused by the severest environment using the spectral density

4. conversion to the maximum response if the environment is deterministic or calculation of the probability of exceeding a critical value if the environment is random

5. comparison with the allowable response to determine if the design is adequate.

6.4.2 Available Form of Vibration Data

Unfortunately, information is not available in the form described above except in isolated instances. The two most popular forms of data presentation are:

1. scatter diagrams of observed peak accelerations (or displacements) vs frequency

2. scatter diagrams of RMS accelerations (or displacements) vs frequency.

Such diagrams include measurements made at many operating conditions and on many vehicles. In this report, envelope curves are drawn over the points obtained by this form of data reduction. Any point on the envelope curve represents the severest measured vibration at that frequency.

6.4.2.1 RMS Envelopes

If an RMS envelope curve is given, the designer may use the following procedure:

1. comparison of allowable fragility curve with input

2. selection of a natural frequency for an isolating system (if required)

3. computation of the RMS response at the natural frequency assuming a sinusoidal excitation with the RMS value equal to the envelope value at that frequency. If the isolator is narrow-band, contributions at other frequencies are probably negligible

4. computation of the maximum response if the environment is deterministic or, if random, computation of the probability of exceeding a critical value

5. comparison of maximum response with allowable response.

6.4.2.2 Peak Envelopes

When an envelope of peak values is given, the procedure is not as simple because problems arise in the interpretation of the envelope. The first difficulty is that the envelope does not really represent the maximum environment at non-dominant frequencies. For example, the envelope value at the natural frequency may be based upon one or two isolated peaks which occurred at that frequency and the steady state excitation at the amplitude indicated by the envelope does not really exist. In other words, the environment at the natural frequency is not as severe as implied. There seems to be no rational method of improving the estimate.

A second difficulty arises when the environment is random. In such a case, it is desirable to know the RMS of the response, which necessitates knowledge of the RMS of the environment. If a peak envelope is given, estimation of the RMS is dependent upon the quality of the data. For example, if the data is good enough so that there is only a 1.1% chance that a peak (at the severest operating condition) will have an amplitude greater than the envelope value, then, assuming a Gaussian process, the true RMS value at the severest condition is about 1/3 the envelope value (based on the Rayleigh distribution of peaks). That is, with such an RMS value, the envelope is assumed to encompass 98.9% of all peaks which will ever be encountered. If the data is considered to be of exceptional quality, then the RMS value may be taken as 1/4 the envelope value and the envelope will encompass more than 99.96% of all peaks. Thus,

$$\text{RMS} = \frac{\text{Peak Envelope Value}}{N}$$

where N = number of standard deviations

and an important part of the analysis is to choose N. Since, as mentioned earlier the data already leads to a conservative design, it appears safe to assume a high confidence level and choose N = 4.

If the environment is sinusoidal the RMS value is simply the peak value divided by $\sqrt{2}$.

Thus, when an envelope of peak values is given, the procedure is:

1. determination of the nature of the environment

   a. if deterministic - direct computation of peak response

   b. if random - conversion from peak envelope to RMS envelope

2. continuation of procedure outlined above.

The assumption of a random environment usually leads to a more unconservative design than does the assumption of a sinusoidal environment.

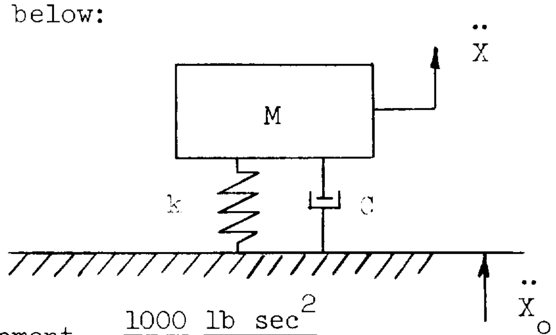
6.5 Illustrative Example

The general principles discussed above will be illustrated and amplified in the following example:

A missile component weighing 1/2 ton (1000 lbs.) is to be transported by an H-37 Helicopter. The trip will take 30 minutes. The unit must be stowed in such a way that it is most susceptible to damage by vertical vibration. Design an isolator so that a fragility rating of 10 g vertically is not exceeded. Specify the minimum isolator travel which can be expected to prevent bottoming shocks.

6.5.1 Analysis of Unit as a Whole

Unless the designer is willing to perform a detailed analysis involving coupled modes, the physical situation may be represented by the single degree of freedom system shown below:



M = mass of equipment - $\frac{1000}{386}\ \frac{\text{lb sec}^2}{\text{in}}$

k = spring constant of isolator - $\frac{\text{lb}}{\text{in}}$ (to be determined)

C = damping constant of isolator - $\frac{\text{lb sec}}{\text{in}}$ (to be determined)

X = absolute vertical displacement of unit - in.

$\ddot{X}$ = absolute vertical acceleration of unit - in/sec^2

X_o = vertical displacement of cargo deck - in

$\ddot{X}_o$ = vertical acceleration of cargo deck - in/sec^2

δ = $X - X_o$ = deflection of isolator spring - in

$\ddot{\delta}$ = relative acceleration of unit with respect to cargo deck - in/sec^2

$\dot{\delta}$ = relative velocity of unit - in/sec

Letting

$\omega_n = \left(\frac{k}{M}\right)^{1/2}$ = natural frequency of package - rad/sec

$\xi = \frac{C}{C_c} = \frac{C}{2(kM)^{1/2}}$ = isolator damping factor (fraction of critical damping)

the equation of motion of the unit may be written

$$\ddot{\delta} + 2\xi\omega_n\dot{\delta} + \omega_n^2\delta = -\ddot{X}_o$$

The following transfer functions for the quantities of interest may be derived from the differential equation:

$$\frac{\delta}{\ddot{X}_o} = |H_\delta(\omega)| = \left[\frac{1}{(\omega_n{}^2 - \omega^2)^2 + 4\xi^2\omega^2\omega_n{}^2}\right]^{1/2}$$

$$\frac{\ddot{X}}{\ddot{X}_o} = |H_{\ddot{x}}(\omega)| = \left[\frac{\omega_n{}^4 + 4\xi^2\omega^2\omega_n{}^2}{(\omega_n{}^2 - \omega^2)^2 + 4\xi^2\omega^2\omega_n{}^2}\right]^{1/2}$$

6.5.1.1 Preliminary Considerations

A choice of isolator parameters must be made. The natural frequency should be chosen either within the vibration-free bandwidth (20-30 cps for propeller aircraft in this report)*, or at a frequency for which the envelope value is small. Therefore, it will be chosen below 15 cps or between 20-30 cps. If it is too low, the displacement response to any shocks which the package may sustain may be too great. If it is too high, the acceleration response to shock may be too great. Furthermore, too high a natural frequency increases the possibility of damage, particularly of fatigue damage, since there are more cycles. Thus, the choice of a natural frequency is a compromise. Similarly too low a damping factor gives a large response at the natural frequencies, while too high a damping factor may produce too great a response at the other frequencies.

Tentative values of natural frequency and damping factor will be chosen as

$$\omega_n = 5 \text{ cps} = 10 \text{ rad/sec}$$

$$\xi = .02$$

Reference to the directional summary for the H-37, shows that the envelope value at this frequency is .09 g R.M.S.

The R.M.S. responses are given by the following equation:

$$\text{RMS response} \approx B|H(\omega_n)|$$

where B is the RMS envelope value at the natural frequency ω_n.

6.5.1.2 Analysis of Excessive Acceleration

The possibility of exceeding the fragility rating will be examined first. The value of the acceleration transfer function, $H_{\ddot{x}}$, at the natural frequency is

$$|H_{\ddot{x}}(5 \text{ cps})| = \left[\frac{(10\pi)^4 + 4(.02)^2 (10\pi)^2 (10\pi)^2}{4(.02)^2 (10\pi)^2 (10\pi)^2}\right]^{1/2} = 25.0$$

*Except for the Pregnant Guppy

The RMS acceleration response is then

$$(\ddot{X})_{RMS} = (.09\ g)\ (25.0) = 2.25g$$

It should be noted that this calculation is approximate if the amplitude is random. If the input is a slowly varying quasi-sinusoid, its spectrum is very narrow band. Thus all contributions may be assumed to occur at the center frequency. It is also assumed that other dominant frequencies are outside the system bandwidth so that their contributions are negligible.

The procedure at this point is dependent upon the nature of the environment. If it is deterministic, the maximum acceleration to which the equipment will be subjected is

$$(2.25\ g) \times \sqrt{2} = 3.19\ g$$

which is well below the fragility rating of 10 g. For the purposes of illustration, however, assume that the environment is random and Gaussian. Then the distribution of peaks follows the Rayleigh law

$$P(a) = 1 - \exp\left[-\frac{a^2}{2\sigma^2}\right]$$

where σ is the RMS value, and $P(a)$ is the probability that any particular peak will have an amplitude smaller than a. Then the probability that any peak acceleration will be less than 10 g is

$$P(10\ g) = 1 - \exp\left[-\frac{1}{2}\left(\frac{10\ g}{2.25\ g}\right)^2\right] = 1 - e^{-9.88} = .9995$$

This probability implied virtual certainty that a given acceleration peak will not exceed 10 g. However, the probability which is really of interest is that of <u>no peak having an amplitude above 10 g throughout the entire flight</u>. This probability is smaller than .99995. (As an analogy, the probability of not getting a head when a fair coin is tossed is 1/2, but the probability of not getting a head when the coin is tossed twice is only 1/4.)

Because the vibration is narrow-band, the amplitudes of adjacent peaks are not independent and it is not possible to compute the desired probability. An order of magnitude, however, may be obtained by assuming independence, and the probability of survival is approximately

$$(1 - e^{-9.88})^N$$

where N represents an appropriate number of acceleration peaks. Since the flight is 30 minutes long, the total number of peaks is

$$30 \text{ min.} \times \frac{60 \text{ sec}}{\text{min.}} \times \frac{5 \text{ cycles}}{\text{sec.}} \times 2 \frac{\text{peaks}}{\text{cycle}} = 18{,}000$$

Then the probability, Q, of not exceeding the fragility rating throughout the entire flight is

$$(1 - e^{-9.88})^{18{,}000} \approx Q$$

Now for $Z << 1$,

$$\ell n(1 + Z) \approx Z$$

Then

$$\ell n\, Q = 18000\, \ell n\, (1 - e^{-9.88}) = -18000\, e^{-9.88} \approx -.90$$

and

$$Q = .406$$

Thus there is a 59% chance of exceeding the fragility rating during the flight. This is too great a risk.

Changing the isolator design so that $\xi = .03$ yeilds

$$|H_{\ddot{x}}\ (5 \text{ cps})| = \left[\frac{(10\pi)^4 + 4(.03)^2(10\pi)^2(10\pi)^2}{4(.03)^2(10\pi)^2(10\pi)^2}\right]^{1/2}$$

$$= 16.7$$

and the RMS acceleration becomes

$$(\ddot{X})_{RMS} = (.09 \text{ g})(16.7) = 1.5 \text{ g}$$

Then

$$P(10g) = 1 - \exp\left[-\frac{1}{2}\left(\frac{10 \text{ g}}{1.5 \text{ g}}\right)^2\right] = 1 - e^{-22.2}$$

and

$$Q \approx (1 - e^{-22.2})^{18000}$$

$$\ell n\, Q \approx -18000\, e^{-22.2}$$

$$\ell n\, (-\ell n\, Q) = \ell n\, 18000 - 22.2 = -012.4$$

$$\ell n\, Q = e^{-12.4}$$

$$Q = 1 - e^{-12.4} > .99999$$

Thus, avoiding damage during the trip is virtually certain.

6.5.1.3 Analysis of Bottoming Shocks

To examine the possibility of bottoming the isolator, the deflection transfer function is evaluated

$$|H_\delta(5\text{ cps})| = \left[\frac{1}{4(.03)^2(10\pi)^2(10\pi)^2}\right]^{1/2} = 1.7 \times 10^{-2}\ \text{sec}^2$$

Then the RMS deflection is

$$(\delta)_{RMS} = (.09\ g) \times 386\ \frac{\text{in/sec}^2}{g} \times 1.7 \times 10^{-2}\ \text{sec}^2$$
$$= .59\ \text{in.}$$

If the environment is deterministic, the maximum deflection is

$$(.59\ \text{in}) \times\sqrt{2} = .84\ \text{in.}$$

and a clearance of 1 inch would be sufficient.

For Gaussian environment, the clearance should be 3.5 inches as shown by the following calculation

$$Q = \left[1 - e^{-\frac{1}{2}\left(\frac{3.5}{.59}\right)^2}\right]^{18000} = \left[1 - e^{-17.6}\right]^{18000}$$
$$\cong 1 - e^{-7.8} = .9996$$

and the probability is high that the isolator will not bottom.

6.5.1.4 Final Isolator Design

The design for the isolator is, therefore,

$$\omega_n = 5\text{ cps} = 10\pi\ \text{rad/sec.}$$
$$\xi = .03$$

The spring and damping constants are:

$$k = M\omega_n^2 = \frac{1000}{386}(10\pi)^2 = 2560\ \text{lb/in}$$

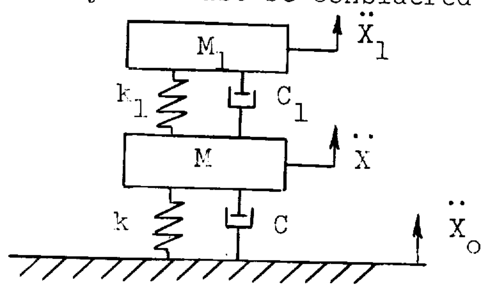
$$C = 2\xi(kM)^{1/2} = 2 \times (.03) \times \left(2560 \times \frac{1000}{386}\right)^{1/2} = 4.9\ \text{lb sec/in}$$

If space limitations require a smaller isolator clearance, the damping factor can be increased.

6.5.2 Analysis of Critical Member

6.5.2.1 Analysis of Excessive Displacement

The analysis for the first excursion damage of critical internal member of the equipment is similar to the procedure just discussed. The only difference is that a two degree of freedom system must be considered



As before, k and C are the parameters for the isolator and M is the mass of the equipment. M_1 is the mass of the critical member and it is assumed that $M_1 << M$. k_1 and C_1 are the associated spring constant and damping constant. X_1 is the absolute displacement of the critical member. The transfer function for the relative displacement, $\delta_1 = X_1$, between M_1 and M is

$$|H\delta_1(\omega)| = \left[\frac{\omega_1{}^4 + 4\xi^2\omega_1{}^2\omega^2}{(\omega^4 - \omega^2\omega_1 - \omega^2\omega_2{}^2 + \omega_1{}^2\omega_2{}^2)^2 + (2\xi\omega^3\omega_1 + 2\xi_1\omega_2\omega^3 - 2\xi\omega\omega_1\omega_2{}^2 - 2\xi_1\omega\omega_2\omega_1{}^2)^2}\right]^{1/2}$$

where

$$\frac{k}{M} = \omega_1{}^2; \quad \frac{k_1}{M_1} = \omega_2{}^2; \quad \frac{c}{\sqrt{kM}} = 2\xi; \quad \frac{C_1}{\sqrt{k_1 M_1}} = 2\xi_1$$

and

$$\frac{M_1}{M} << 1; \quad \xi\xi_1 << 1$$

Evaluated at the two natural frequencies, the resonance values are

$$|\ H\delta_1\ (\omega_1)\ | = \frac{\sqrt{1 + 4\xi^2}}{2\,\xi|\omega_1{}^2 - \omega_2{}^2|}$$

$$|H\delta_1(\omega_2)| = \frac{\omega_1\sqrt{\omega_1^2 + 4\xi^2\omega_2^2}}{2\xi_1\omega_2^2|\omega_2^2 - \omega_1^2|}$$

6.5.2.2 Analysis of Fatigue

A proper analysis of the fatigue damage to the critical member is more complex. To perform it corectly, data must be given for each operating condition along with the time durations. Because such information is not available, the envelope must be used. Since this will lead to a conservative design, (because the unit is not always subject to so severe an environment), it seems unnecessary to take the added precaution of considering the randomness of the vibration and protecting against the possibility of very large excursions. Thus, all that need be done is to calculate the total number of cycles for the trip and check the stress vs cycles to failure fatigue curve to determine whether or not the peak stress exceeds the critical stress.

In this connection, there again arises the problem of finding the peak value (of excitation or response) if the RMS envelope of excitation is given. For a deterministic environment, peak = RMS x $\sqrt{2}$, while for a random environment, peak = N x RMS where, as stated before, N may be taken between 3 and 4. A compromise seems in order with the factor chosen between 1.4 and 4. Its value might be influenced by the presence of notches or other areas of stress concentration which could lead to trouble, and by consideration of whether or not partial accumulation of damage is acceptable. In other words, the magnitude would be determined by the required margin of safety.

Another difficulty which arises is that the vibration does not occur at a single frequency but has components at the two natural frequencies. Thus, even if the environment were deterministic, the response time history would be a superposition of two sinusoids. There is no established method of applying fatigue curves derived from sinusoidal testing to this situation. It may turn out, however, that the contribution to the total response at one of the frequencies (usually ω_1, with the assumptions listed in 6.5.2.2) is much smaller than the contribution at the other frequency, so that the response can be considered to be a single frequency process.

6.5.3 Random vs Deterministic Interpretation

The question which has remained unanswered is whether the environment should be treated as random or deterministic. Because of the nature of the source of the excitation, there is no doubt that the environments found in road vehicles and jet aircraft should be considered random. But, as pointed out previously, the major source of vibration in propeller aircraft and ships is definitely periodic and the amplitude should not vary when operating conditions are kept constant. The available data does not allow a definite conclusion to be made. It should be pointed out that a random approach can lead to greater damping and clearance requirements, as can be seen from the example given.

6.6 Analysis of Shock Excitations

The discussion thus far has been concerned with stationary vibration. Another environment which must be considered is shock. A shock analysis is particularly important if the equipment is to be shipped by rail, for it may be subjected to coupling and switching impacts.

The calculation of the response of a system to a shock input must be handled in a manner different from the calculation of the response to a steady excitation. The reason is that a shock is a transient phenomenon and cannot be expressed in terms of temporal averages (such as mean square spectral density). Thus, no meaningful averages of the response to a shock can be computed and it becomes necessary to compute the entire response in order to obtain any useful information. Since the response to a shock depends critically upon the pulse shape, the time history must be known fairly accurately.

6.6.1 Shock Spectra

In some instances, it may be sufficient to know the maximum value of the response. A different method of presenting shock data is to give, not the excitation history, but the effect of the shock upon single degree of freedom systems of various natural frequencies and damping factors. The maximum value of the response is found for each system (either analytically or by analog) and plotted against the natural frequency with the damping factor as a parameter. Such a presentation is called a shock spectrum. It is particularly useful when the nature of a shock applied to an equipment is independent of the impedance of the equipment. This might be the case with shocks applied to freight car lading during switching and coupling. By mounting measuring devices in the car, the maximum responses to the shocks, at various natural frequencies and damping factors, can be determined. If it is necessary to design isolators to protect cargo, the shock spectra can be examined and the optimum natural frequency found.

6.6.1.1 Types of Shock Spectra

Shock spectra are of various types such as initial spectra, which give the extrema which occur while the shock is being applied, residual spectra, which give extrema which occur after the shock has acted, positive spectra, which give positive extrema, and negative spectra. In addition, spectra may be defined for displacement, velocity and acceleration. Usually, velocity and displacement spectra are derived from the acceleration spectrum by dividing the values by the frequency and square of the frequency, respectively. This is an approximation, but the relationship between acceleration and displacement becomes exact as the damping approaches zero. An interesting property of the residual displacement spectrum for an undamped system is that it is proportional to the product of the frequency and the absolute value of the Fourier decomposition of the acceleration shock history.

6.6.1.2 Use of Shock Spectra for Two Degree-of-Freedom Systems

The use of shock spectra can be extended to obtain the approximate response of a multi degree-of-freedom system by using normal modes. The approximation is due to the fact that it is not known how the normal mode responses should be combined.

An an illustration, consider the two degree-of-freedom systems discussed earlier. Assuming a fixed base, the equations of motion for the free vibrations of the undamped system are

$$\ddot{\delta} + \ddot{\delta}_1 + \omega_2^2 \delta_1 = 0$$

$$\ddot{\delta} + \omega_1^2 \delta - \omega_2^2 \mu \delta_1 = 0$$

where $\mu = \frac{M_1}{M}$ and the other symbols have the same meaning as before. Assuming a sinusoidal response, these equations may be written

$$\delta_1(\omega^2 - \omega_2^2) + \delta(\omega^2) = 0$$

$$\delta_1(\mu\,\omega_2^2) + \delta(\omega^2 - \omega_1^2) = 0$$

The natural frequencies are found by setting the determinant of the systems of equations equal to zero which results in the quartic equation

$$\omega^4 - \omega^2\,\omega_2^2(1+\mu) + \omega_1^2 + \omega_1^2\,\omega_2^2 = 0$$

Assuming $\mu << 1$, the natural frequencies are approximately ω_1 and ω_2. The mode shapes are approximagely

$$\frac{\delta^1}{\delta_1^{\,1}} \approx \frac{\omega_2^2}{\omega_1^2} - 1, \quad \frac{\delta^2}{\delta_1^{\,2}} \quad \frac{\mu\,\omega_2^2}{\omega_1^2 - \omega_2^2}$$

where the superscripts refer to the mode and δ^1 and $\delta_1^{\,1}$ correspond to ω_1, etc.

A measure of the contribution of each mode to the total response is given by the participation factor. For this system, the factors for δ_1 are:

$$\Delta_1^n = \frac{\left(\frac{\delta^n}{\delta_1^{\,n}}\right) + \mu\left(\frac{\delta^n}{\delta_1^{\,n}} + 1\right)}{\left(\frac{\delta^n}{\delta_1^{\,n}}\right)^2 + \mu\left(\frac{\delta^n}{\delta_1^{\,n}} + 1\right)^2}$$

where n = 1, 2. The corresponding factors for δ are obtained by multiplying Δ_1^n by the appropriate mode shape.

To get the contribution of each mode to the total response, the participation factor is multiplied by the value of the shock spectrum at that frequency.

The problem is in combining the contribution of each mode to obtain the maximum response. There are two methods. In the first, the absolute value of the contributions are added and the result is the upper bound for the maximum response. The second method involves finding the square root of the sum of the squares of the contributions. This gives the most probable value of the maximum response, but the sign of the error is unknown.

Let it be required to estimate the maximum deflection (δ_1) of the spring k_1, when the shock is applied to the base. Letting $S(\omega)$ be the displacement shock spectrum, the contribution of frequency ω_1 is $S(\omega_1)\Delta^1$, and the contribution at frequency ω_2 is $S(\omega_2)\Delta_1^2$. Then the upper bound for the maximum deflection is

$$|S(\omega_1)\Delta_1^1| + |S(\omega_2)\Delta_1^2|$$

and the most probable value is

$$\sqrt{\left[S(\omega_1)\Delta_1^1\right]^2 + \left[S(\omega_2)\Delta_1^2\right]^2}$$

6.6.2 Use of Railroad Shock Spectra

If an item is to be shipped by rail, a coupling shock analysis is a necessary part of the design procedure. Since such an analysis consists of inspecting a shock spectrum and choosing a natural frequency and damping factor which results in an allowable maximum response, it is best to perform this selection first. These parameters can then be used in checking the other design criteria.

The maximum responses are dependent upon the velocity of the switching or coupling operation. A curve is given showing the frequencies of occurrence of velocities encountered in practice. Rather than perform a random analysis, the design can be chosen on the basis of, say, an 8 MPH impact, and a specification given to the carrier that all operations are to be done at a maximum of 6 MPH.

In summary, the procedure used in designing against railroad shock damage is:

1. selection of design velocity and specification of maximum permissible velocity

2. inspection of proper shock spectrum and selection of natural frequency and damping factor which limit response to an allowable maximum (with due regard to factors discussed earlier)

3. use of these parameters in analyzing other damage criteria.

APPENDIX A

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APPENDIX A

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APPENDIX B

LISTING OF AGENCIES AND ORGANIZATIONS CONTACTED FOR SHOCK AND VIBRATION ENVIRONMENTS OF TRANSPORT VEHICLES

Barry Controls
700 Pleasant Street
Watertown, Massachusetts

The Boeing Company
Aero-Space Division
Minuteman Branch
Seattle, Washington

Lord Manufacturing Company
1635 West 12th Street
Erie, Pennsylvania

Florida Engineering and
Industrial Experiment Station
University of Florida
Gainesville, Florida

General Motors Corporation
GMC Truck and Coach Division
660 South Boulevard, East
Pontiac, Michigan

Neway Equipment Co.
Muskegon, Michigan

Southern Pacific Company
65 Market Street
San Francisco 5, California

The Atchison, Topeka, & Santa Fe
Railway System
80 East Jackson Boulevard
Chicago, Illinois

Hendrickson Tandem Co.
8001 West 47th Street
Lyons, Illinois

Freightliner Corporation
P. O. Box 3591
Portland, Oregon

Trailmobile, Inc.
31st and Robertson
Cincinnati, Ohio

New York Central System
266 Lexington Ave.
New York 17, New York

The Pennsylvania Railroad
6 Penn Plaza
Phileadelpha 4, Pennsylvania

The Chesapeake and Ohio Railway
1610 Terminal Tower
Cleveland, Ohio

Naval Weapons Handling Laboratory
Earle, New Jersey

U. S. Army Natick Laboratories
Natick, Massachusetts

Southern Railway System
P.O. Box 1808
Washington 13, D. C.

Union Pacific Railroad
Transportation Division
1416 Dodge Street
Omaha 2, Nebraska

U. S. Naval Research Laboratory
Washington 25, D. C.

Office of the Director of
Defense Research and Engineering
Washington 25, D. C.

Thiokol Chemical Corp.
Utah Division
Brigham City, Utah

Diamond T. Motor Company
1331 South Washington
Lansing, Michigan

Ford Motor Company
Rotunda Drive at Southfield Rd.
P. O. Box 658
Dearborn, Michigan

White Motor Company
842 East 79th Street
Cleveland, Ohio

U. S. Atomic Energy Commission
Technical Information Division
P. O. Box 62
Oak Ridge, Tennessee

U. S. Army
Transportation Engineering Agency
Office of the Chief of Transportation
Washington, D. C.

Aerojet-General Nucleonics
P. O. Box 77
San Ramon, California

Lockheed Missile & Space Division
Sunnyvale, California

Aerojet-General Corporation
Sacramento, California

Fruehauf Corporation
Detroit, Michigan

Ordnance Tank Automotive Command
Research and Development Division
Land Locomotion Research Branch
Midland, Michigan

Bureau of Naval Weapons
Special Projects Office SP-27
Washington, D. C.

Society of Naval Architects and
Marine Engineers
79 Trinity Place
New York, New York

Michigan State University
School of Packaging
East Lansing, Michigan

David W. Taylor Model Basin
Washington, D. C.

Engineering Research and Development
Laboratories
Fort Belvoir, Virginia

North American Aviation, Inc.
Columbus, Ohio

Development and Proof Services
Automotive Engineering Laboratories
Aberdeen Proving Ground, Maryland

Air Force Flight Dynamics Laboratory
Research and Technology Division
Air Force Systems Command
Wright-Patterson Air Force Base
Ohio

Lyon Aircraft Services
2701 North Ontario Street
Burbank, California

U. S. Dept. of Commerce
Maritime Administration
Washington, D. C.

General Dynamics
Electric Boat Division
Groton, Connecticut

General Electric Co.
Valley Forge Space Technology Center
Philadelphia, Pennsylvania

Raytheon Company
Bedford, Massachusetts

American Trucking Association, Inc.
1424 Sixteenth St. N. W.
Washington, D. C.

Bureau of Ships
Navy Department
Washington, D. C.

United Technology Center
Sunnyvale, California

Lessells and Associates, Inc.
303 Bear Hill Road
Waltham, Massachusetts

U. S. Army Signal Corps
R & D Laboratory
Fort Monmouth, New Jersey

Crysler Corporation
Detroit, Michigan

Sandia Corporation
Albuquerque, New Mexico

National Academy of Sciences
National Research Council
Washington, D. C.

Association of American Railroad
Chicago, Illinois

Boeing Airplane Company
Commercial Airplane Division
Renton, Washington

National Highway Users Conference
National Press Building
Washington, D. C.