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THESIS

SHOCK QUALIFICATION OF COMBAT SYSTEMS EQUIPMENT USING TUNED MOUNTING FIXTURES ON THE U.S. NAVY MEDIUMWEIGHT SHOCK MACHINE

by

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June, 1992

Thesis Advisor: Y. S. Shin

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SHOCK QUALIFICATION OF COMBAT SYSTEMS EQUIPMENT USING TUNED MOUNTING FIXTURES ON THE U.S. NAVY MEDIUMWEIGHT SHOCK MACHINE

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requirements for the degree of

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ABSTRACT

Shipboard combat systems must be designed to withstand moderate to severe excitation induced by underwater explosion. Current specifications for combat systems shock qualifications are mandated in MIL-S-901D. Analyzing the differences and relationships between the predicted shock excitation, as derived from previous ship shock trials, and that shock excitation which is produced by the U.S. Navy Mediumweight Shock Machine required by MIL-S-901D, a proposed modification to the existing shock test procedure is presented which will better represent the shock phenomena experienced by combat systems exposed to underwater explosion.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

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I. INTRODUCTION

Shipboard combat systems equipments must be designed to withstand severe shock excitations induced by underwater explosion, either conventional or nuclear in origin. The underwater explosion delivers violent forces to the ship in the form of an incident shock wave pressure, gas bubble oscillations, cavitation closure pulses and various reflection wave effects. These complex shock induced forces propagate through the ship to various combat systems equipments, severely damaging them unless they are designed and tested to withstand such violent excitations. The ability of a naval vessel to carry out its mission, after being subjected to an underwater explosion, depends on the survivability of these equipments.

Current specifications for building ships and shipboard equipments contain the requirements for shock loading which must be met by the vendor of shipboard equipments. In general, all critical equipment is required to pass a series of shock tests which are outlined in Military Specification (MIL-S-901D), "Shock Tests, High Impact; Shipboard Machinery, Equipment and Systems, Requirements For." This document specifies the shock qualification test procedures which are required of all shipboard machinery, equipment and systems which must resist high impact mechanical shock. Three different shock test methods are outlined, these include shock testing by a lightweight shock machine, a mediumweight shock machine, or a floating platform barge. Selection of the shock test method depends on the item's size and weight. All these devices deliver high impact mechanical shock excitation to items affixed to them. The purpose of these tests, again, is to determine the suitability and survivability of machinery and equipment for use during and after exposure to severe shock excitation which may occur in wartime.

The response of combat systems equipments to underwater explosion is basically vibrational in nature. The equipment tends to vibrate at its fundamental natural frequency, or a low range of natural frequencies, when excited by the shock wave. The maximum amplitude of the vibration usually occurs after the shock wave passes the ship. The shock waveform is remarkably different at different levels within the ship, due largely to the ship's structural and material characteristics which cause the shock waveform to lengthen in duration and decrease in frequency as it propagates upward through the ship. In essence, the ship acts as a low pass mechanical filter which alters the characteristics of the propagating shock wave from one possessing high frequency components to one that contains relatively low frequency components, as noted by Scavuzzo, Lam and Hill (1988). Figure 1 depicts the described phenomena. Thus, the study of shock qualification for combat systems equipments, which are usually located in upper levels of the ship, is a vibration problem in which relatively low frequency equipment support foundation excitations are observed.

The U.S. Mediumweight Shock Testing Machine (MWSM), required by MIL-S-901D, is currently used for shock qualification of shipboard equipment ranging from about 250 to 6000 lbs. This machine and its application is the primary focus of this

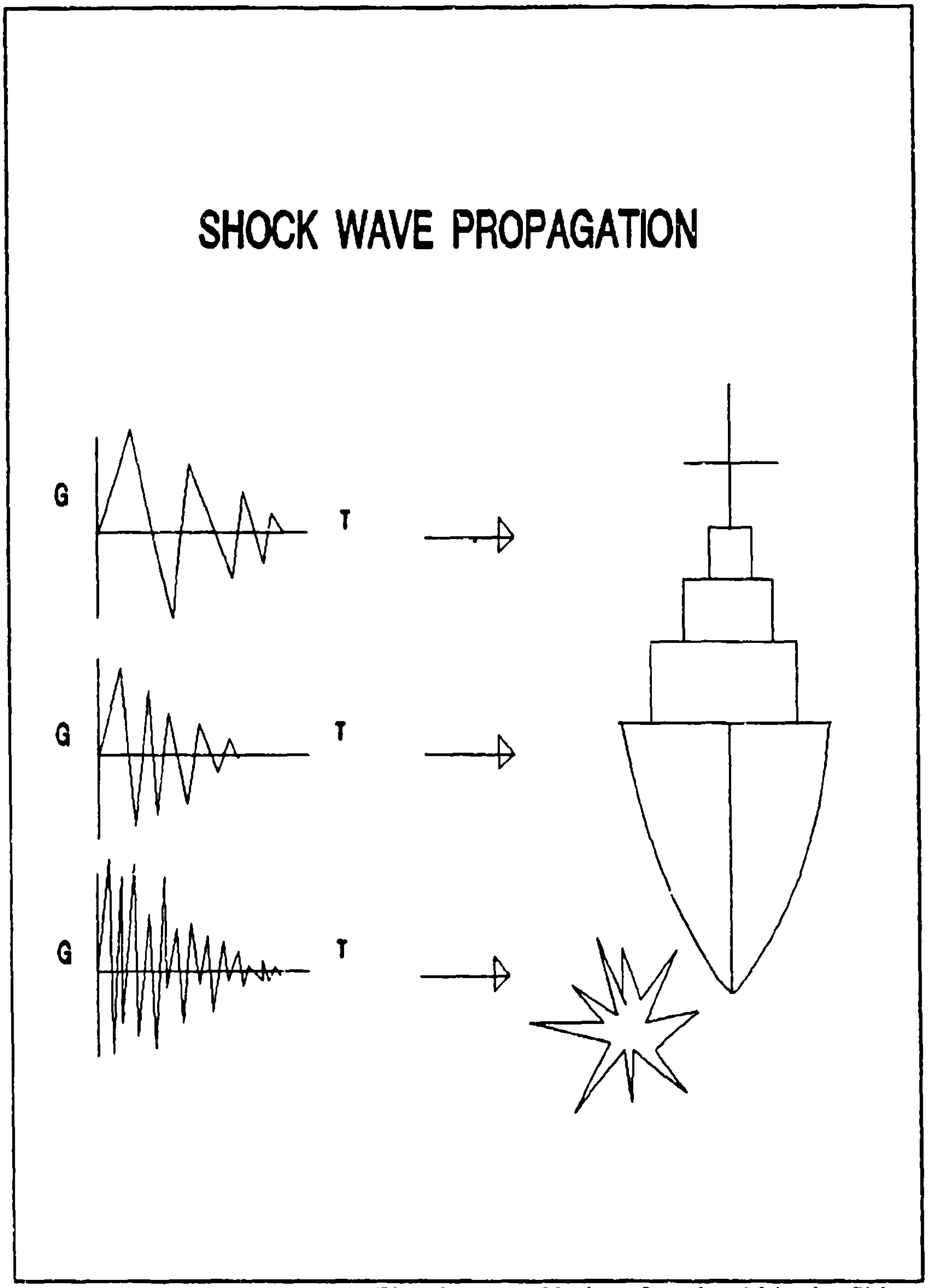

Figure 1. Acceleration Waveforms at Various Levels within the Ship.

study. The MWSM generates short-duration, high-impulse, high-frequency excitation which is transmitted directly to the mounted test object. This type of high frequency excitation waveform is significantly different from the actual waveforms that have been observed at various equipment locations during ship shock trials. The differences can be reduced by substituting a specially designed "tuned" test mounting fixture for the default mounting fixtures currently used to affix test items to the MWSM. A tuned mounting fixture, designed to respond at specific natural frequencies when excited by the MWSM, will provide a better simulation of the actual shock phenomena experienced by shipboard equipment.

This study examines the differences between the predicted shock waveform characteristics that can be observed in a ship shock trial, as reported for three representative pieces of equipment modeled in the Underwater Research Division of David Taylor Research Center (DTRC/UERD) DDG-51 Class Ship Pre-Shock Trial Analyses provided by Costanzo and Murray (1991), and those produced by the MWSM, as required by MIL-S-901D. The use of a "tuned" mounting fixture is proposed as a modification to existing shock test procedures. This proposal will afford a better representation of the actual shock phenomena experienced by surface shipboard combat systems equipments when they are qualified on the U.S. Navy Mediumweight Shock Machine.

II. BACKGROUND PRESENTATION

A. U.S. NAVY HIGH-IMPACT SHOCK MACHINE FOR MEDIUMWEIGHT EQUIPMENT (MWSM)

1. Development

The need for shipboard equipment shock qualification was recognized during World War II when substantial damage to shipboard equipment resulted not from direct hit by a shell, but rather by the blast effects of explosions which occurred within the vicinity of the ship. The shock wave traveled through the structures within the ship causing excessive vibration and permanent deformation, which rendered vital combat equipment useless.

In 1940, the first shock qualification test machine was developed by General Electric for the Navy. Called the Navy High-Impact Shock Machine for Lightweight Equipment, it was only capable of testing equipment which weighed up to 250 lbs. The need for a machine to test heavier equipment was recognized and, in 1942, Westinghouse Electric Corporation developed the first Shock Machine for Mediumweight Equipment. It was capable of testing equipment which ranged from 250 to about 4500 lbs. Today, it remains virtually the same, however, the rating has been extended to handle equipments weighing up to 6000 lbs and special equipment mounting fixtures are permitted, as outlined in MIL-S-901D.

2. Description

Perhaps the best description of the U.S. Navy High-Impact Shock Machine for Mediumweight Equipment (MWSM) can be found in the Naval Research Laboratory Report 7396 by Clements (1972). Paraphrasing his description, the MWSM is a hammer-anvil table apparatus, as noted in Figure 2. It consists of a hammer, weighing 3000 lbs., which swings through an arc of up to 270 degrees. The hammer height is adjusted from a position 180 degrees away from the hammer impact and the total weight on the anvil table is used as an argument to determine this height. The hammer strikes the 4500 lb. anvil table from below and imparts an upward, uniaxial acceleration and velocity to it. The anvil table has a 60 by 60 inch mounting surface upon which the test items are affixed. The entire anvii table apparatus is bolted to the machine's foundation. These bolts permit the anvil table apparatus to travel up to 3 inches vertically after hammer impact. The table travel distance can be decreased by using pneumatic jacks to vertically reposition the anvil table. The machine is embedded in a massive concrete block resting on heavy coil springs which isolate it from its surroundings. The impacting surfaces of the hammer and anvil are fitted with spherical hardened-steel impact plates which render the collision elastic.

MIL-S-901D mandates that test items will be mounted to the anvil table by a fixture in a manner characteristic of its designed shipboard orientation, along with any anticipated supporting structures which may mitigate the shock experience. The equipment and mounting fixture configuration cannot exceed 7400 lbs. Steel channels are used to construct the mounting fixture for the test item. The number and type of channels to be used are specified in MIL-S-901D. Both equipment weight and distance between anvil table mounting holes determine the number of channels to be used for the mounting fixture. The specified configuration tends to keep the natural frequency of the test equipment-mounting fixture-anvil table system between 55 and 72 Hertz. This, as noted by Clements (1972), was not by design, but rather an effort to keep the maximum stress in the channels to less than 35,000 psi in a static acceleration field of 50 g's.

3. MWSM Shock Waveform

The MWSM may be modeled quite simply as a mass-spring-damper system subjected to base excitation, as noted in Figure 3, and presented in Clements (1972). The system base excitation is provided by the hammer and anvil elastic impact which results in vertical motion of the anvil table. The equipment mounting fixture's stiffness properties and the loss of energy, due to friction at bolted joints and imperfections in material design, substantiate this simplified model. More elaborate models may be required to describe and analyze intricate test structures, but meaningful results can be obtained with this model.

The mechanical shock waveform afforded by the MWSM can be described by a velocity or acceleration waveform generated by the hammer and anvil impact, as noted in Clements (1972). Paraphrasing the description, the hammer and anvil impact produces a well defined half-sinusoidal acceleration pulse having an approximate duration of one millisecond. This pulse imparts an upward velocity and

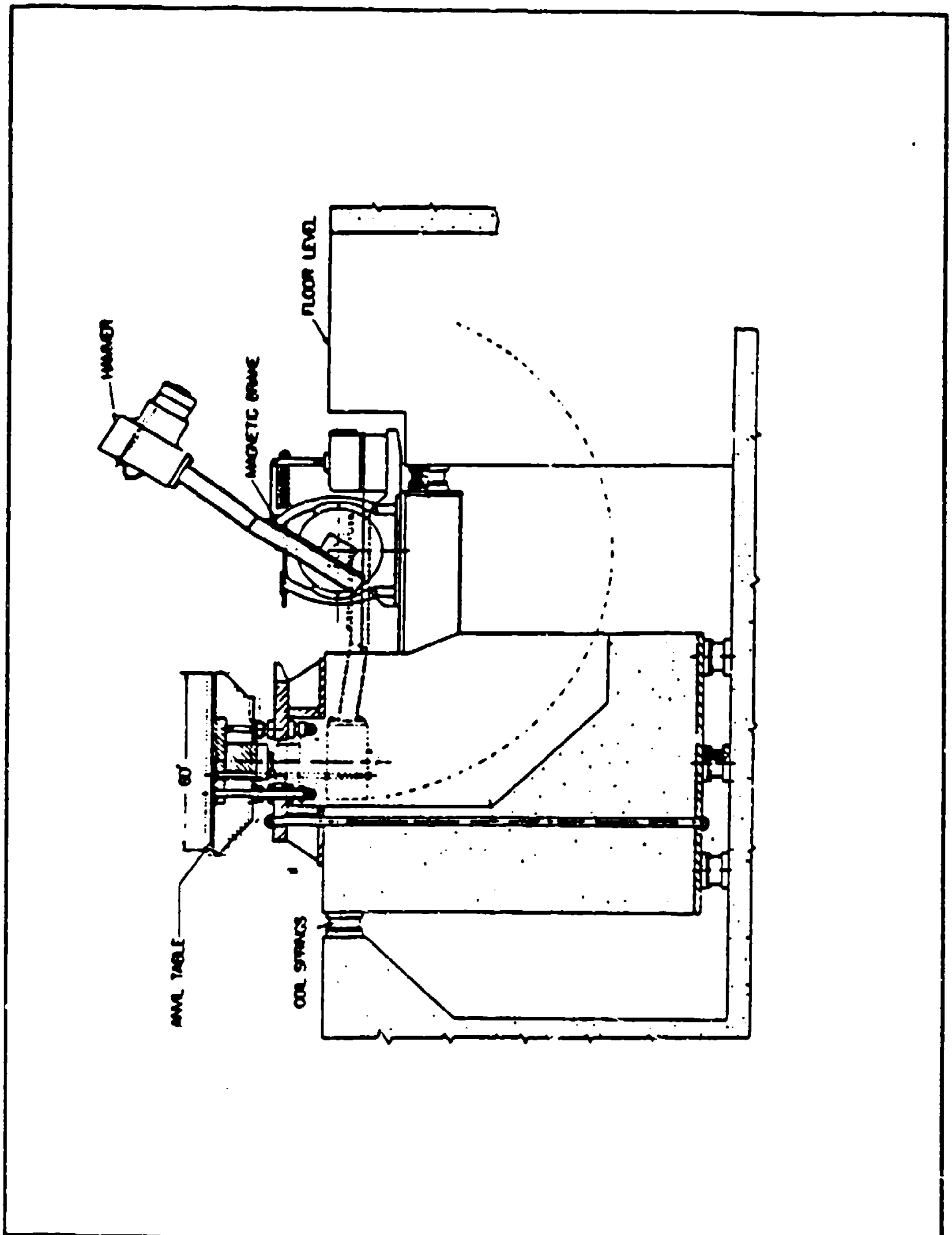

Figure 2. The Navy High-Impact Shock Machine for Mediumweight Equipment (MWSM). The Dotted Line Shows Hammer Path. Courtesy Clements (1972).

acceleration to the anvil table which continues until the table travel distance is achieved at the stops, some two to four milliseconds after impact. At this event, a new set of transients occur which may interfere with the motion established. This "table reversal" is followed by another transient which occurs when the anvil table comes to rest. In addition, the half-sinusoidal acceleration pulse excites a 750 Hertz longitudinal mode of the anvil table. This appears as a damped vibration that persists for about five cycles. Together, these events produce a very high-energy, high-frequency complex waveform with peak accelerations ranging from 220g to 580g, depending on the hammer height.

Figure 4 depicts the peak anvil table accelerations versus hammer height and hammer impact velocity. The relationship between peak anvil table accelerations and associated hammer height and hammer impact velocity is linear, reinforcing the elastic impact argument. The "table reversal" acceleration pulse, occurring sometime later, will be somewhat smaller than the initial peak impact acceleration largely due to frictional factors. It follows that the transient acceleration pulse arising when the anvil table comes to rest will be even smaller than the "table reversal" transient acceleration pulse.

Thus, the major features of the measured MWSM anvil table acceleration waveform can be described as a series of three half-sine acceleration pulses. The first, due to initial hammer-anvil impact, with a duration of one millisecond, followed by a second, oppositely directed, smaller pulse occurring sometime later at table reversal, then, lastly, an even smaller pulse when the anvil table comes to rest. The

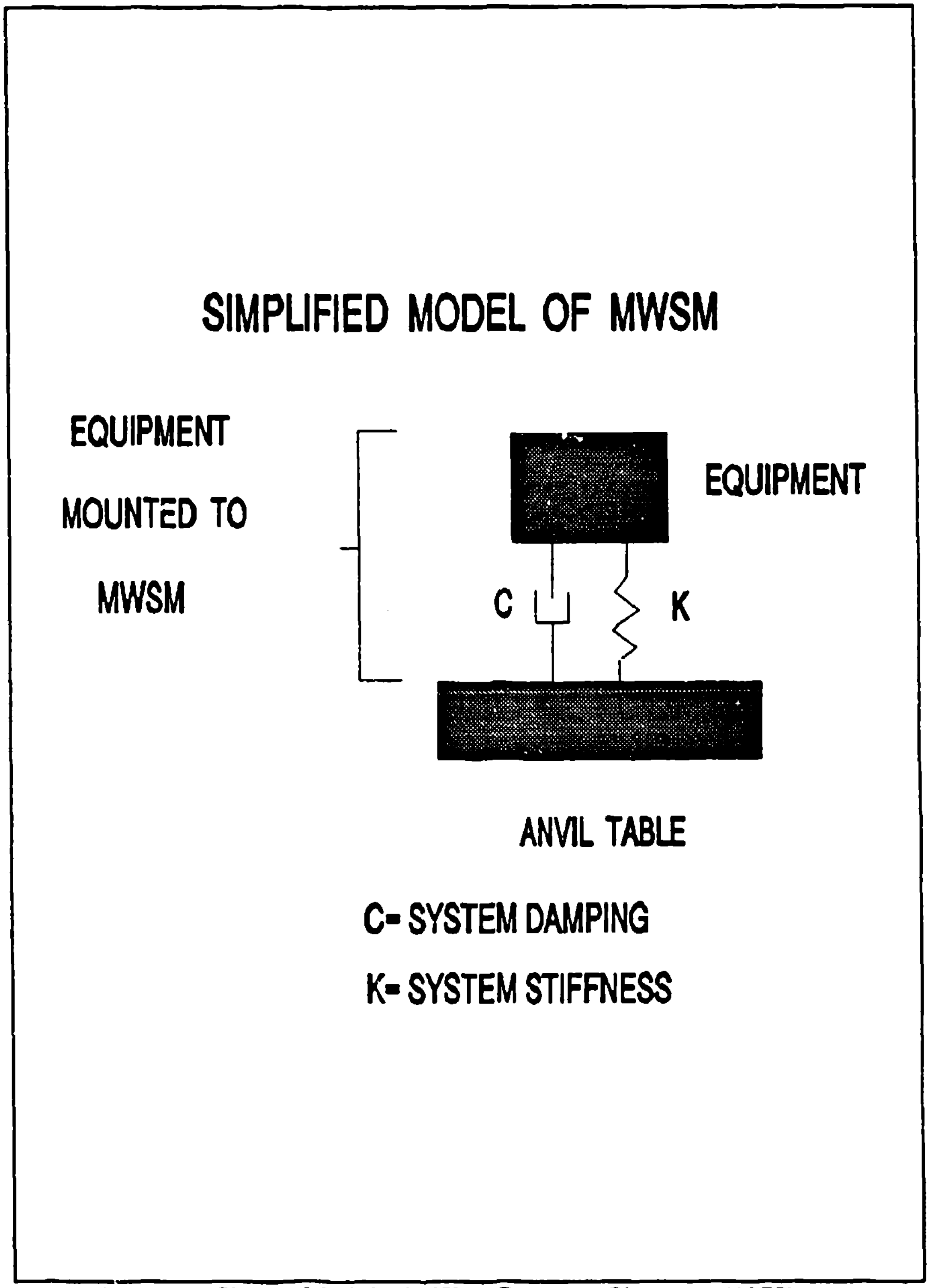

Figure 3. Simple Model MWSM. Courtesy Clements (1972).

latter two smaller pulses depend on the adjustable table travel distance. For analysis purpose, only the initial and, by consequence, severest, half-sine anvil table peak acceleration pulse will be simulated, for the complexities of structural damping influences and the effects table travel distances preclude the accurate simulation of the latter two smaller acceleration pulses. This first acceleration pulse will be used as the base excitation in analyzing the tuned fixture model response.

B. DTRC/UERD DDG-51 CLASS DECK HOUSE PRE-SHOT TRIAL ANALYSES

A transient shock analysis of the DDG-51 Class Deck House was conducted by the DTRC/UERD in preparation for the forthcoming DDG-51 Class shock trial. The preliminary report, by Costanzo and Murray (1991), was obtained along with the predicted shock excitation histories and analyses for various weight combat equipments located on the 0-3 level of the DDG-51 Class Ship. This information was crucial in assessing the relationships between the shock phenomena experienced by surface shipboard equipment exposed to underwater explosion and that shock phenomena associated with the MWSM. Their findings are summarized below.

1. Finite Element Model for DDG-51 Class Forward Deck House

A finite element model of the DDG-51 Class Forward Deck House was developed which included all major structural members and supporting equipments. The model analysis was performed using COSMIC/NASTRAN code. Figure 5 presents a depiction of the finite element model of the Forward Deck House. The model extends vertically from the 0-1 Level to the Sea Director Level.

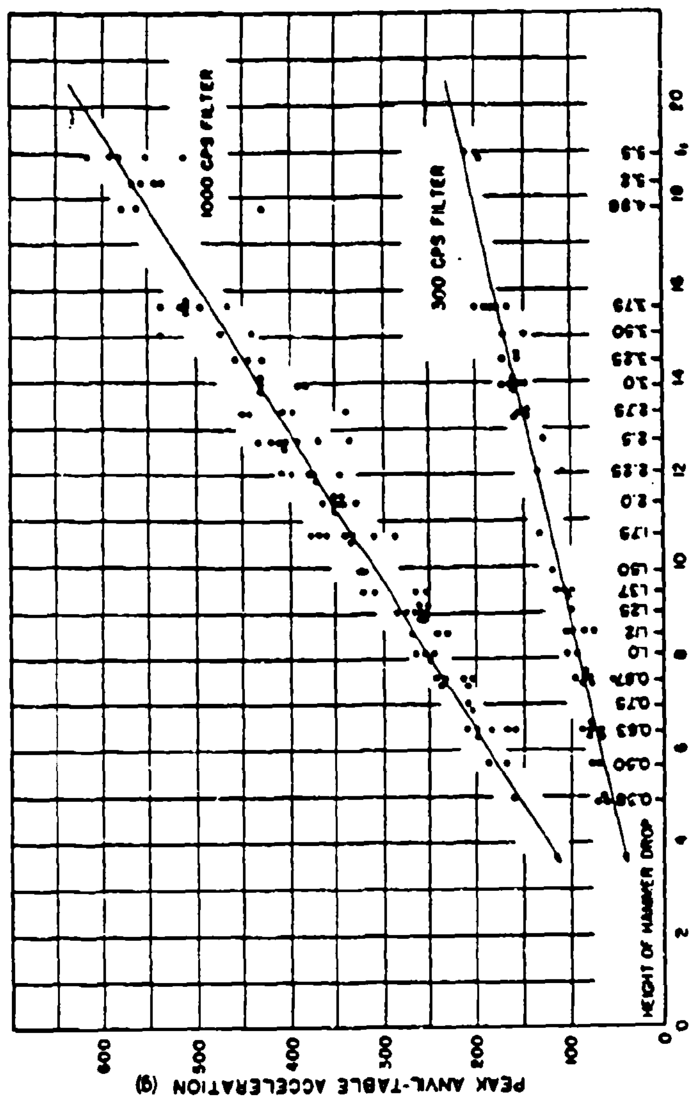

Figure 4. MWSM Peak Anvil Table Accelerations Versus Hammer Height. Courtesy Clements (1972).

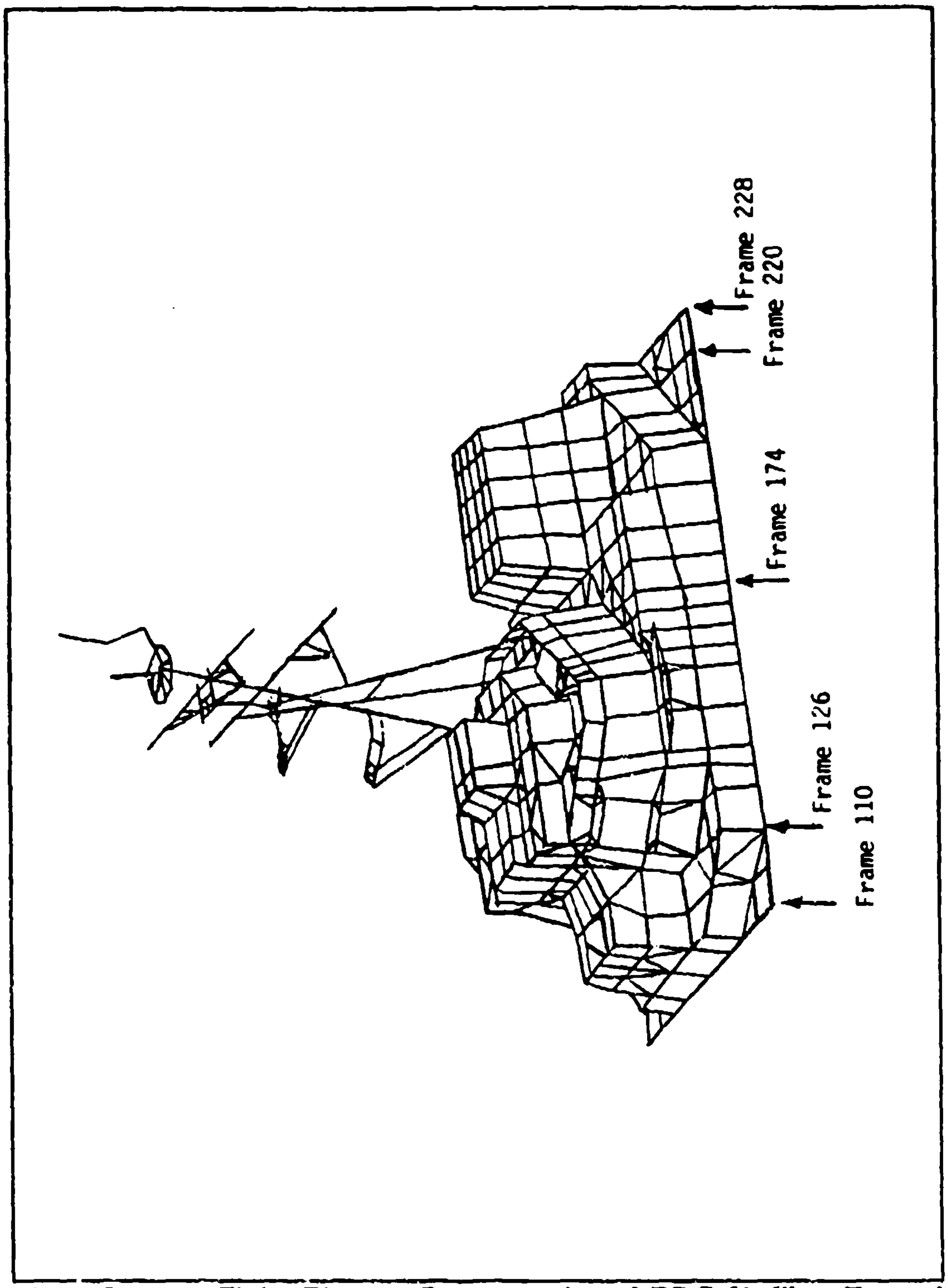

Figure 5. Finite Element Representation of DDG-51 Class Forward Deck House. Courtesy Costanzo and Murray (1991).

The primary combat systems equipments present in the deck house were modeled by Costanzo and Murray (1991). The equipments were represented as lumped, rigid masses in the model. These masses were distributed to the various nodes present in the deck house model which represent the corresponding shipboard equipment locations in the actual deck house. In cases where the equipment center of mass was known, an appropriate offset was employed in the modeling procedure to position the equipment mass at the proper location above the respective deck or foundation attachments. Of particular interest were three combat systems equipments ranging from 325 to 4600 lbs, all located on the 0-3 Level. The three included a Radar Receiver Transmitter (RT-1293/SPS-67) weighing 325 lbs, a Beam Programmer (MX-10873\SPY-1D) weighing 1000 lbs and a Radio Frequency Amplifier (AM-7159\SPY-1B) weighing 4600 lbs. Their range of weights would be useful in characterizing the behavior of the MWSM for various weight class equipments, low, medium, and high, when a tuned mounting fixture is applied.

2. Analysis of DDG-51 Class Forward Deck House Model

The finite element model was shock analyzed for maximum shock trial severity, shot four in a series of four underwater explosion shots. Transient shock response calculations for all nodes were performed using shock excitations to frames 126, 174, and 220 at the 0-1 Level of the model, as noted in Figure 5. These three frames are the major supporting bulkheads of the DDG-51 Class Forward Deck House. The shock excitations were obtained from a full ship's hull girder model of the DDG-51 class and actual shock trial data from previous cruiser shock trials. For each significant deck house equipment, the dynamic responses at the equipment foundation locations were presented in the form of acceleration-time histories for the three principal directions, vertical, athwartships and fore/aft. For these directions, the transient shock responses for all three equipments were computed out to 70 milliseconds. This was considered enough time for the ship to reach its maximum vertical displacement due to initial shock wave effect.

Of primary importance to this thesis is an understanding of the Shock Spectra that results from these predicted transient acceleration excitations. The Shock Spectra defines the absolute maximum response envelope, over a wide range of system natural frequencies, of an undamped single degree of freedom mass-spring system subjected to a specific excitement. For a given excitement, the resultant Shock Spectra will reveal peak resonance responses which are of vital concern in the design and shock testing of equipment modeled as such a mass-spring system expo ed to that excitement.

Costanzo and Murray (1991) analyzed shock phenomena in three principal orientations: athwartships, fore/aft and vertical. Of particular interest is the vertical orientation shock analysis since this is the most severe type of shock experienced by surface vessels exposed to underwater explosion, as revealed in the DTRC/UERD Shock Spectra comparisons presented in Appendix A. Results of their vertical transient shock analysis, performed for the three equipments mentioned earlier, are shown in Figures 6 through 11. These are the predicted acceleration waveform excitements for each equipment's foundation and the resultant Shock Spectras. It should be noted that structural damping was omitted in their analysis for the following two reasons. One, damping is not constant throughout such a complex structure as the DDG-51 Class Forward Deck House and, two, the omission of structural damping generally results in conservative computed response levels.

As noted in Figures 6 through 11, the predicted acceleration waveforms and associated Shock Spectras for the selected equipments, all located in the same compartment on the 0-3 Level, are dramatically different. Figures 6 through 8 shows the predicted equipment's foundation acceleration excitement due to the shock wave. Each acceleration waveform is significantly different, possessing different frequency components and amplitudes. Placing each of these equipment on the MWSM with the default mounting fixture outlined in MIL-S-901D and then exciting it with an acceleration pulse will not simulate the same shock phenomena depicted in those Figures. Thus, the need to apply a special MWSM mounting fixture, "tuned" to emulate the frequencies of interest, is necessary in order to provide the same characteristic shock phenomena observed in the field.

The identification of the necessary characteristics of a tuned mounting fixture is the focus of this thesis. To this end, the three equipments studied provide an excellent representation of the weight ranges, low, medium and high, of the equipments tested on the MWSM. The findings presented in this thesis will enable the design, construction and implementation of tuned mounting fixtures on the MWSM.

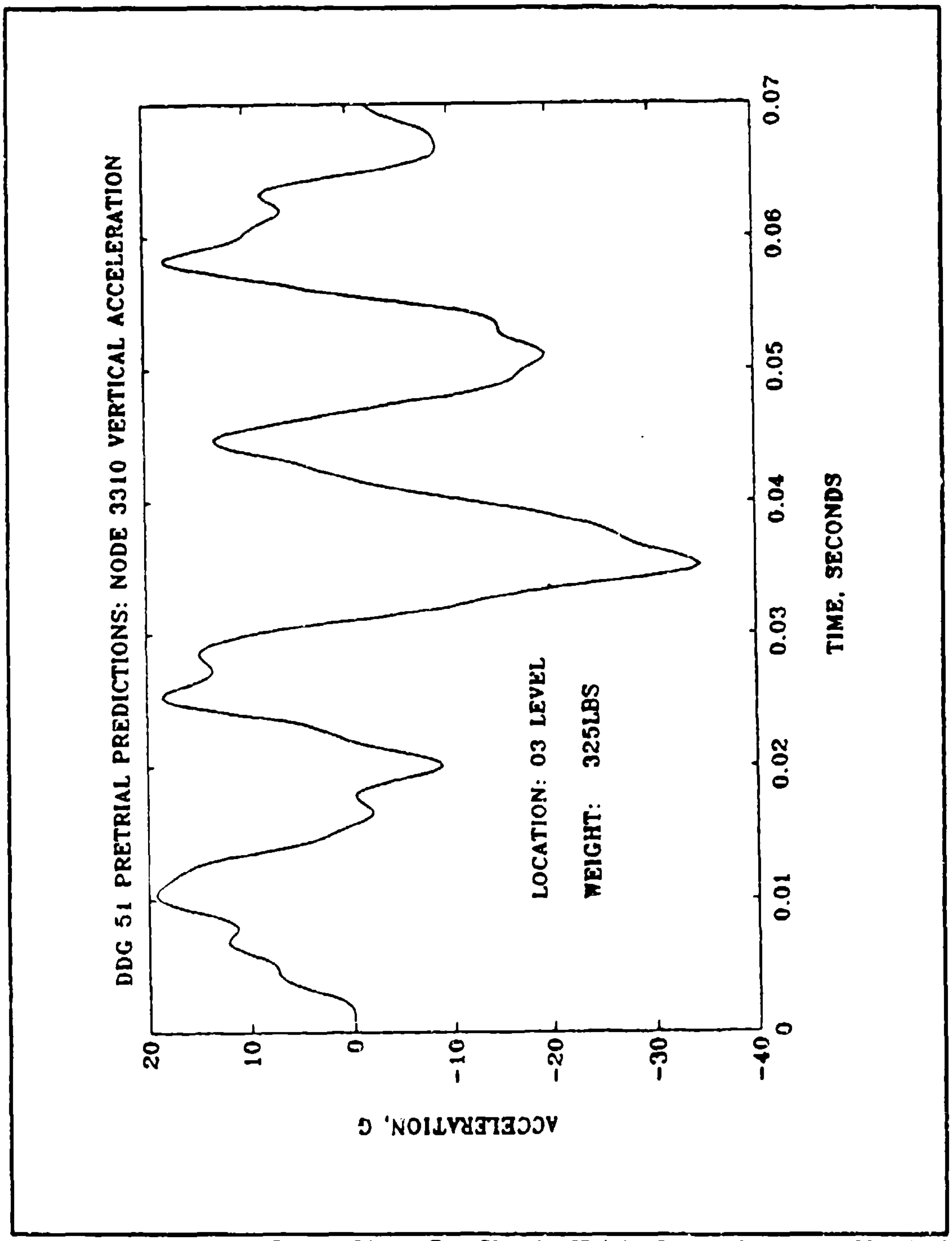

Figure 6. DDG-51 Class Pre-Shock Trial Predictions of Vertical Acceleration Time History for Radar Receiver/ Transmitter, RT-1293/SPS-67. Equipment Weight 325 lbs. Courtesy of Costanzo and Murray (1991).

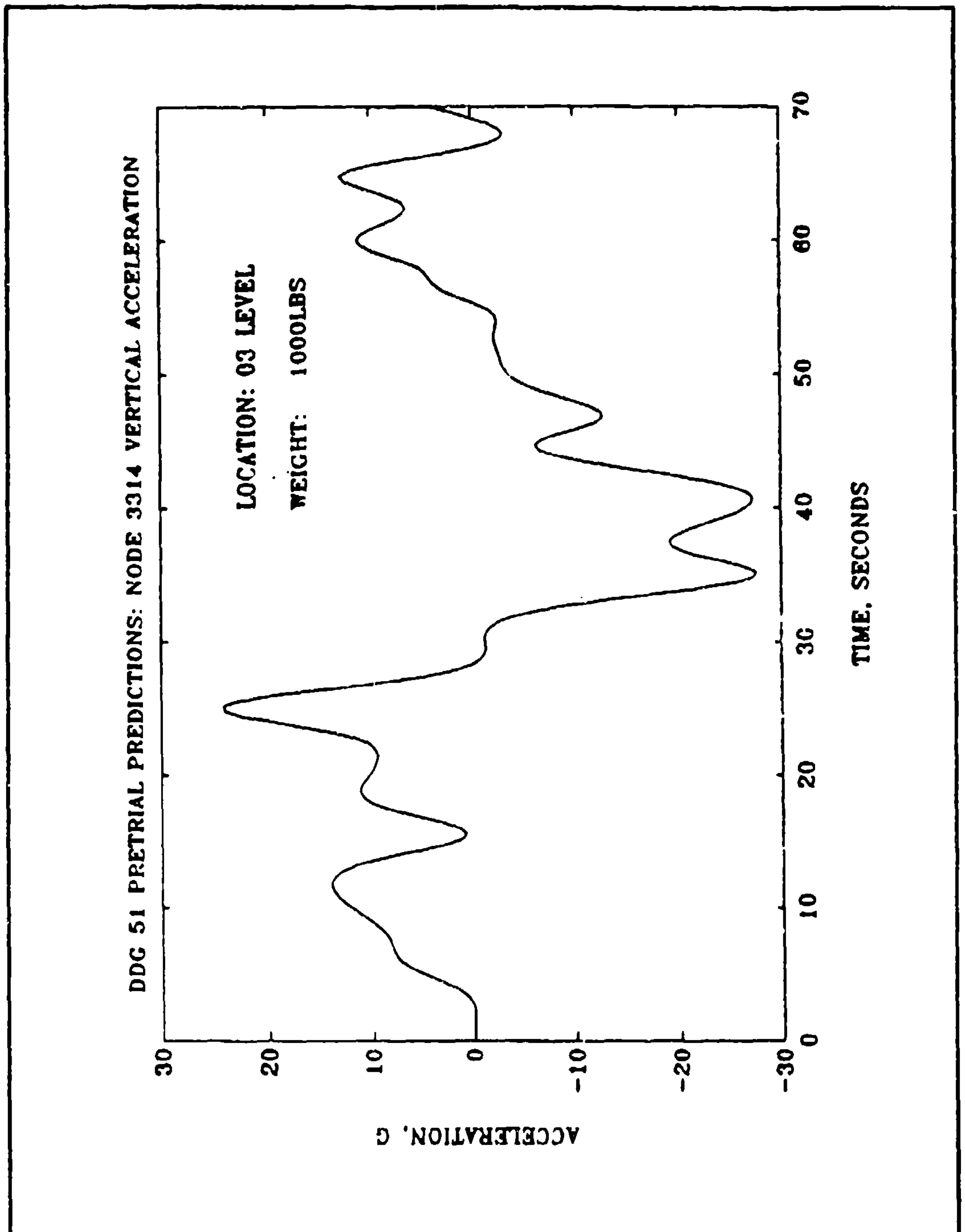

Figure 7. DDG-51 Class Pre-Shock Trial Prediction of Vertical Acceleration Time History for Beam Programmer, MX-10873/SPY-1D. Equipment Weight 1000 lbs. Courtesy of Costanzo and Murray (1991).

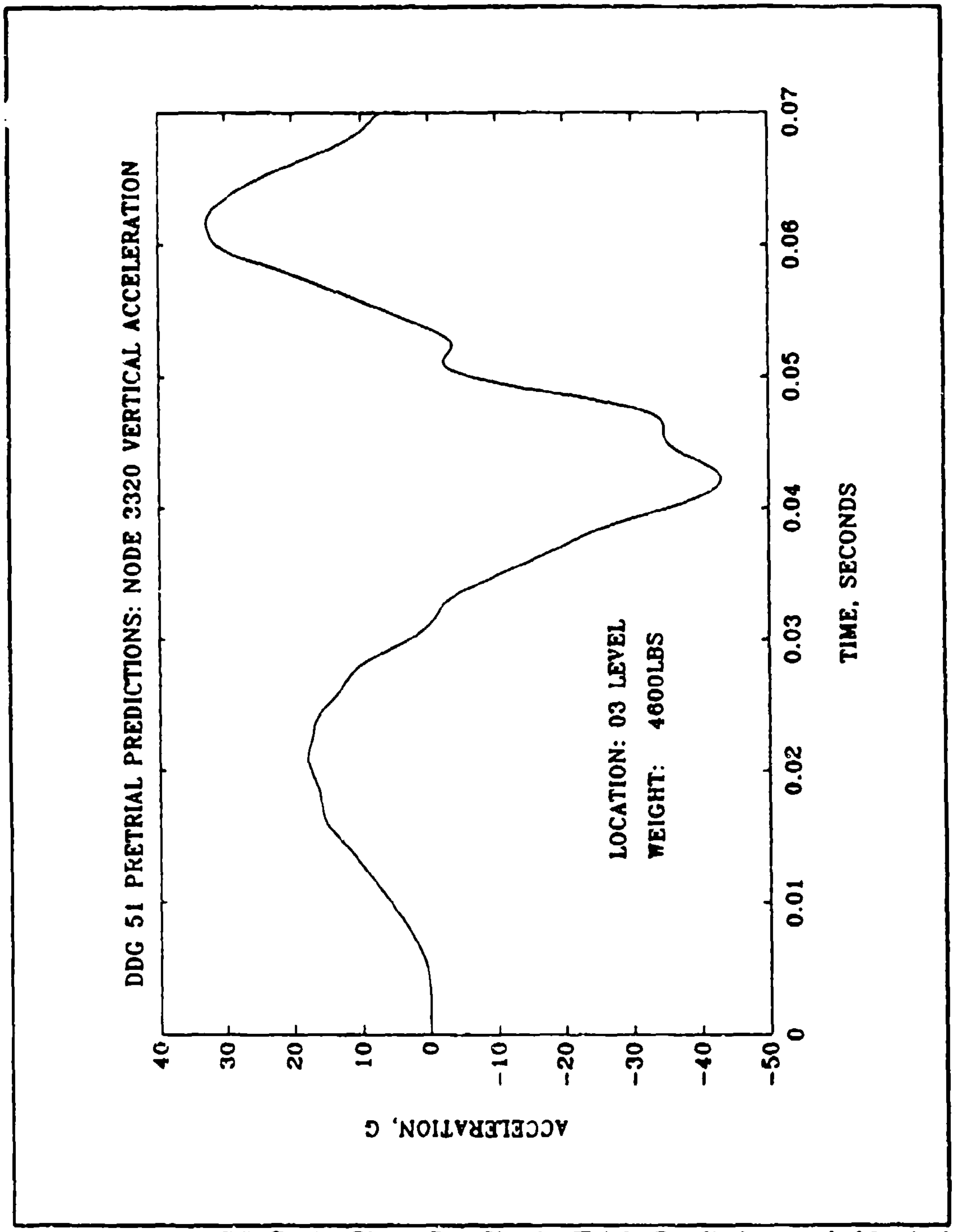

Figure 8. DDG-51 Class Pre-Shock Trial Predictions of Vertical Acceleration Time History for Radio Frequency Amplifier, AM-7158/SPY-1B. Equipment Weight 4600 lbs. Courtesy of Costanzo and Murray (1991).

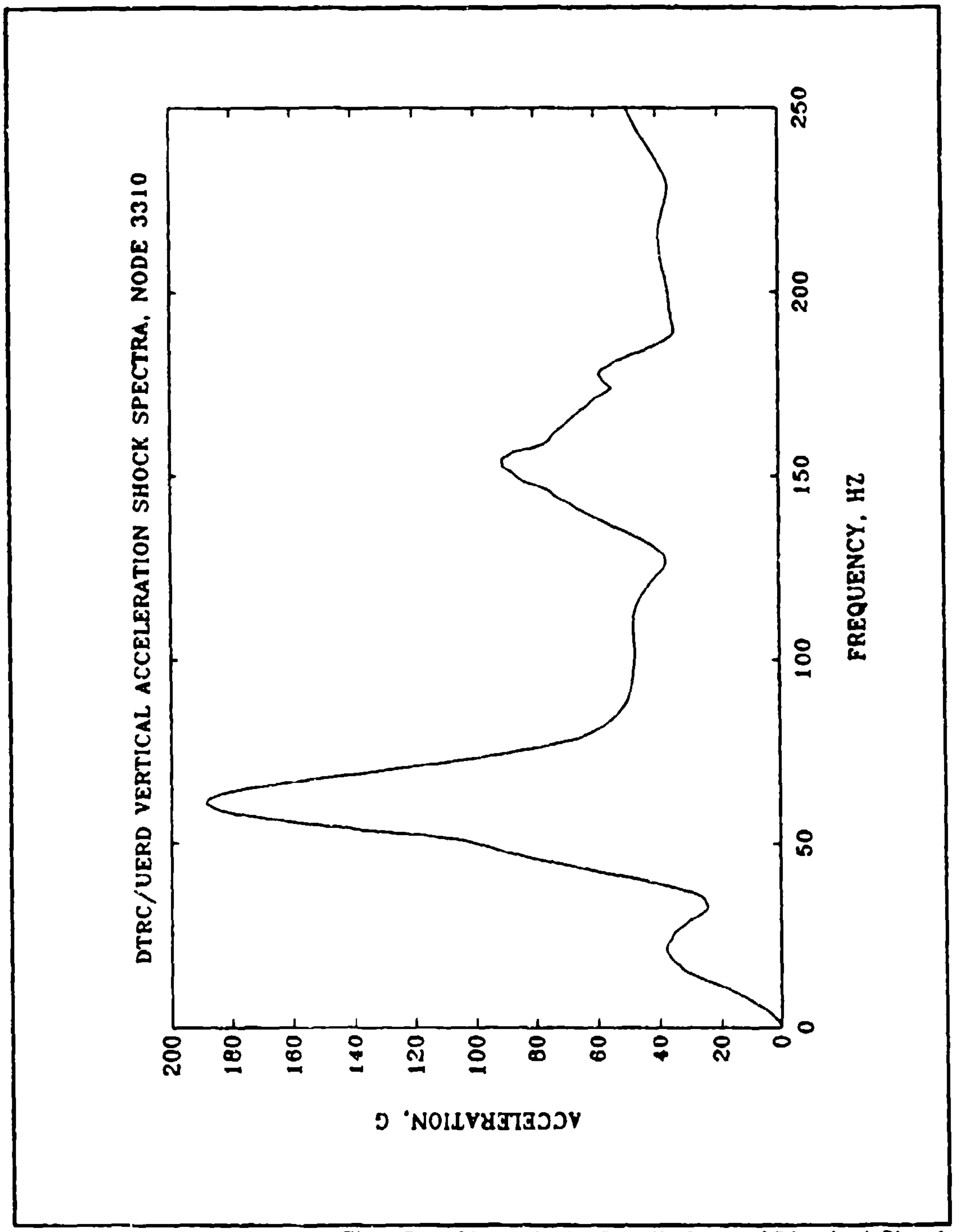

Figure 9. DDG-51 Class Pre-Shock Trial Predictions of Vertical Shock Spectra for Radar Receiver/Transmitter, RT-1293/SPS-67. Equipment Weight 325 lbs. Courtesy of Costanzo and Murray (1991).

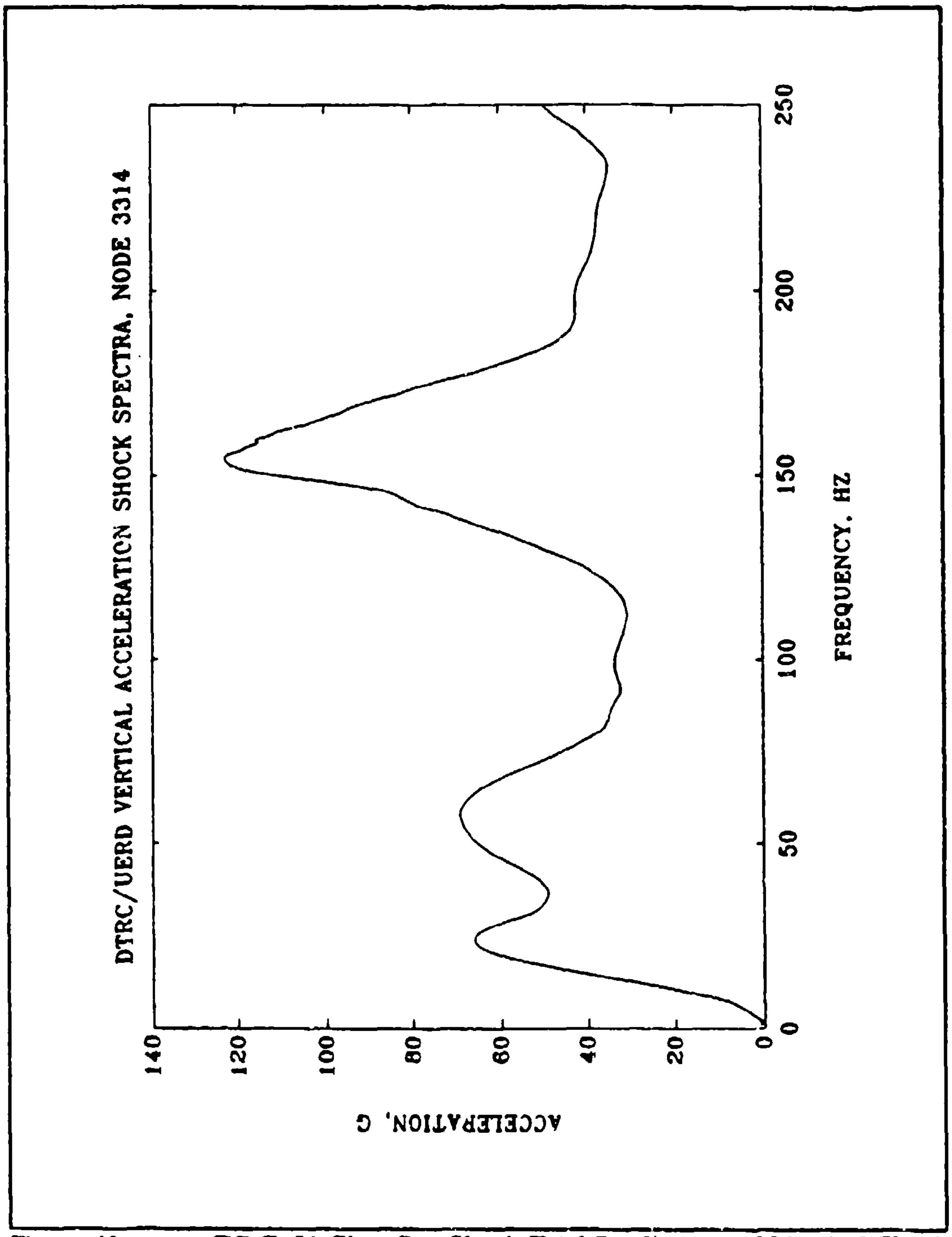

Figure 10. DDG-51 Class Pre-Shock Trial Predictions of Vertical Shock Spectra for Beam Programmer, MX-10873/SPY-1D. Equipment Weight 1000 lbs. Courtesy of Costanzo and Murray (1991).

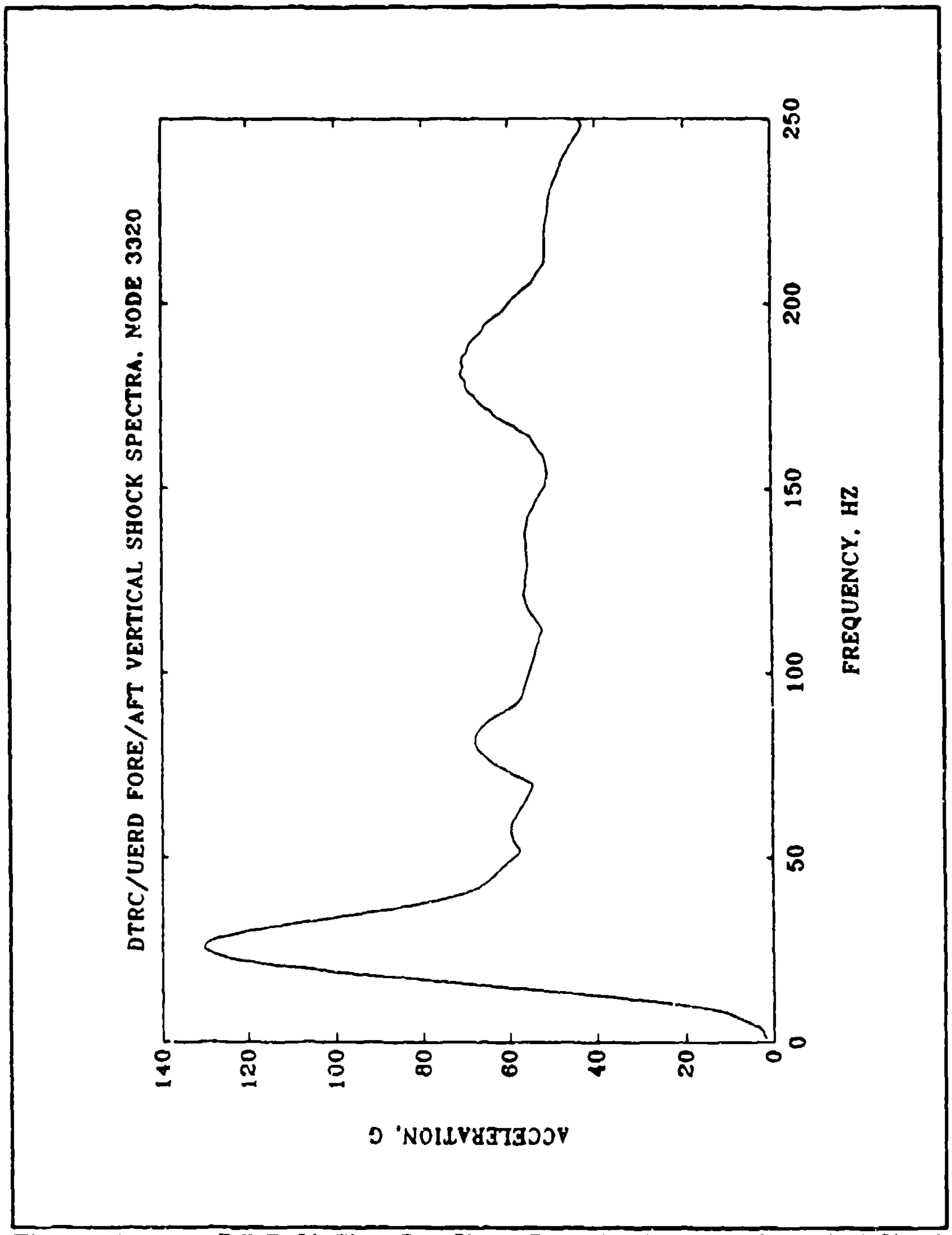

Figure 11. DDG-51 Class Pre-Shock Trial Predictions of Vertical Shock Spectra for Radio Frequency Amplifier, AM-7159/SPY-1B. Equipment Weight 4600 lbs. Courtesy of Costanzo and Murray (1991).

III. SINGLE DEGREE OF FREEDOM TUNED MOUNTING FIXTURE

A. DTRC/UERD PRE-SHOT TRIAL SHOCK SPECTRA ANALYSIS

If the item to be tested is mounted to the MWSM by a special fixture, which has a designed fundamental frequency or frequencies when the item is affixed to it, it will experience the fixture's vertical response to the MWSM excitation as its shock excitation, as presented by Chalmers and Shaw (1989). As the MWSM is an uniaxial machine, the tuned mounting fixture must be designed to provide a desired frequency response along a single direction, vertical for this study. The question then arises as to what fundamental frequency or frequencies to select. The answer can be revealed by analyzing the item's predicted Pre-Shot Trial Shock Spectra.

The Vertical Orientation Shock Spectra, for each equipment provided by DTRC/UERD, is presented in Figure 12. Each spectra shows a variety of peak responses at discrete frequencies. Beginning with the low weight range equipment Shock Spectra, a Radar Receiver/Transmitter weighing 325 lbs, analysis shows two dominant peaks, one at about 60 Hertz and another at about 155 Hertz. The peak ratio is about 2:1. Clearly, this equipment's predicted foundation acceleration excitement possesses two dominant waveform components. Thus, a two degree of freedom (DOF) uniaxial mounting fixture will be necessary in order to simulate the dominant acceleration waveform characteristics present, as revealed within this Shock Spectra.

Next, the medium weight range equipment Shock Spectra, a Beam Programmer weighing 1000 lbs, shows three well defined dominant peaks. One at about 23 Hertz and another at about 60 Hertz, both with about the same magnitude. An absolute dominant peak, by a factor of two, occurs at about 155 Hertz. A two DOF uniaxial tuned mounting fixture would be required as a minimum in order to simulate the two most important frequency characteristics depicted within this Shock Spectra.

Finally, the high weight range equipment Shock Spectra, a Radio Frequency Amplifier weighing 4600 lbs, shows a single dominant peak at about 23 Hertz. A single DOF tuned mounting fixture would simulate the frequency and acceleration waveform characteristics found within this Shock Spectra.

B. SINGLE DEGREE OF FREEDOM TUNED MOUNTING FIXTURE MODEL

Advancing the principle proposed by Chalmers and Shaw (1989), the item to be tested is affixed to the MWSM by a single DOF mounting fixture, which has a designed fundamental frequency when the item is affixed to it. The test item will experience the fixture's vertical response to the MWSM excitation as its shock excitation. Figure 13 depicts the described concept.

1. Single Degree of Freedom Tuned Mounting Fixture Mathematical Model

Figure 13 depicts a single DOF mass-damper-spring system subjected to foundation excitement, the model representing the single DOF tuned mounting

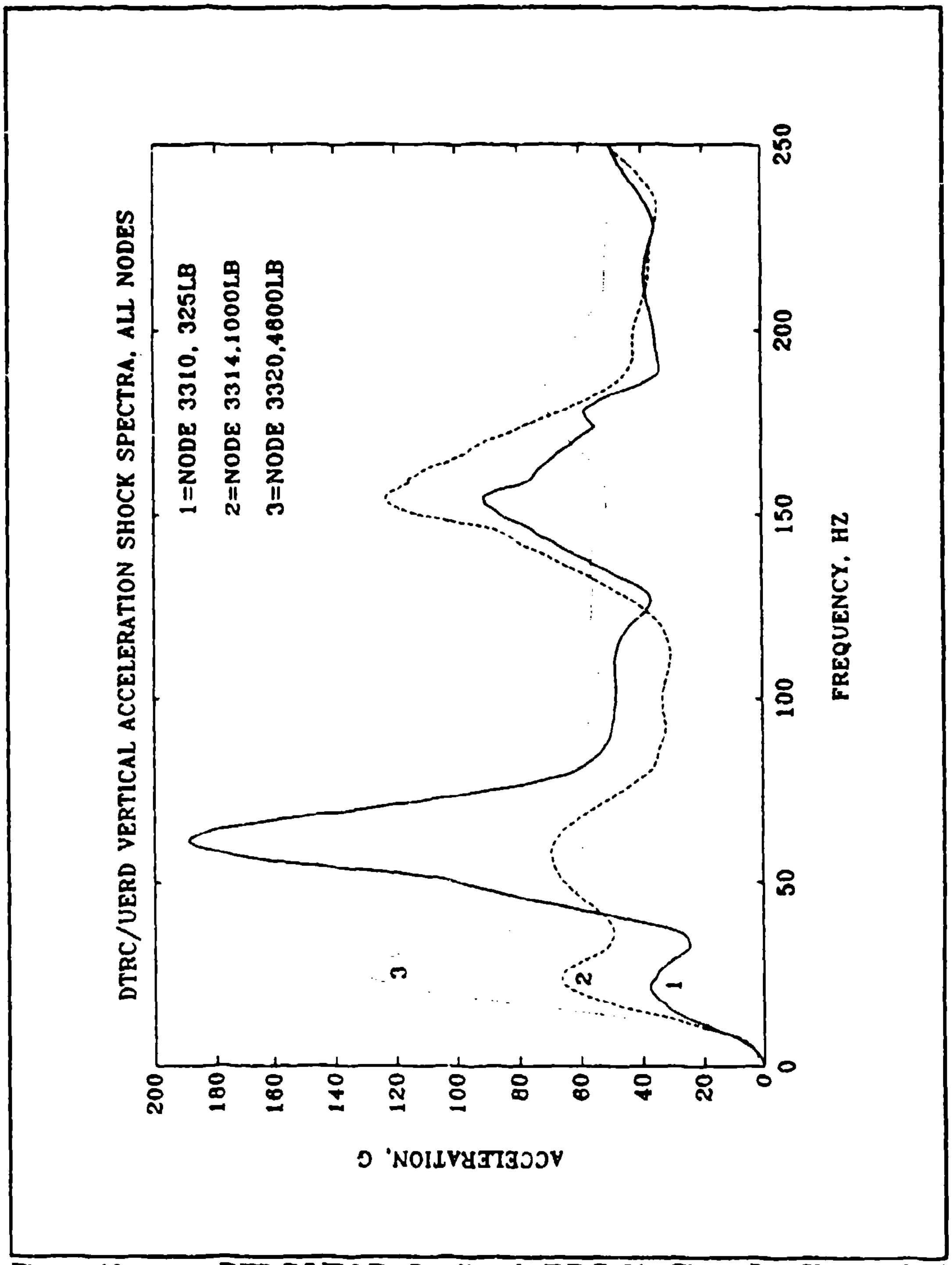

Figure 12. DTRC/UERD Predicted DDG-51 Class Pre-Shock Trial Vertical Orientation Shock Spectra. Courtesy of Costanzo and Murray (1991).

fixture. The equation of motion for this system can be expressed in terms of relative motion coordinates. This will facilitate the solution to the problem. The mass in this model represents the combined equipment and mounting fixture mass. Let the absolute motion of the mass be expressed by the x coordinate and the foundation motion by the z coordinate. Then equations 1 through 3:

$$y=x-z \tag{1}$$

$$\dot{y}=\dot{x}-\dot{z} \tag{2}$$

$$\ddot{y}=\ddot{x}-\ddot{z} \tag{3}$$

are the relative coordinate transformations for displacement, velocity and acceleration, respectively.

The system's natural frequency is a function of system mass, m, and stiffness, k, and can be expressed either in radians per second or in Hertz, cycles per second. Equations 4 and 5 note those respective relationships.

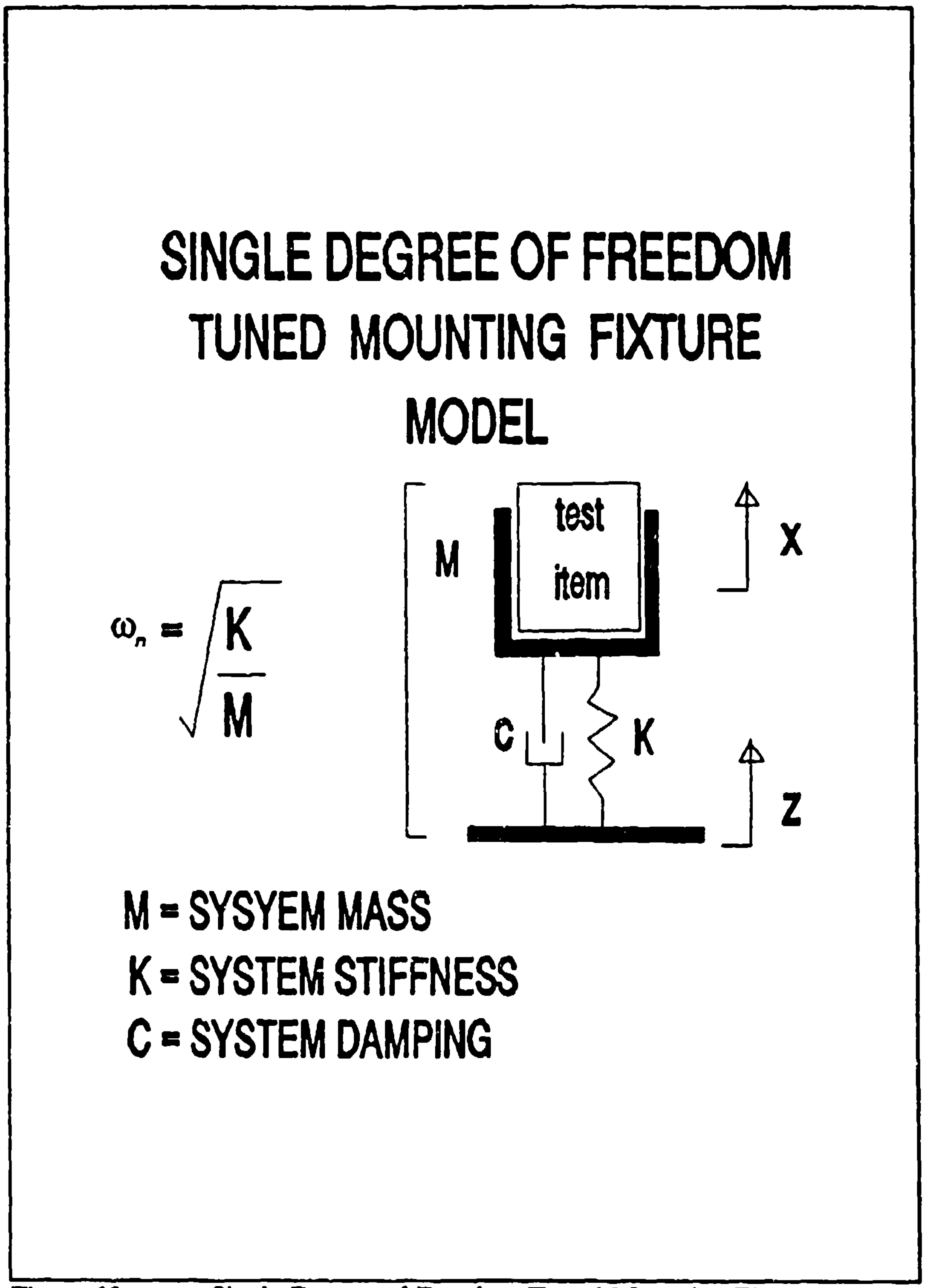

Figure 13. Single Degree of Freedom Tuned Mounting Fixture.

$$\omega_n=\sqrt{\frac{k}{m}} \tag{4}$$

$$f_n=\frac{1}{2\pi}\sqrt{\frac{k}{m}} \tag{5}$$

The equation of motion for this system can be written as follows:

$$\ddot{y}+2\xi\omega_n\dot{y}+\omega_n^2 y=-\ddot{z} \tag{6}$$

where ξ is the damping factor and must be estimated.

Equation 6 was numerically integrated using an unconditionally stable numerical integration scheme outlined by Craig (1981). The Fortran code listing is presented in Appendix B.

Once the relative displacements, velocities, and accelerations are known, equations 1 through 3 can be solved for the absolute quantities:

$$x=y+z \tag{7}$$

$$\dot{x}=\dot{y}+\dot{z} \tag{8}$$

$$\ddot{x}=\ddot{y}+\ddot{z} \tag{9}$$

where equation 9 presents the single degree of freedom tuned mounting fixture's response acceleration, which is the equipment's foundation excitement. This acceleration excitement is used for the development of the Shock Spectra. The Fortran code listing for the Shock Spectra formulation is listed in Appendix C.

2. Single Degree Of Freedom Tuned Mounting Fixture Modeled Application

The high weight range equipment, a Radio Frequency Amplifier weighing about 4600 lbs, was an excellent candidate for the single DOF tuned mounting fixture application. The DTRC/UERD Shock Spectra revealed a peak at about 23 Hertz, thus, the mounting fixture must possess a fundamental frequency of 23 Hertz when the equipment is affixed to it. This can be easily obtained by first, fixing the mass of the system, then designing the system stiffness, noting the relationships presented in equations 4 and 5.

As stated earlier, the major feature of the MWSM excitation is a half-sine acceleration pulse of approximately one millisecond duration. Selection of MWSM hammer height and consequent peak anvil table acceleration is dependent on total anvil table top weight, as required by MIL-S-901D. For the single DOF tuned mounting fixture application analysis, a peak acceleration of 425g's was selected. This corresponds to a hammer height of three feet, which is nominal for the first series of hammer blows corresponding to total anvil table top weights in the range of 4600 to 7400 lbs. Figure 14 depicts the simulated MWSM pulse used for the single DOF tuned fixture application analysis.

A sensitivity study in damping was conducted to correlate and validate the single DOF tuned mounting fixture's modeled response with respect to actual MWSM calibration test data. Examination of MWSM calibration data, as compiled by Costanzo and Clements (1988), showed that the test weight acceleration responses appeared to dissipate within .2 to .4 seconds after initial MWSM excitement. Thus, the selection of the damping factor in equation 6 could be estimated based on that information. Figures 15 through 19 show the damped acceleration response study for the single DOF tuned mounting fixture subjected to the MWSM acceleration pulse excitation.

Figure 18, with a damping factor equal to .08, presents the best decaying characteristics consistent with the MWSM test calibration data. The single DOF tuned mounting fixture, damped at this factor, was further processed to yield the acceleration waveform and comparative results presented in Figures 20 to 24. Figure 20, the Fourier Transform of the acceleration response, shows a well defined peak at 23 Hertz, as expected. Figure 21 shows the comparison between the DTRC/UERD predicted acceleration waveform excitement for the Radio Frequency Amplifier and that produced in the first 70 milliseconds by the single DOF tuned mounting fixture. There is close agreement in shape and magnitude, as expected.

Figure 22 is the resultant Shock Spectra, which is typical for a decaying sinusoidal acceleration excitement. The modeled single DOF tuned mounting fixture Shock Spectra is compared with the predicted DTRC/UERD Shock Spectra in Figure 23. There is excellent agreement in spectral shape. The magnitude difference

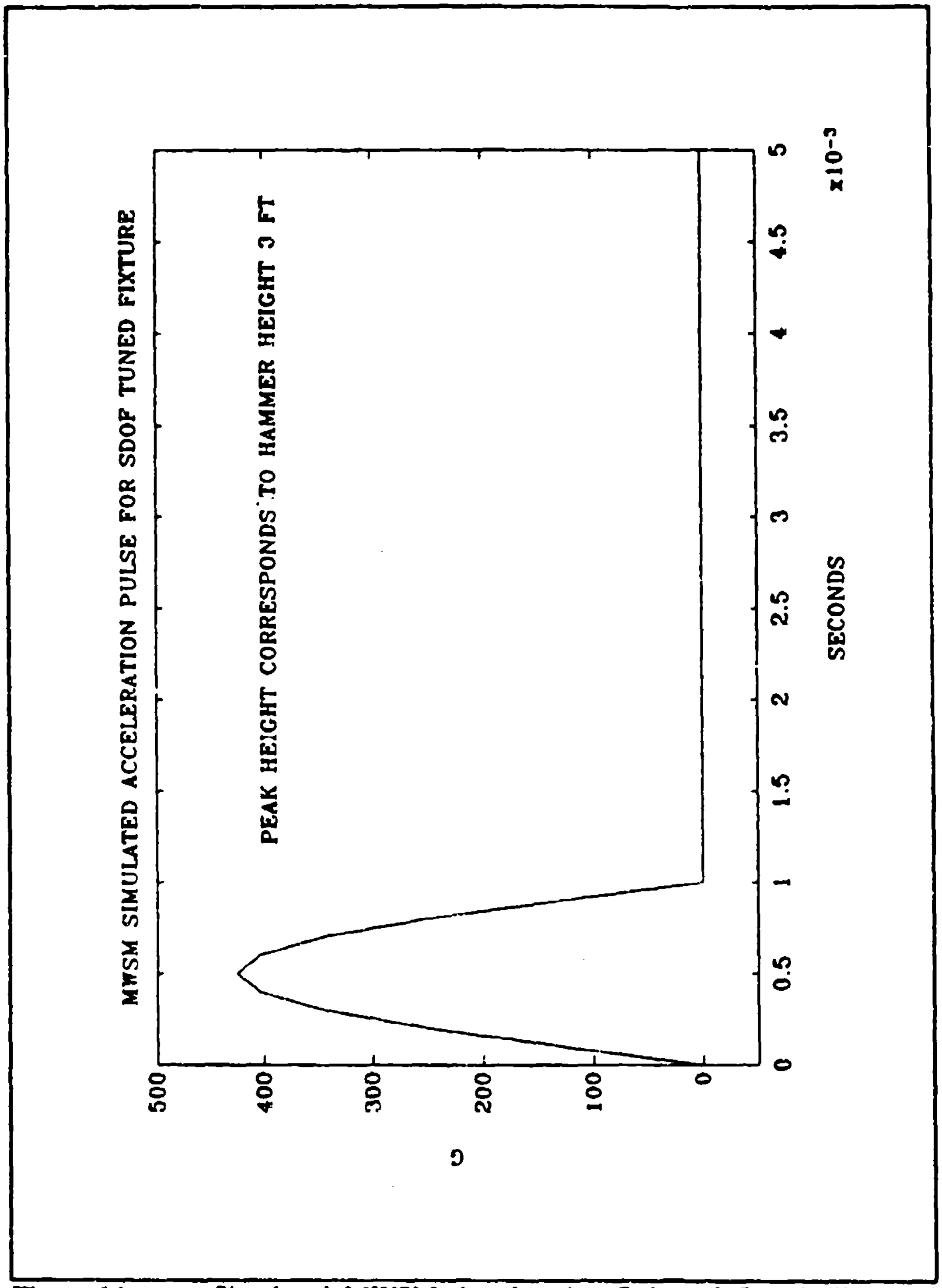

Figure 14. Simulated MWSM Acceleration Pulse of One Millisecond Duration. Corresponding Hammer Height is Three Feet.

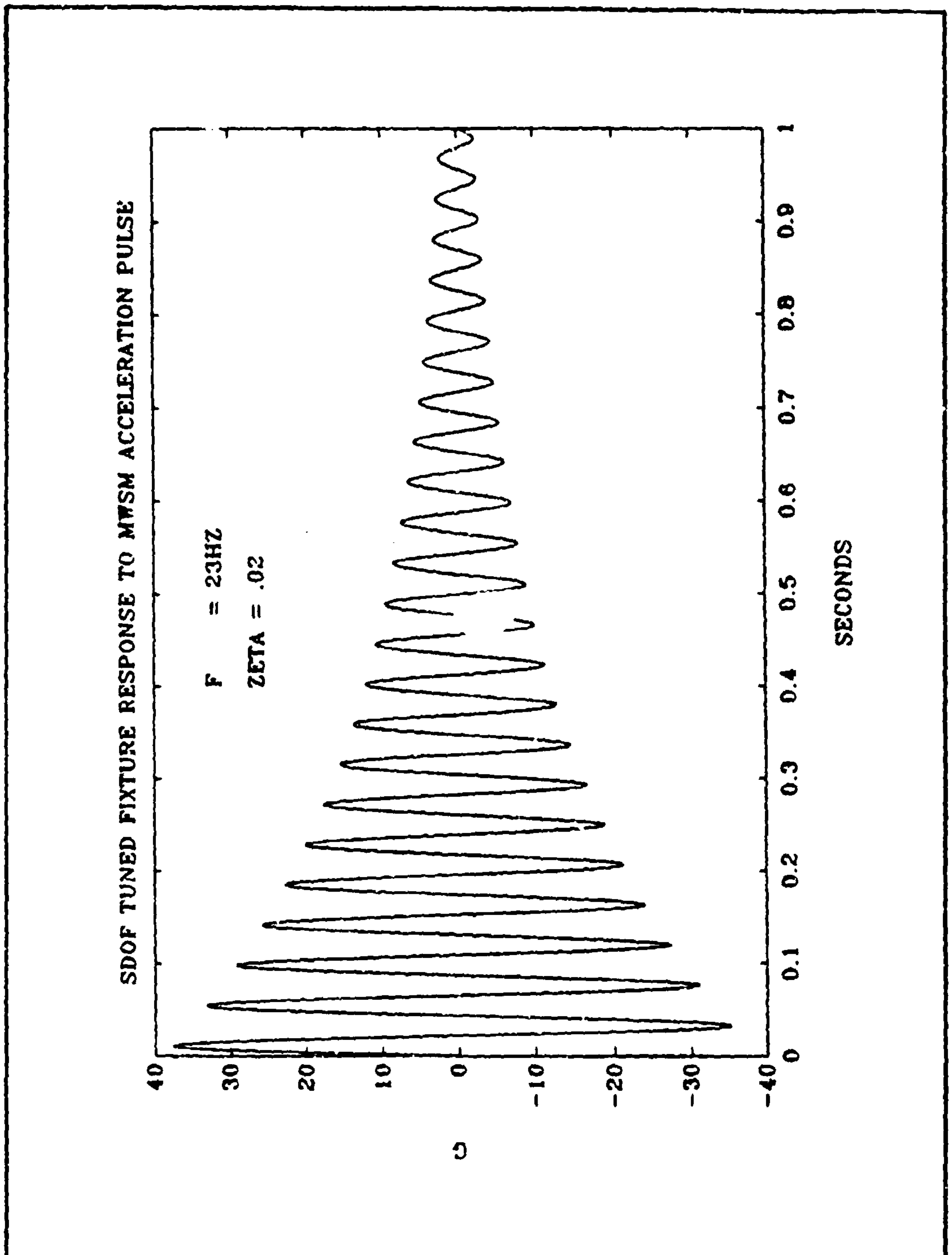

Figure 15. Modeled Single DOF Damped Tuned Mounting Fixture Acceleration Response to MWSM Acceleration Pulse of Peak 425g's.

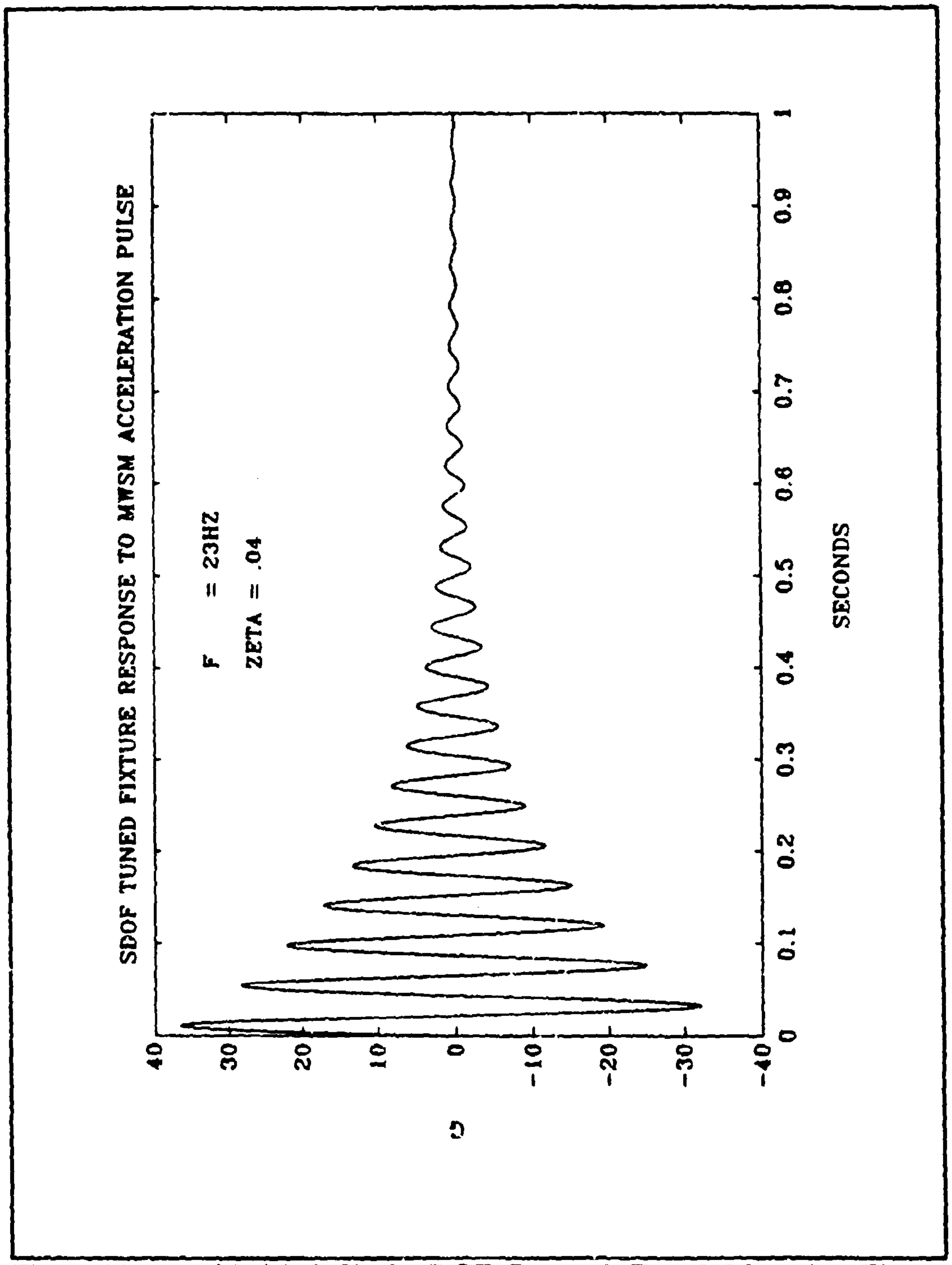

Figure 16. Modeled Single DOF Damped Tuued Mounting Fixture Acceleration Response to MWSM Acceleration Pulse of Peak 425g's.

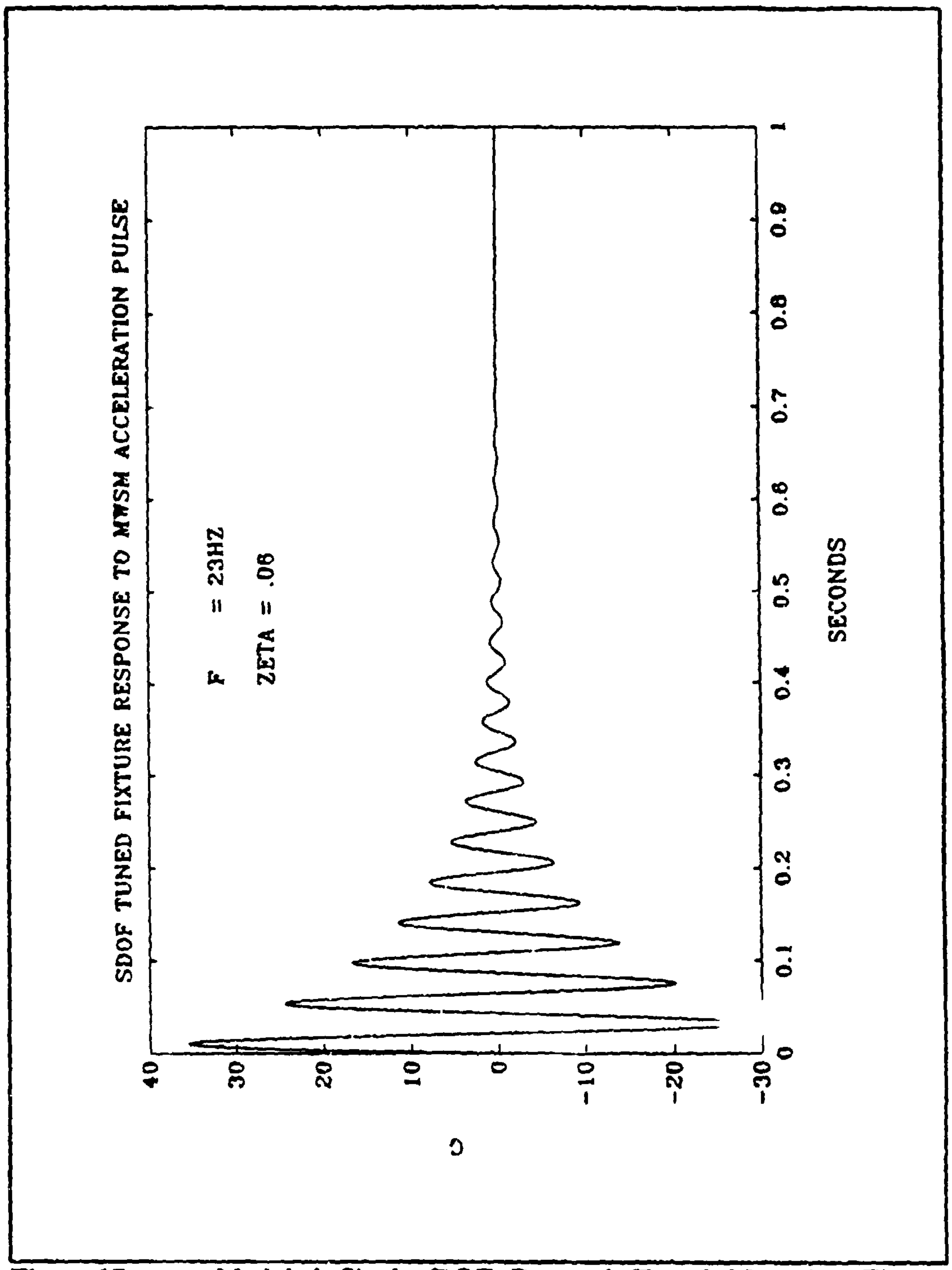

Figure 17. Modeled Single DOF Damped Tuned Mounting Fixture Acceleration Response to MWSM Acceleration Pulse of Peak 425g's.

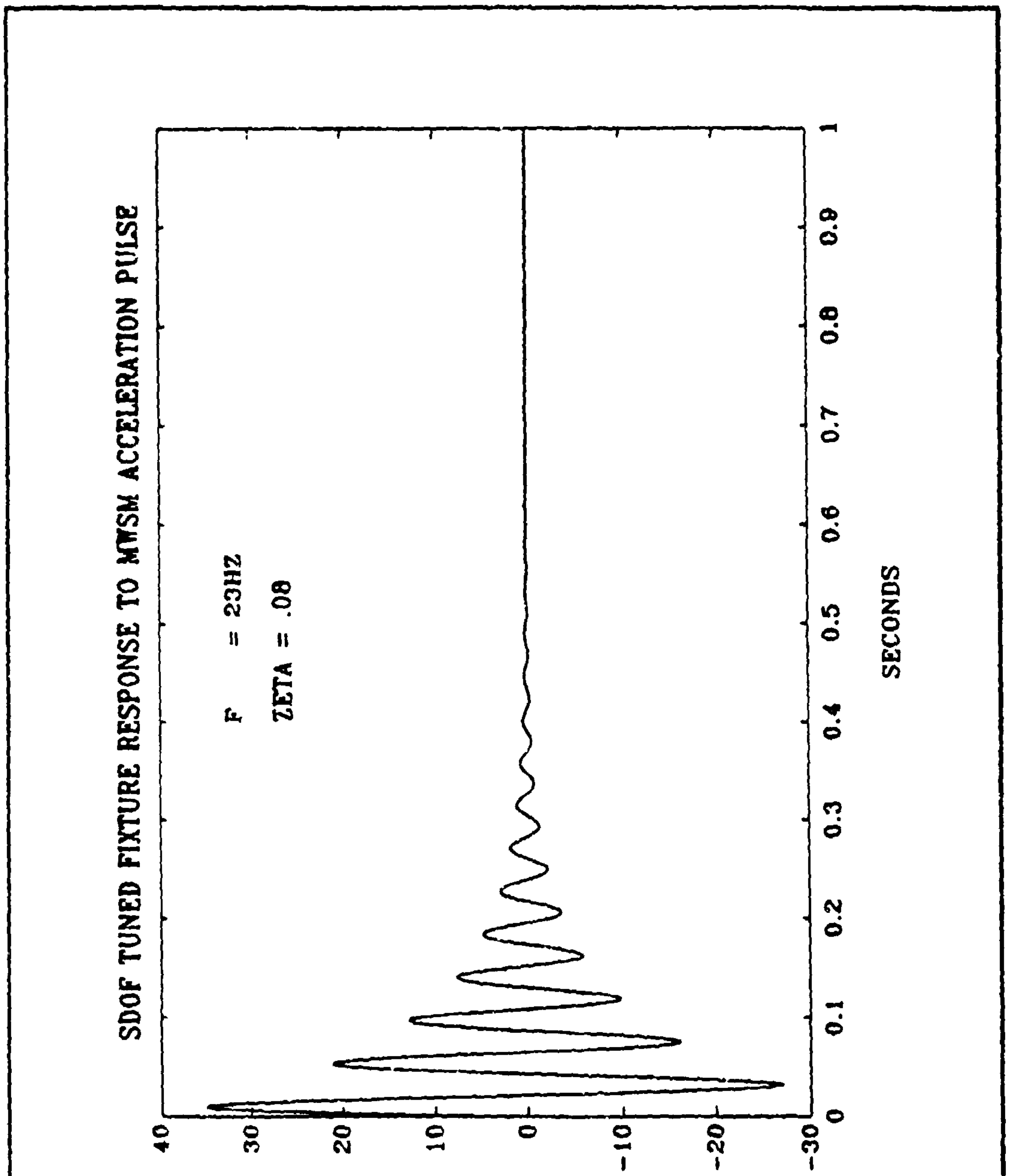

Figure 18. Modeled Single DOF Damped Tuned Mounting Fixture Acceleration Response to MWSM Acceleration Pulse of Peak 425g's.

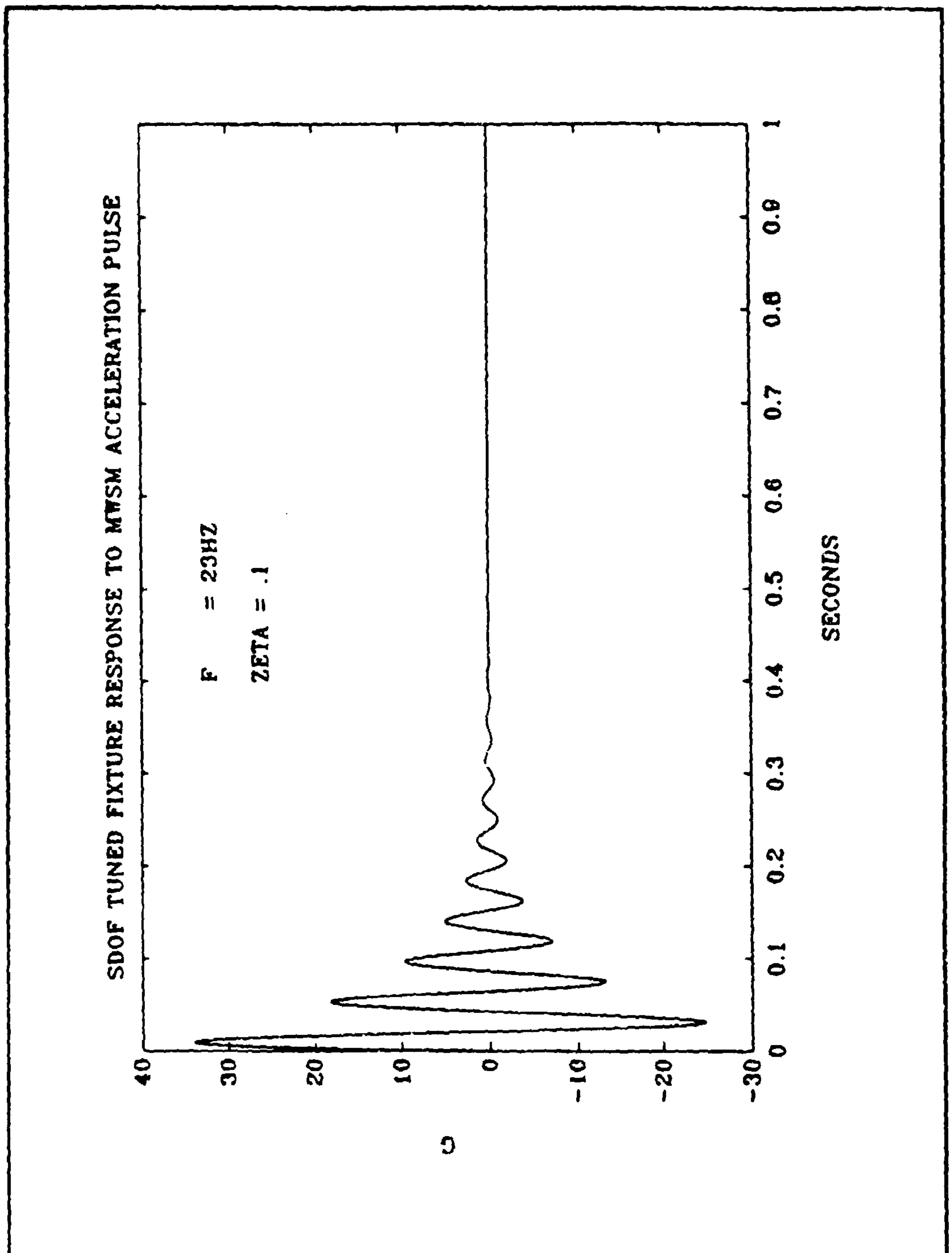

Figure 19. Modeled Single DOF Damped Tuned Mounting Fixture Acceleration Response to MWSM Acceleration Pulse of Peak 425g's.

arises due to the duration of excitement. For this study, the complete decaying sinusoidal base excitement was allowed in the formulation of the Shock Spectra. In the DTRC/UERD study, only 70 milliseconds of undamped base excitation was permitted, as this would allow the ship to achieve maximum vertical displacement due to initial shock excitation and, as stated earlier, damping influences were omitted. Closer agreement in shock spectra shape and magnitude at resonance would result with a longer duration of base excitement in the pre-shock trial analysis, after the complexities of damping are investigated. Figure 24 shows a the response of an equipment, modeled as an undamaged single DOF system, subjected to the tuned fixture's acceleration excitation.

The development of an MWSM single DOF tuned mounting fixture has been under investigation for sometime, as noted by Chalmers and Shaw (1989). The Soft Deck Simulator, shown in Figure 25, is such a device. The Soft Deck Simulator, developed by Naval Underwater Systems Center, was intended for the shock qualification of submarine combat systems equipments on the MWSM. It is a simple single DOF mass-spring system composed of springs, which are inserted in a parallel configuration by cartridges. Each cartridge possesses 12 springs which are sandwiched between support rails. The total weight of springs, support rails and test equipment must be considered when selecting the frequency response characteristics of the system. By varying the number of cartridges, the desired frequency response can be achieved. It is capable of achieving frequency responses in the range of about 19 to

30 Hertz. By the above analysis, this device could be implemented for the shock qualification of surface combat systems equipments with very promising results.

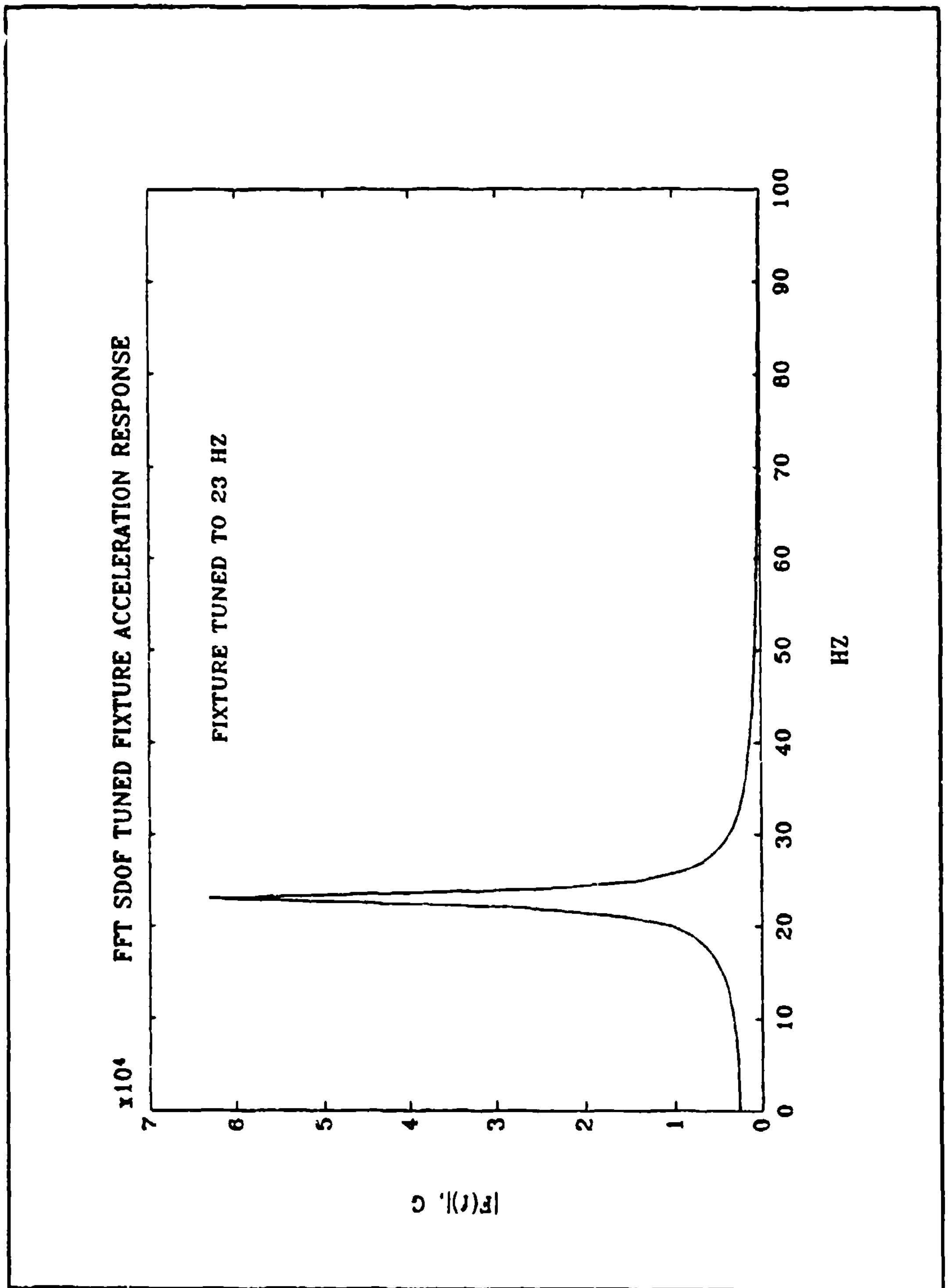

Figure 20. Fourier Transform of Single DOF Damped Tuned Mounting Fixture Acceleration Response. (Zeta = .08)

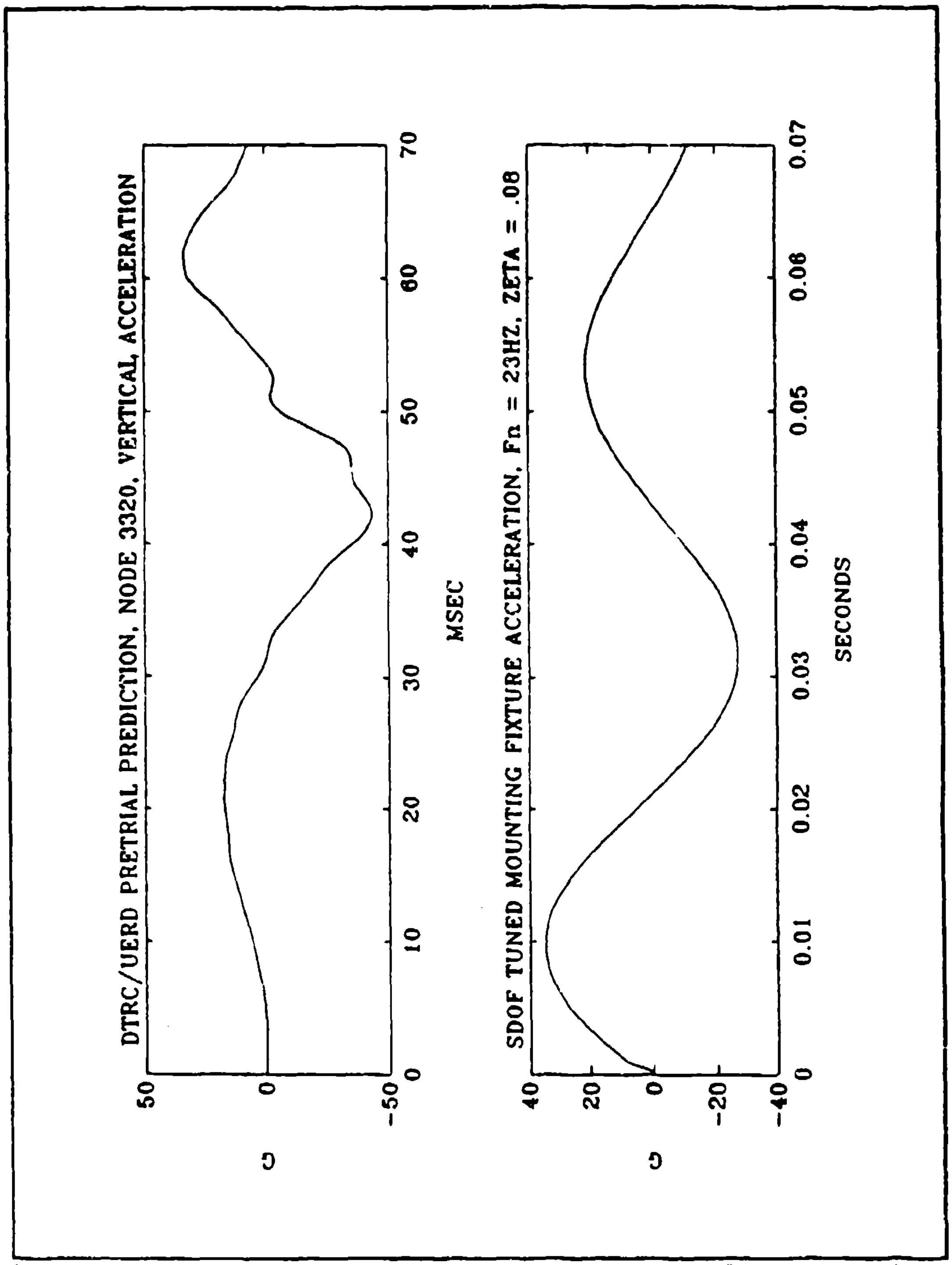

Figure 21. Acceleration Waveform Comparison. DTRC/UERD Predicted Pre-Shock Trial for Node 3320, Weight 4600 lbs, and Single DOF Tuned Mounting Fixture.

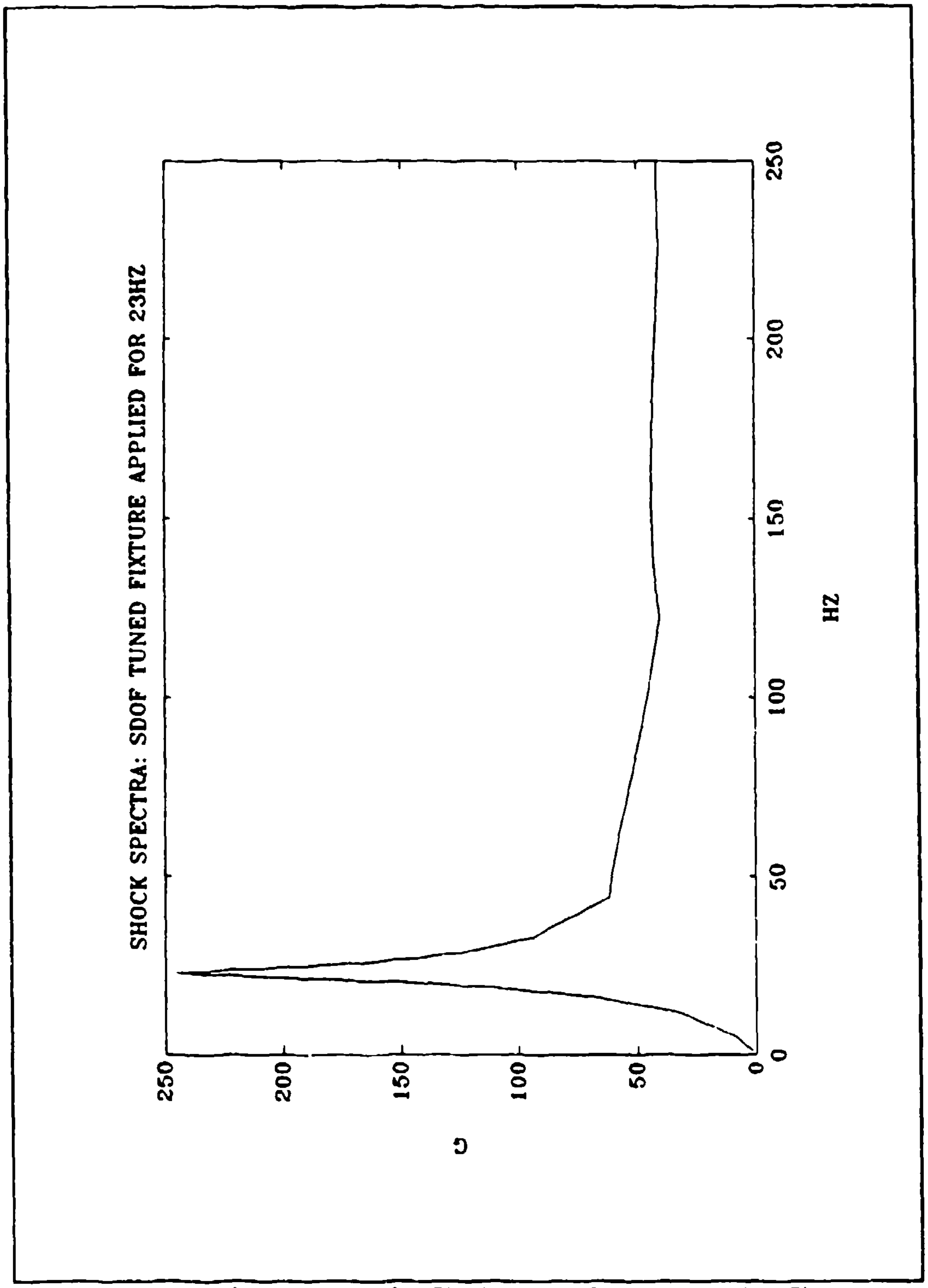

Figure 22. Shock Spectra for Single DOF Tuned Mounting Fixture.

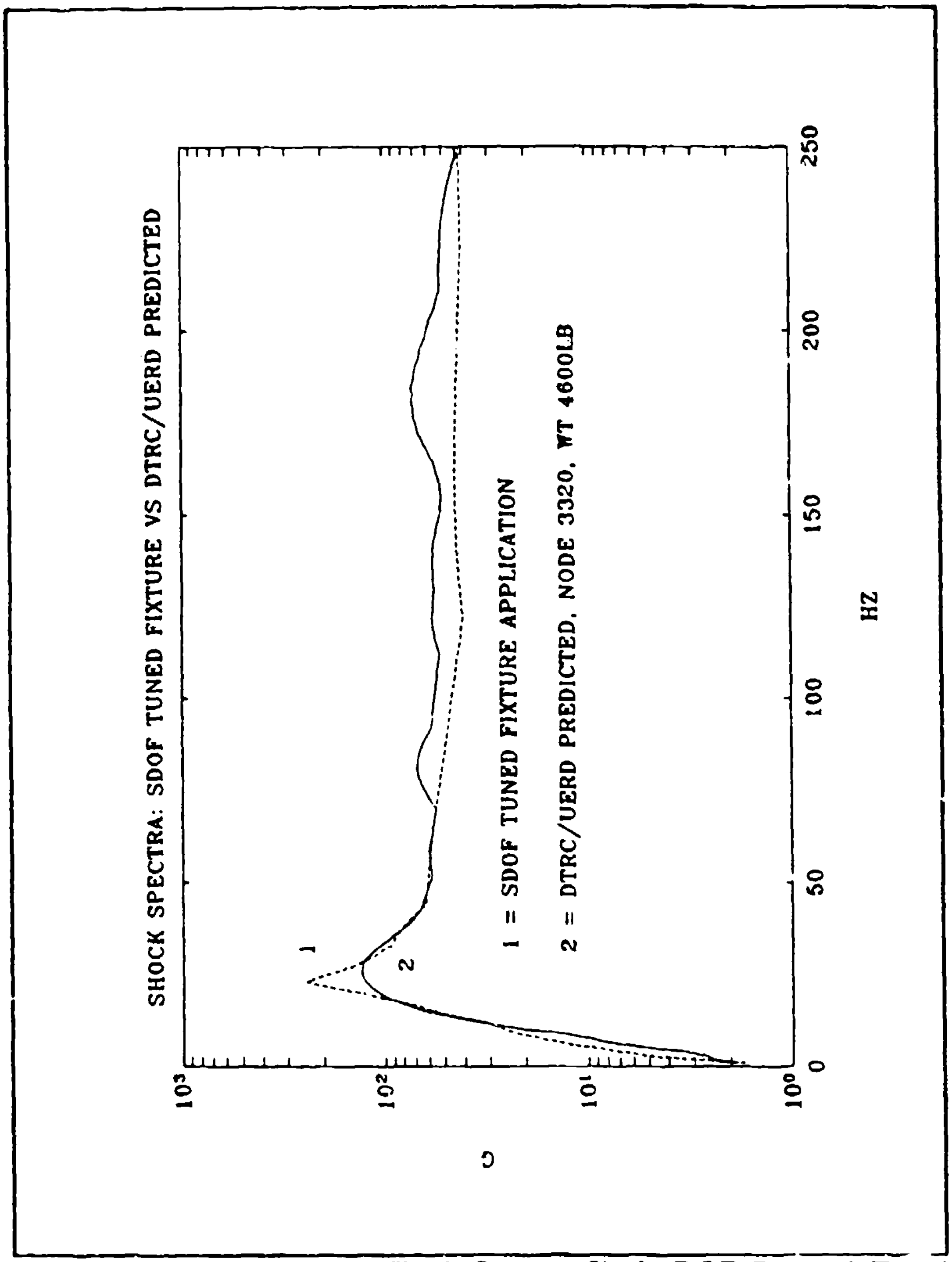

Figure 23. Comparative Shock Spectra. Single DOF Damped Tu[illegible] Mounting Fixture and DTRC/UERD Predicted Pre-Shock Trial for Frequency Amplifier Weight 4600 lbs.

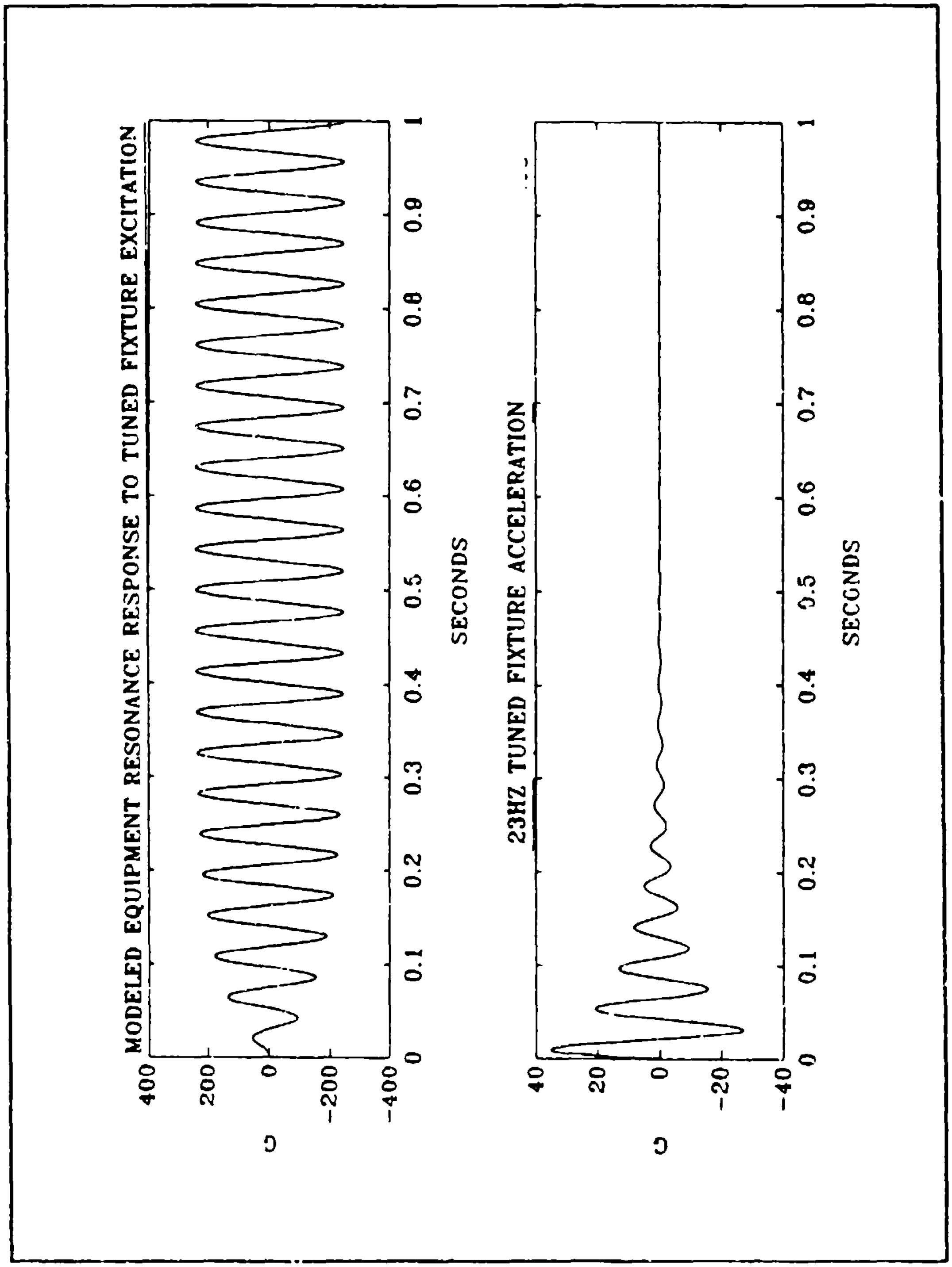

Figure 24. Resonance Response of a Undamped Single DOF Mass-Spring System to a Decaying Sinusoidal Base Acceleration Excitement.

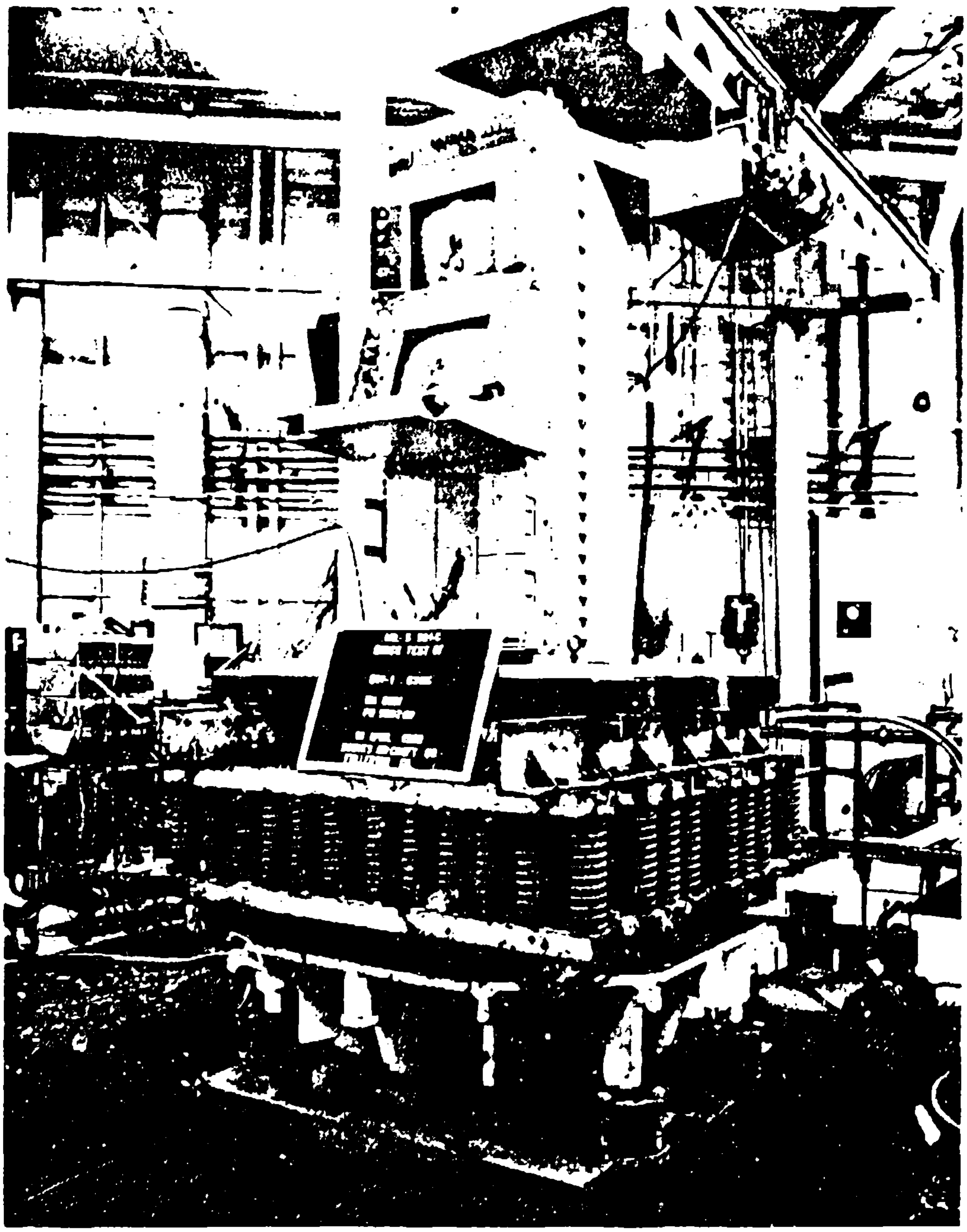

Figure 25. Single DOF Tuned Mounting Fixture Soft Deck Simulator for Shock Qualification on the MWSM. Courtesy of Hughes Aircraft.

IV. TWO DEGREE OF FREEDOM TUNED MOUNTING FIXTURE

A. DTRC/UERD PRE-SHOT TRIAL SHOCK SPECTRAL ANALYSIS

As presented earlier, both the light weight range and medium weight range equipment possess Vertical Orientation Shock Spectra that reveal two or more well defined dominant peaks at discrete frequencies. The light weight range equipment, a Radar Receiver/Transmitter weighing 325 lbs, possesses two peaks, one at 60 and the other at 155 Hertz, with a 2:1 magnitude ratio. A two DOF uniaxial tuned mounting fixture would provide the frequency characteristics necessary to simulate this phenomena.

The medium weight range equipment, a Beam Programmer weighing 1000 lbs, possesses three well defined dominant peaks. Two peaks, with about the same magnitude, occur at about 23 and 60 Hertz, respectively. The third and absolute dominant peak, by a factor of two, occurs at about 155 Hertz. A two DOF uniaxial tuned mounting fixture could simulate the two most important frequency characteristics revealed within this Shock Spectra. Knowing that higher accelerations are experienced by equipment possessing higher fundamental frequencies, the 60 Hertz and 155 Hertz frequencies are deemed the two most important frequency characteristics for this study. Further investigation of other combinations are possible.

B. TWO DEGREE OF FREEDOM TUNED MOUNTING FIXTURE MODEL

Extending the concept developed by Chalmers and Shaw (1989) to a two DOF tuned mounting fixture is the focus of this chapter. Figure 26 depicts the extended concept. The upper tier of the system possesses the test item of interest. The coupled response of this tier to base shock excitation will be the shock excitement experienced by the item tested. The mass, damping and stiffness relations of the two DOF tuned mounting fixture model will need to be investigated.

1. Two Degree of Freedom Tuned Mounting Fixture Mathematical Model

Referring to figure 26 again, let the upper tier, with mass m_1, be comprised of the equipment tested along with its associated support mountings. The lower tier, with mass m_2, is comprised of test weights and support mountings. Each tier possesses characteristic damping and stiffness properties, c and k, respectively. Expressing the absolute coordinates, x_1 and x_2, in terms of relative coordinates yields:

$$\begin{aligned} y_1 &= x_1 - z \\ \dot{y}_1 &= \dot{x}_1 - \dot{z} \\ \ddot{y}_1 &= \ddot{x}_1 - \ddot{z} \end{aligned} \tag{10}$$

$$\begin{aligned} y_2 &= x_2 - z \\ \dot{y}_2 &= \dot{x}_2 - \dot{z} \\ \ddot{y}_2 &= \ddot{x}_2 - \ddot{z} \end{aligned} \tag{11}$$

where the z coordinate represents the foundation motion. Once again, if the relative coordinate quantities are known, then it is a simple matter to obtain the absolute quantities.

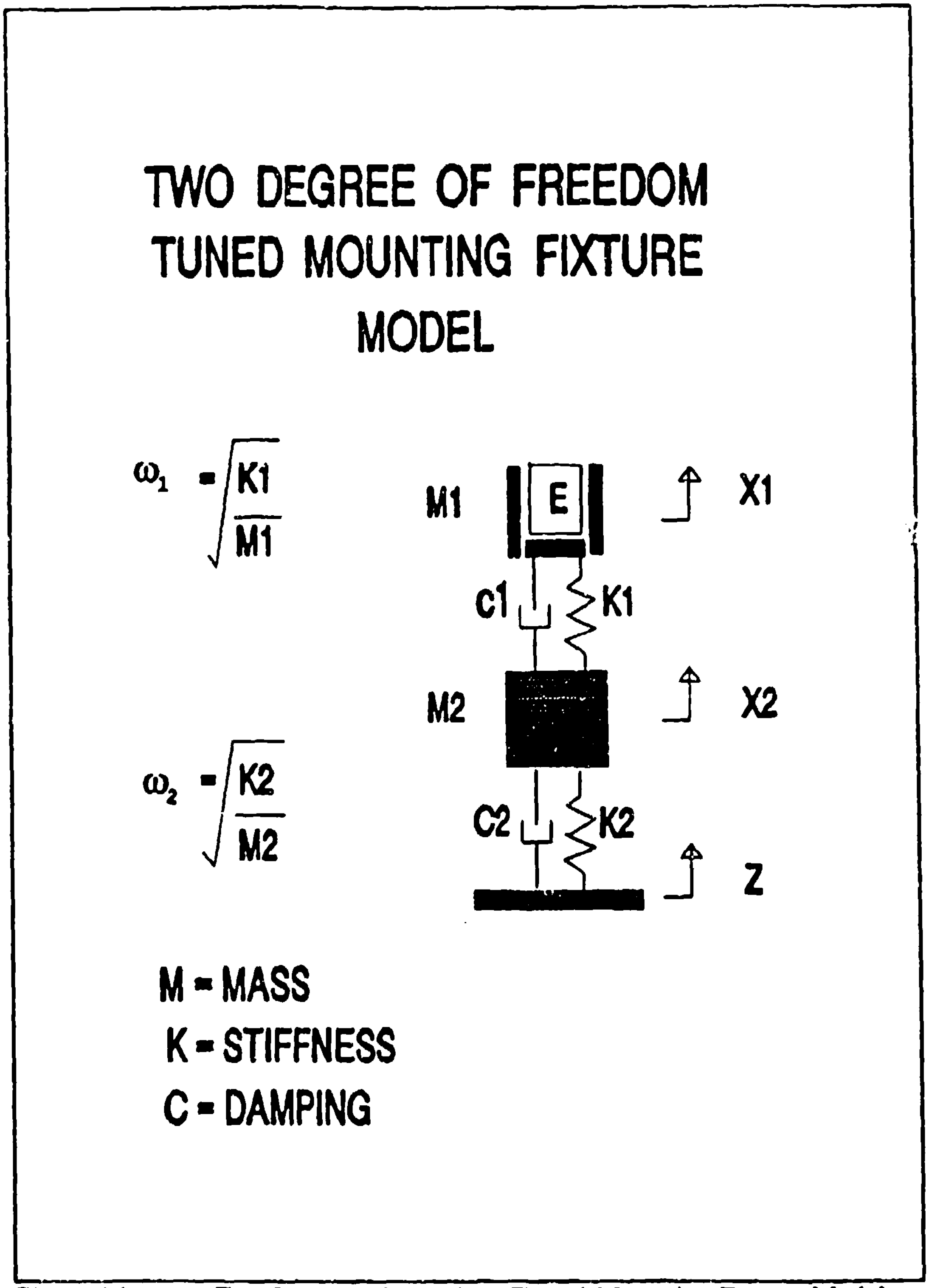

Figure 26. Two Degree of Freedom Tuned Mounting Fixture Model.

The coupled equations of motion for this system can be obtained and are expressed in matrix form below.

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \begin{Bmatrix} \ddot{y}_1 \\ \ddot{y}_2 \end{Bmatrix} + \begin{bmatrix} c_1 & -c_1 \\ -c_1 & (c_1+c_2) \end{bmatrix} \begin{Bmatrix} \dot{y}_1 \\ \dot{y}_2 \end{Bmatrix} + \begin{bmatrix} k_1 & -k_1 \\ -k_1 & (k_1+k_2) \end{bmatrix} \begin{Bmatrix} y_1 \\ y_2 \end{Bmatrix} = \begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \begin{Bmatrix} -\ddot{z} \\ -\ddot{z} \end{Bmatrix} \tag{12}$$

Each tier has its own natural frequency which may be expressed in either radians per second or in Hertz, cycles per second. Equations 13 through 16 present those relationships.

$$\omega_1 = \sqrt{\frac{k_1}{m_1}} \tag{13}$$

$$f_1 = \frac{1}{2\pi}\sqrt{\frac{k_1}{m_1}} \tag{14}$$

$$\omega_2 = \sqrt{\frac{k_2}{m_2}} \tag{15}$$

$$f_2 = \frac{1}{2\pi}\sqrt{\frac{k_2}{m_2}} \tag{16}$$

If one were to separate the tiers, each could be viewed as a single DOF mass-damper-spring system and the overall system response could be formulated by modal analysis. Equations 14 and 16 are the uncoupled natural frequencies of each tier. For this study, the coupled natural frequencies are of interest, since it is at those frequencies that the system response is observed. Defining the ratio of the upper tier mass to the lower tier mass as follows:

$$\alpha=\frac{mass_1}{mass_2} \tag{17}$$

it can be shown, as noted in Shin (1981), that the relationship between the system's coupled natural frequencies and uncoupled frequencies are:

$$fn_1=\frac{1}{\sqrt{2}}[f_1^2+\alpha f_1^2+f_2^2-[(f_1^2+\alpha f_1^2+f_2^2)^2-(2f_1f_2)^2]^{1/2}]^{1/2} \tag{18}$$

$$fn_2=\frac{1}{\sqrt{2}}[f_1^2+\alpha f_1^2+f_2^2+[(f_1^2+\alpha f_1^2+f_2^2)^2-(2f_1f_2)^2]^{1/2}]^{1/2} \tag{19}$$

where fn_1 and fn_2 are the system's coupled natural frequencies in Hertz, which are easily converted to radians per second as noted below.

$$\omega n_1=2\pi fn_1 \tag{20}$$

$$\omega n_2=2\pi fn_2 \tag{21}$$

Knowing both the system's coupled and uncoupled natural frequencies, along with the system's mass ratio, equation 12 may be decoupled by modal analysis, as presented in Mierovitch (1986), and solved as two single DOF systems. Once the response of each is single DOF is known, the coupled response may be obtained. Equations 22 and 23 show the decoupled equations of motion using the coupled natural frequencies, ωn_1 and ωn_2. It must be noted that the system damping is assumed to be a linear combination of system mass and stiffness. This valid assumption will permit the modal analysis approach.

$$\ddot{q}_1+2\xi\omega n_1\dot{q}_1+\omega n_1^2 q_1=-\ddot{z}_1 \tag{22}$$

$$\ddot{q}_2+2\xi\omega n_2\dot{q}_2+\omega n_2^2 q_2=-\ddot{z}_2 \tag{23}$$

The natural coordinates, q_1 and q_2, are related to the relative coordinates, y_1 and y_2, by the modal matrix as follows:

$$\begin{Bmatrix} y_1 \\ y_2 \end{Bmatrix} = \begin{bmatrix} u_{11} & u_{12} \\ u_{21} & u_{22} \end{bmatrix} \begin{Bmatrix} q_1 \\ q_2 \end{Bmatrix} \tag{24}$$

Equations 22 and 23 were solved with an unconditionally stable numerical integration scheme and then the absolute motions were obtained per the relations in equations 10 and 11. The Fortran code listing for this numerical solution is presented in Appendix D.

2. Two Degree of Freedom Tuned Mounting Fixture Modeled Application

The DTRC/UERD shock spectral analysis revealed two common frequencies of interest between the light weight and medium weight range equipment. Those frequencies were 60 and 155 Hertz. These are the coupled response frequencies that must be observed in the modeled application. As stated earlier, knowing the two uncoupled system natural frequencies, f_1 and f_2, along the mass ratio, the coupled natural frequencies, fn_1 and fn_2, may be obtained by equations 18 and 19. However, only the coupled frequencies, fn_1 and fn_2 are known, thus, a sensitivity study is required in order to attain reasonable values for the uncoupled tier frequencies, f_1 and f_2, and system mass ratio.

Figures 27 through 31 present the results of a study performed where the lower tier natural frequency, f_2, was held constant and iterations of f_1 were conducted to ascertain the combination of f_1, f_2 and mass ratio that would provide the desired coupled frequency response at 60 and 155 Hertz. Mass ratios of 1 and .1 were studied. In all cases, the coupled frequencies for a mass ratio of .1 were bounded by those of mass ratio 1, thus defining a mass ratio coupled frequency envelope. Figure 30 shows that, at a mass ratio of 1 and f_2 equal to 100 Hertz, a selection of f_1 equal to 94 Hertz would provide the desired coupled frequency response for this study. In addition, a mass ratio of 1 would minimize the total anvil table top weight, thus permitting a wider application of the two DOF tuned mounting fixture. For the remainder of this analysis, the upper tier natural frequency is 94 Hertz and the lower tier possesses a natural frequency of 100 Hertz. The Fortran code for this iterative scheme is provided in Appendix E.

Knowing both the set of coupled and uncoupled natural frequencies, along with a mass ratio of 1, an analysis of the two DOF tuned mounting fixture was then permitted. Tier weights of 1700 lbs each were assumed. This would allow a wide range of possibilities for equipment and support mounting combinations and would result in a moderate total anvil table top weight of 3400 lbs. Further, knowing the respective tier mass and natural frequency properties, the equivalent tier stiffnesses could be calculated.

The MWSM acceleration half-sine acceleration pulse for this analysis is presented in Figure 32. As this model is a relatively stiff system where higher accelerations can be expected, the peak MWSM acceleration pulse had to be adjusted to provide meaningful results. A peak acceleration of 75g's was used in the modeled application analysis of the two DOF tuned mounting fixture. The MWSM is capable of delivering such a peak acceleration. A MWSM calibration study, based on the analysis of the two DOF tuned mounting fixture, will be necessary to determine suitable hammer heights for shock testing equipment mounted to the machine by this fixture.

The same type of sensitivity study in damping was conducted as in the single DOF tuned mounting fixture analysis. For simplicity, both tiers were subjected to the same damping factor, however, other combinations are possible. Figures 33 through 36 show the effects of damping on the upper tier acceleration response.

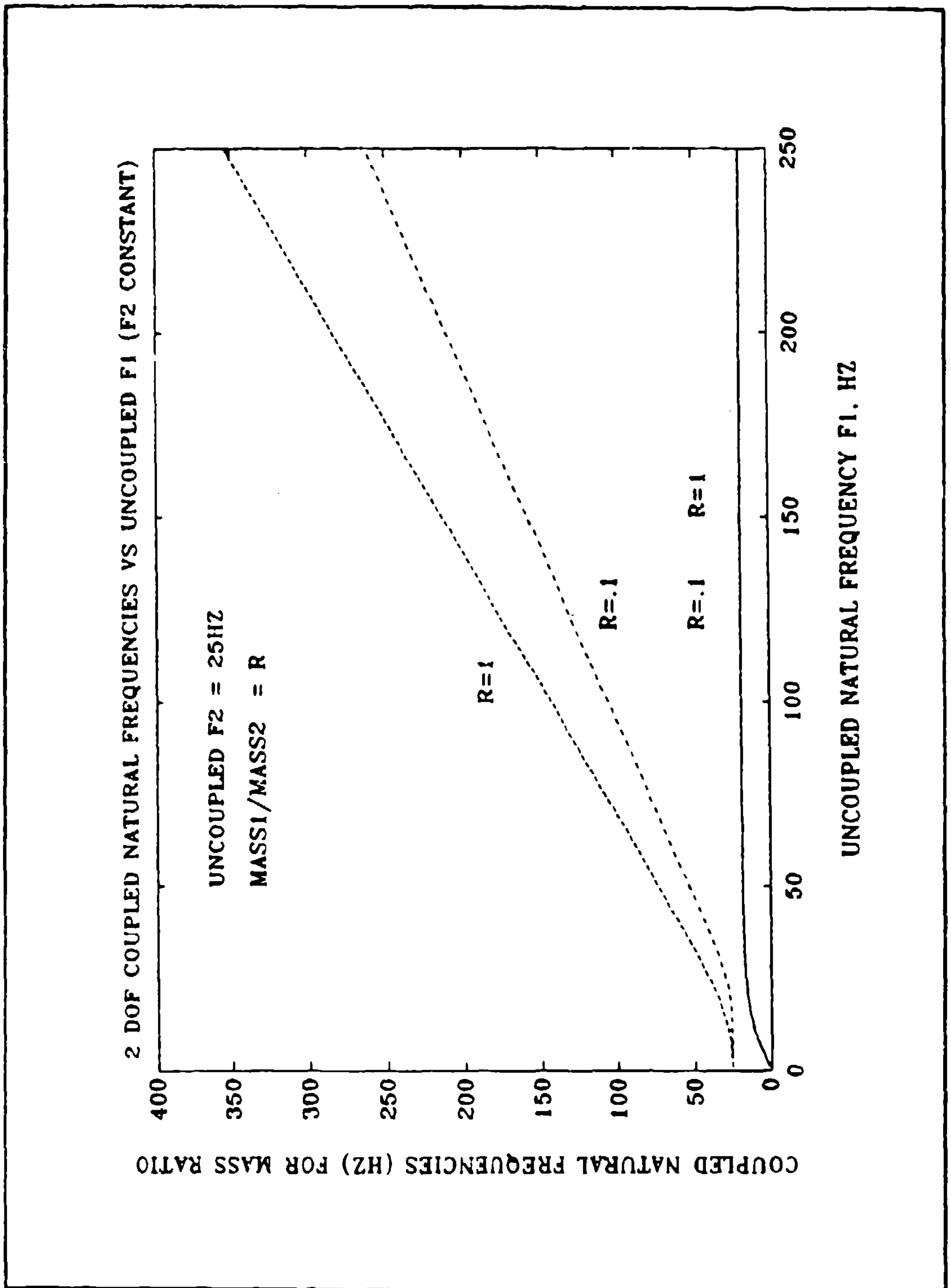

Figure 27. Coupled Natural Frequencies as a Function of f_1 and Mass Ratio.

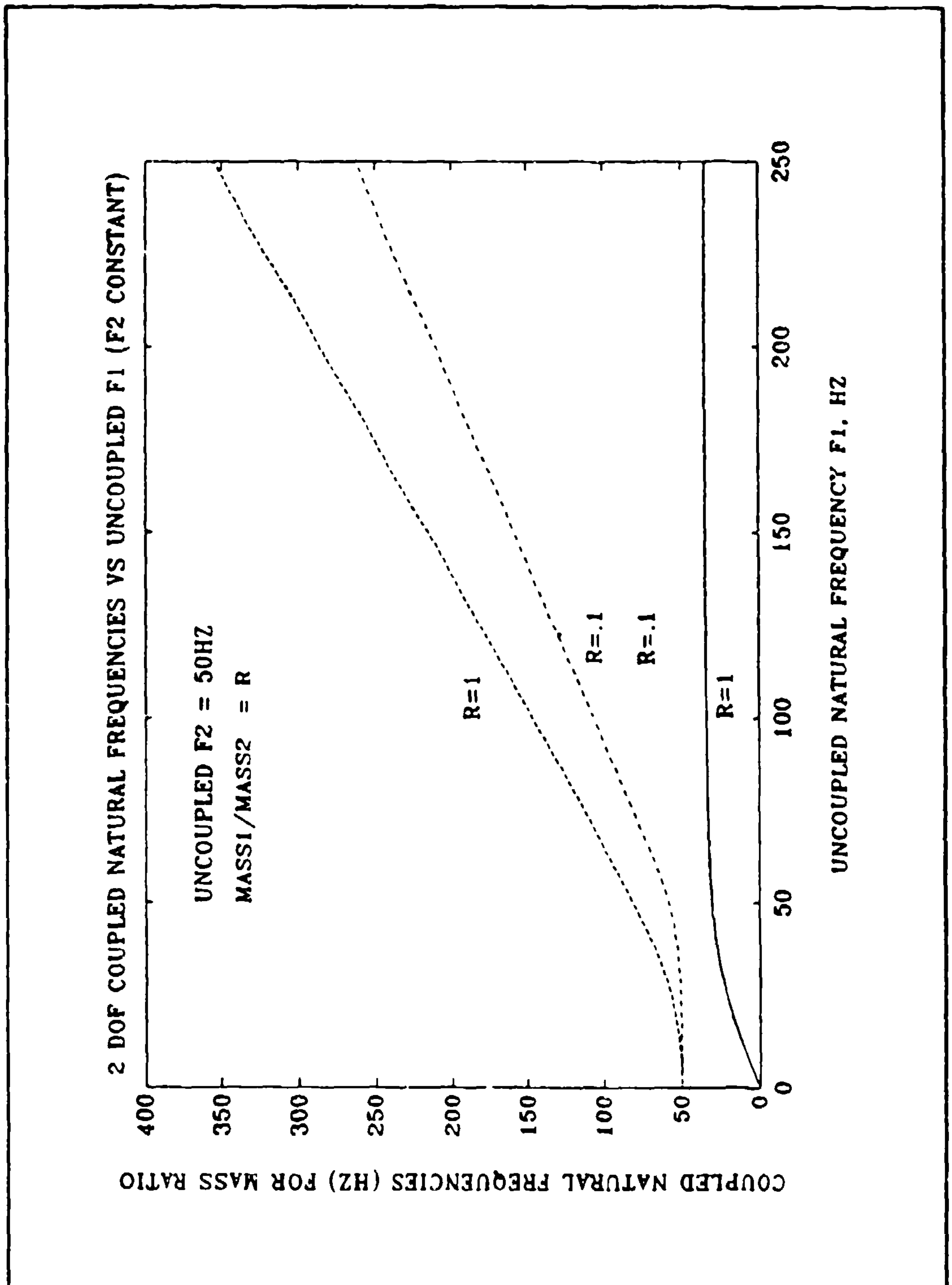

Figure 28. Coupled Natural Frequencies as a Function of f_1 and Mass Ratio.

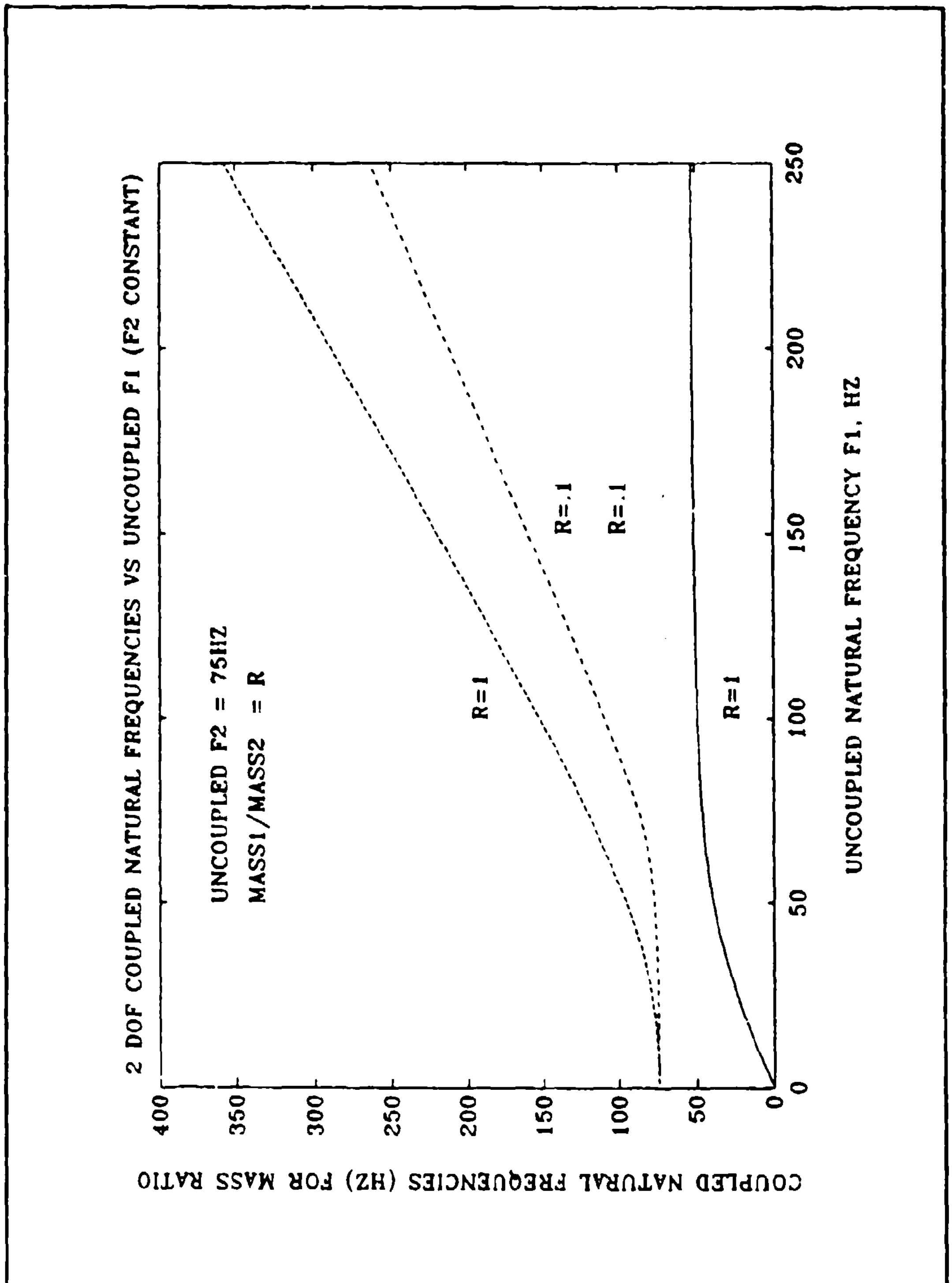

Figure 29. Coupled Natural Frequencies as a Function of f_1 and Mass Ratio.

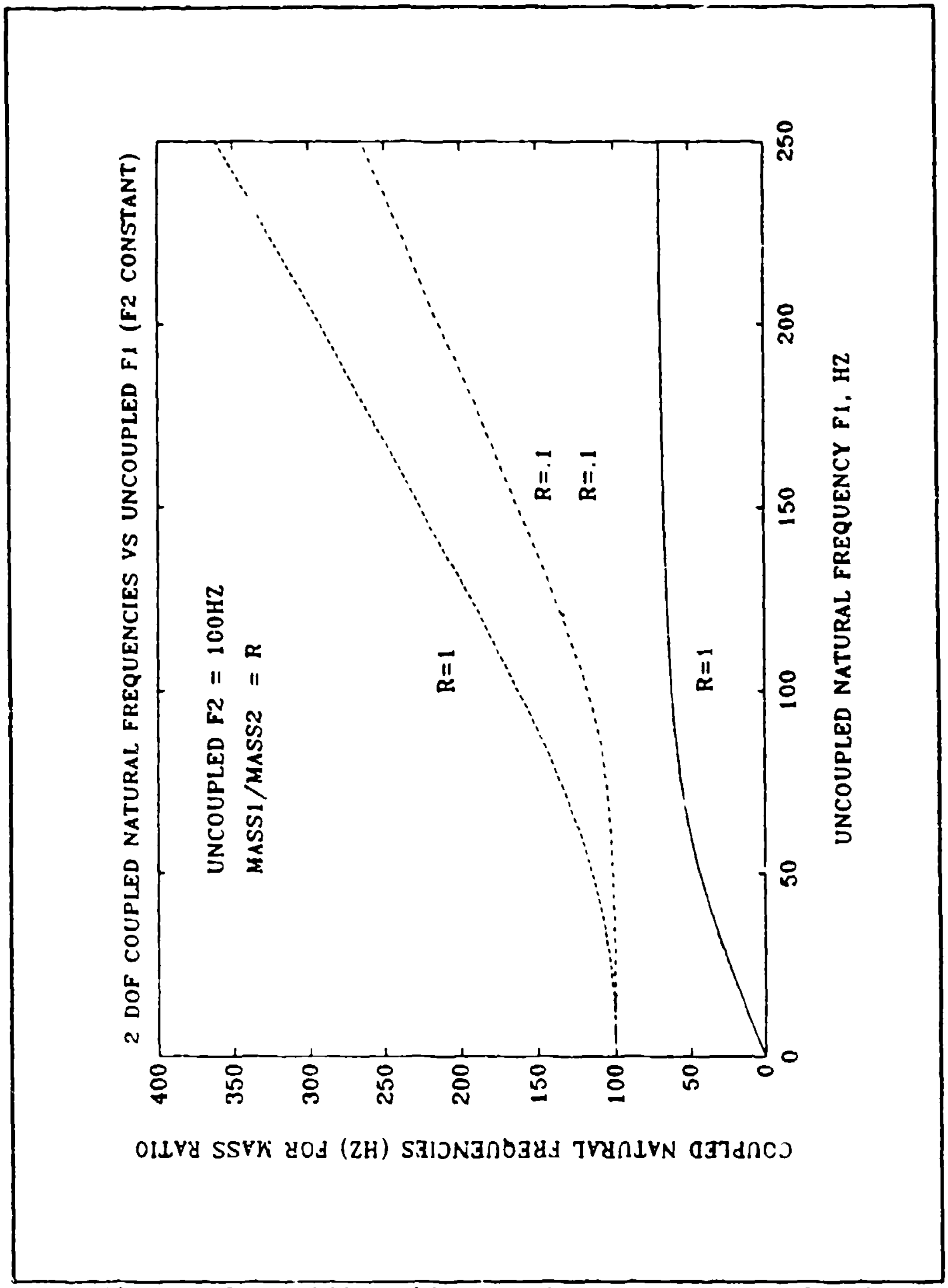

Figure 30. Coupled Natural Frequencies as a Function of f_1 and Mass Ratio.

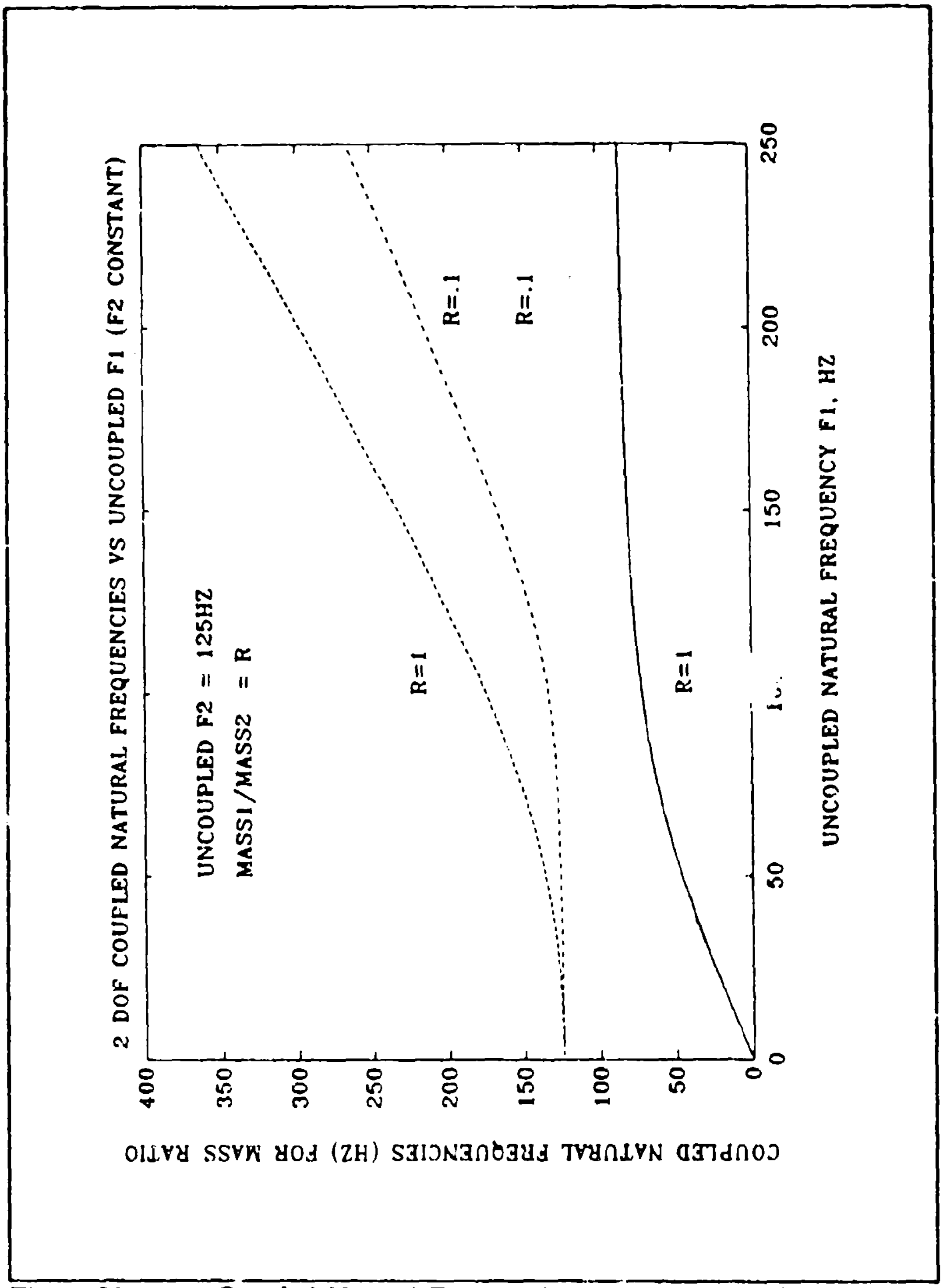

Figure 31. Coupled Natural Frequencies as a Function of f_1 and Mass Ratio.

Figure 34 , with a damping factor of .02, shows the best decaying characteristics consistent with the MWSM calibration data, compiled by Costanzo and Clements (1988), and was used in the further analysis of the two DOF tuned mounting fixture.

Figure 37 shows the Fourier Transform of the upper tier acceleration. As expected, there are two peaks, one at 60 Hertz and the other at 155 Hertz. The magnitude of each peak is proportional to the amplitude of that frequency component in the acceleration waveform. The dominant first frequency component indicates that the first mode of vibration has a greater contribution to the response. Figures 38 and 39 show the comparisons of the first 70 milliseconds of the modeled upper tier's acceleration waveform with those predicted by the DTRC/UERD study for the low weight and medium weight range equipment. Very close agreement exists between the modeled acceleration waveform and the acceleration waveform for the low weight range equipment. Further studies as to the effects of tier weights on the acceleration waveforms will provide closer agreement between the modeled upper tier waveform and that waveform predicted by a pre-shock trial analysis.

Figure 40 shows the resultant Shock Spectra using the upper tier's acceleration response as base excitation. The shape and relative peak magnitudes are typical for such an excitement. Figures 41 and 42 show the comparison between the DTRC/UERD predicted Shock Spectra and that resultant from this study of a two DOF tuned mounting fixture. There is very close agreement in apectral shape and magnitude with respect to the low weight range equipment's Shock Spectra. Closer agreement in the Shock Spectra of the medium weight range equipment will result after further studies are conducted as to the effects of tier weights in the coupled response. Further, closer agreement would result in the spectras if longer duration base excitation were permitted in the pre-shock trial analysis. This, as noted previously, can occur once the complexities of damping are investigated, as noted previously. Figure 43 shows the typical resonance response of an equipment, modeled as an undamped single DOF system, subjected to the upper tier's decaying sinusoidal acceleration, a waveform consisting of two frequency components, one at 60 Hertz and the other at 155 Hertz.

Development of a two DOF tuned mounting fixture with a coupled frequency response of 60 and 155 Hertz can be obtained once the mass ratio is selected. For this study, a mass ratio of 1 optimized the total anvil table top weight and provided reasonable tier natural frequencies of 94 and 100 Hertz. To achieve those relatively high tier natural frequencies, the proposed model in Appendix F can be constructed. The equivalent stiffness properties of each tier are provided by the tier support mounting beam configurations and the manner in which those beams are loaded. A study into the design and construction of such a mounting fixture is warranted by the above analysis.

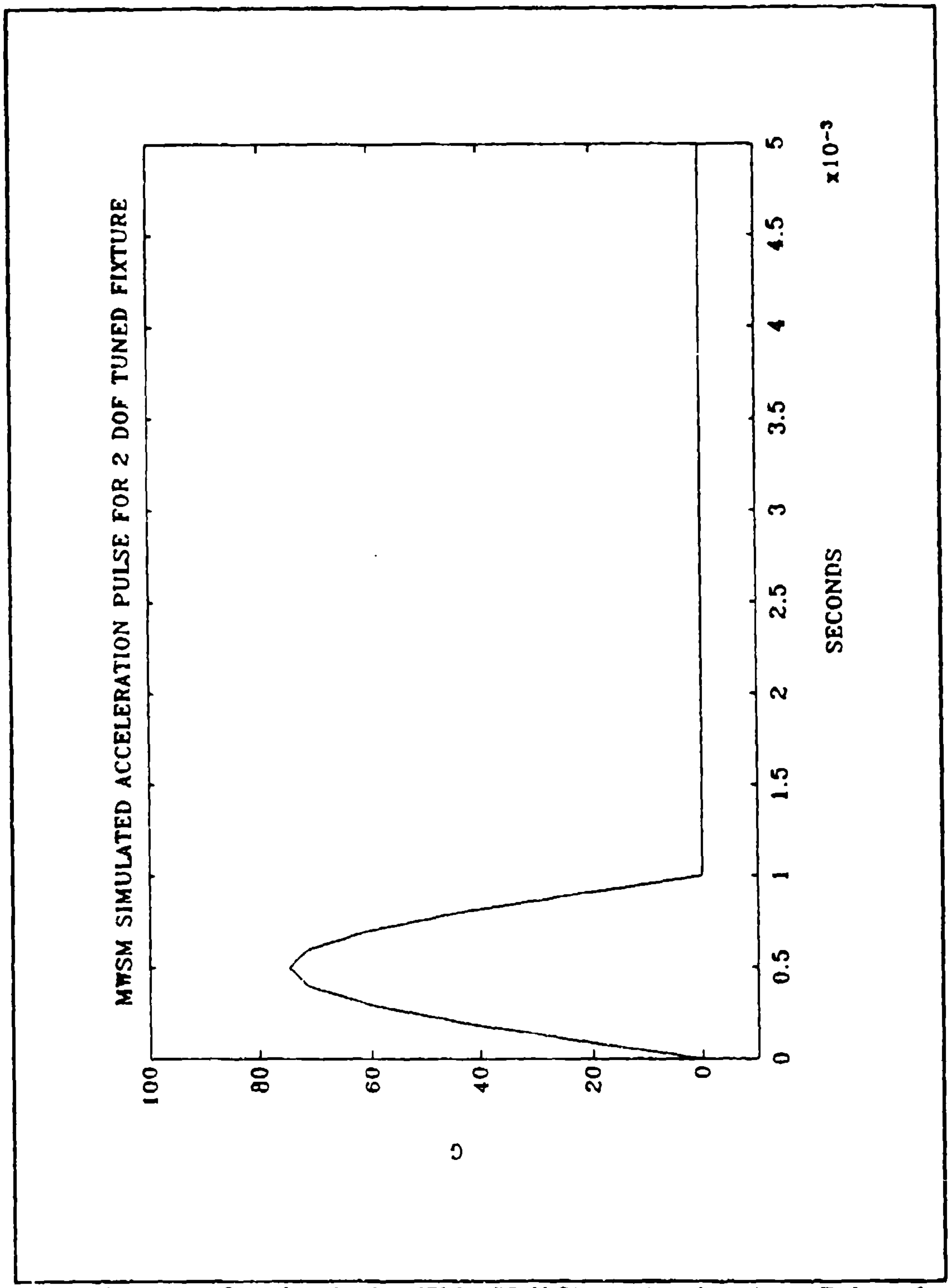

Figure 32. Simulated MWSM Half-Sine Acceleration Pulse for Evaluation of the Two DOF Tuned Mounting Fixture.

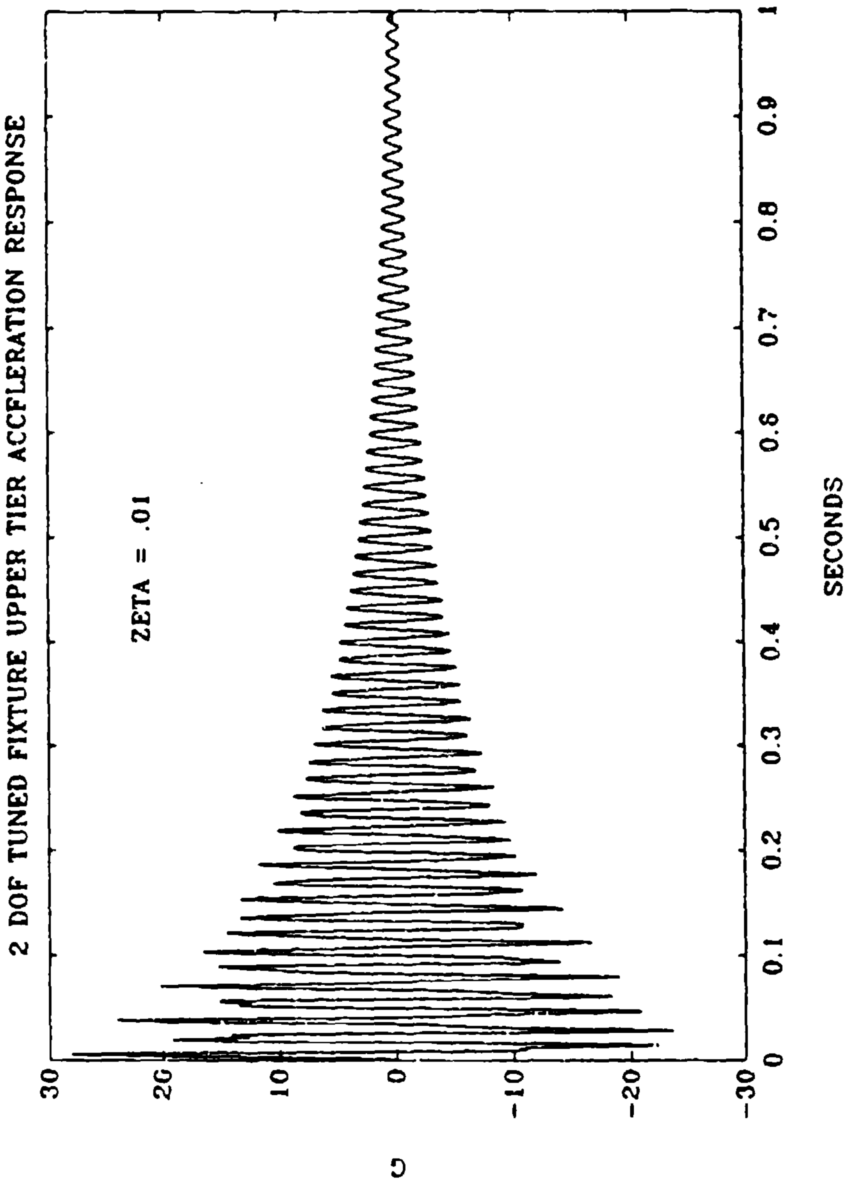

Figure 33. Two DOF Tuned Mounting Fixture Upper Tier Damped Acceleration Waveform. Zeta = .01

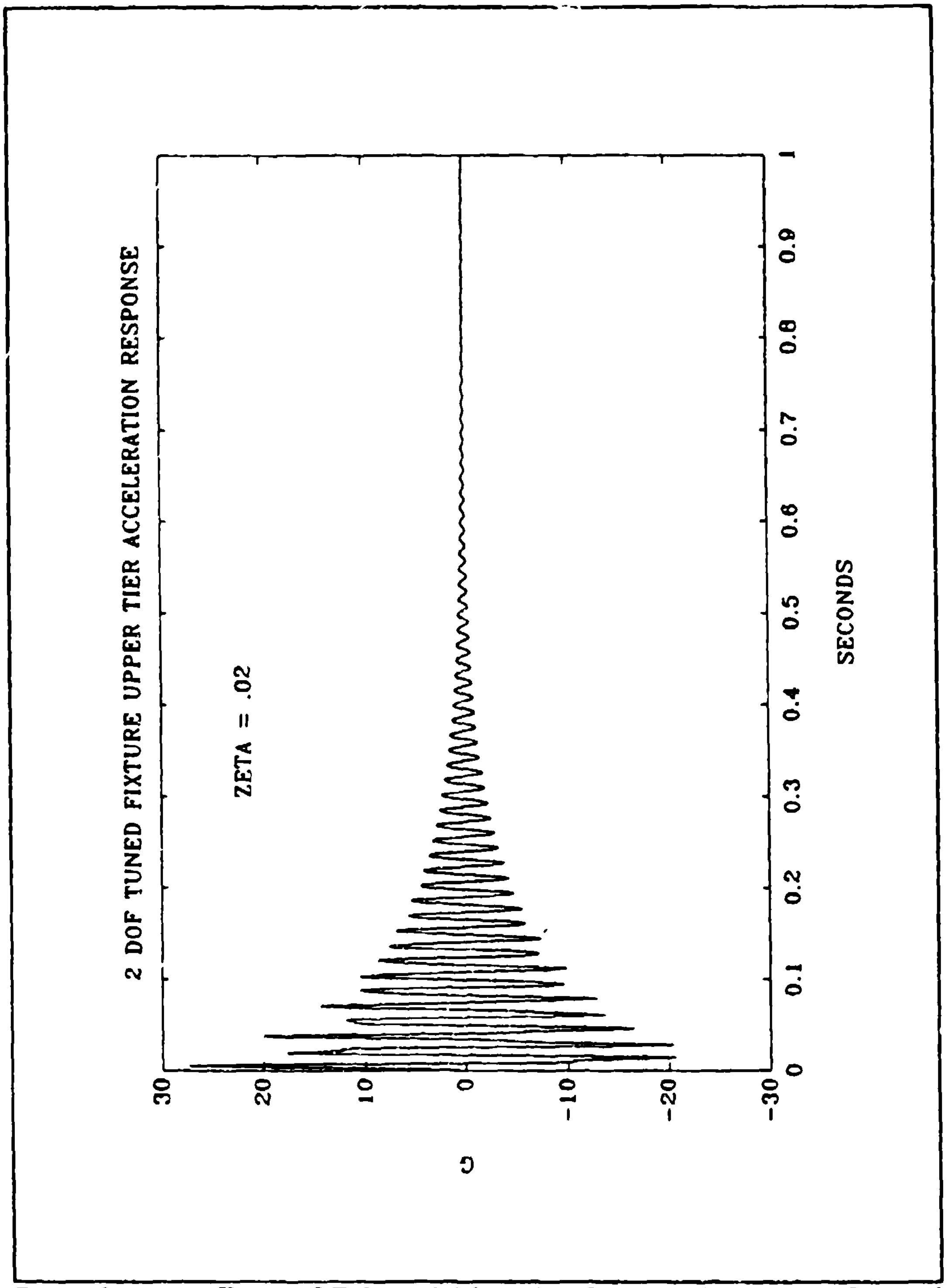

Figure 34. Two DOF Tuned Mounting Fixture Upper Tier Damped Acceleration Response. Zeta = .02.

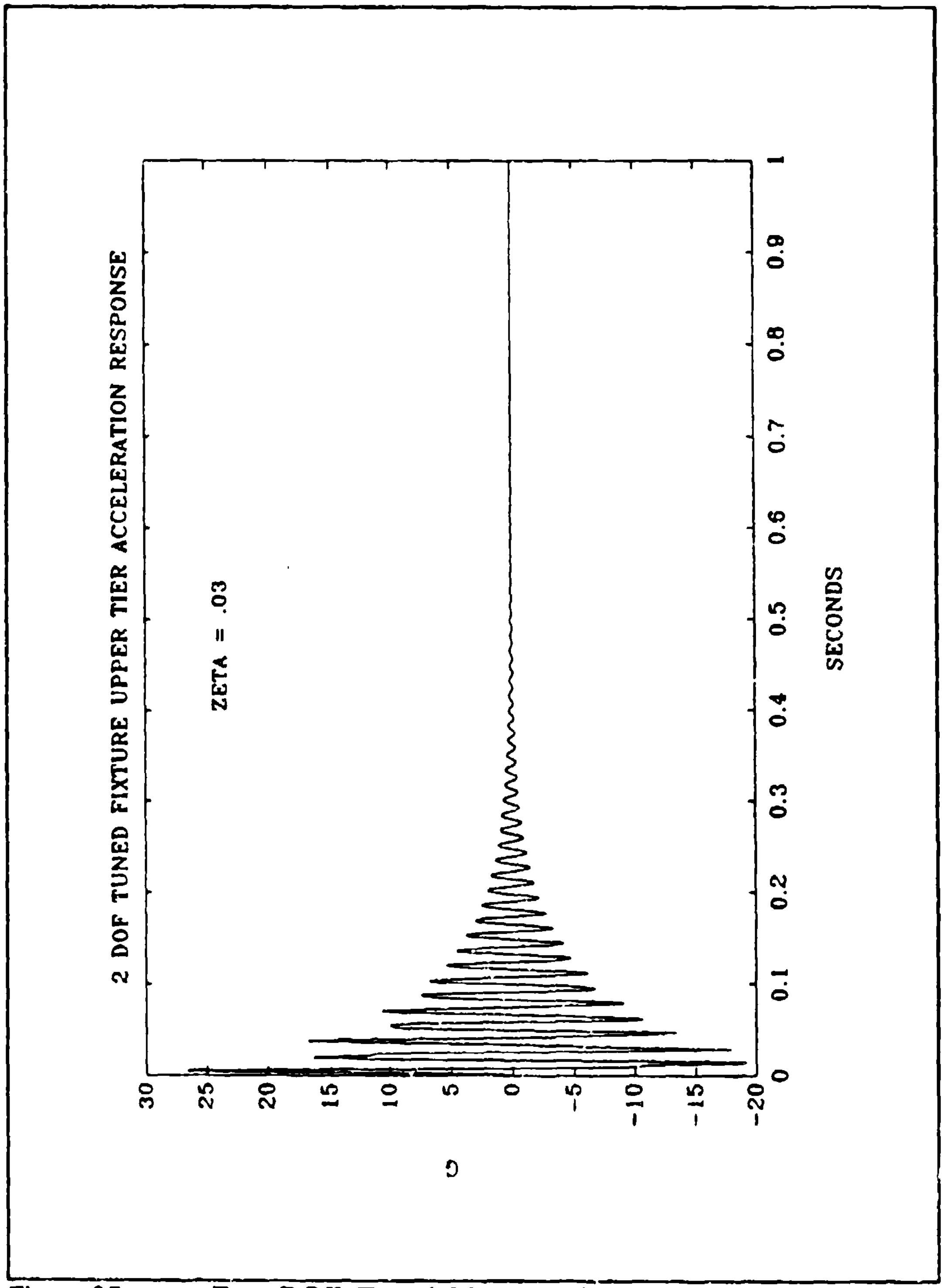

Figure 35. Two DOF Tuned Mounting Fixture Upper Tier Damped Acceleration Response. Zeta = .03.

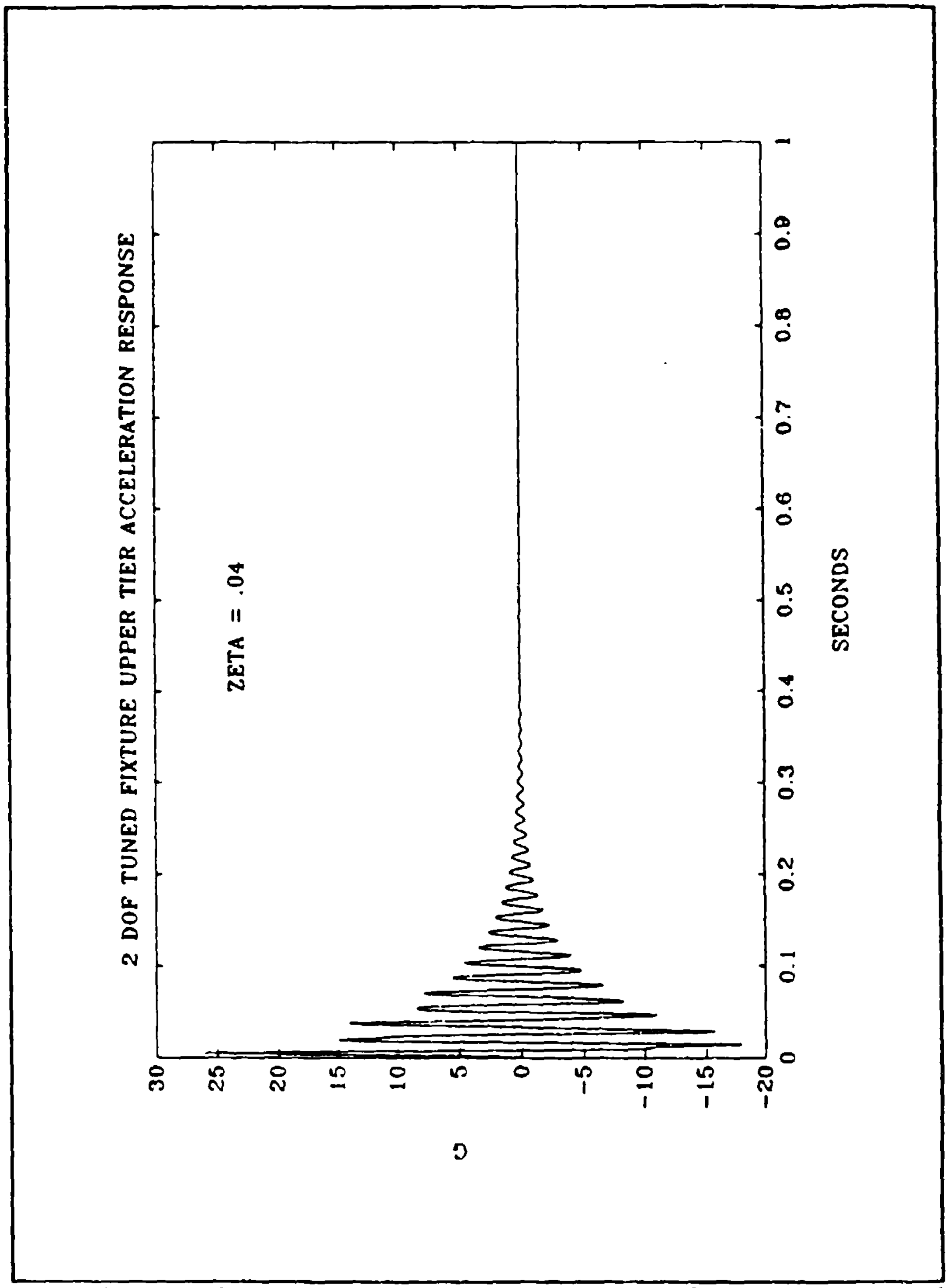

Figure 36. Two DOF Tuned Mounting Fixture Upper Tier Damped Acceleration Response. Zeta = .04.

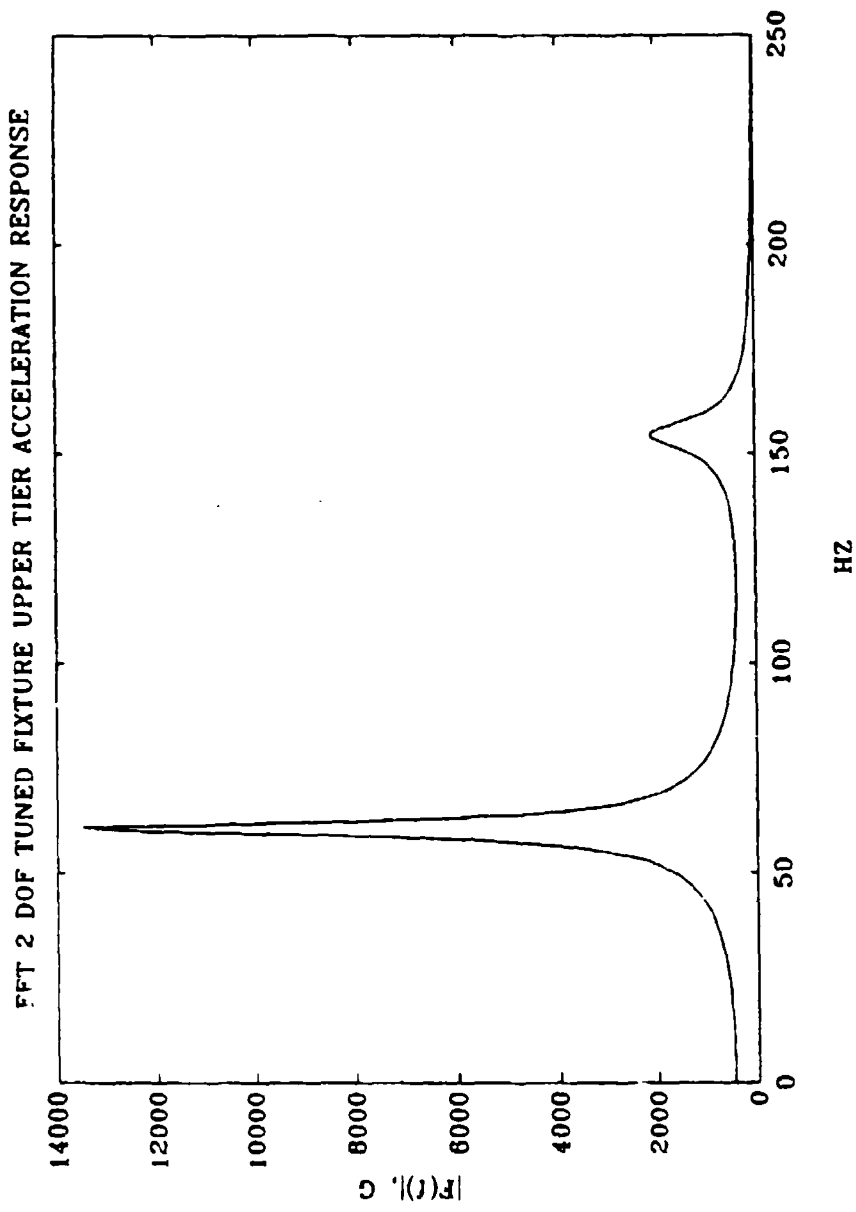

Figure 37. Fourier Transform of Upper Tier Acceleration Waveform. Zeta = .02.

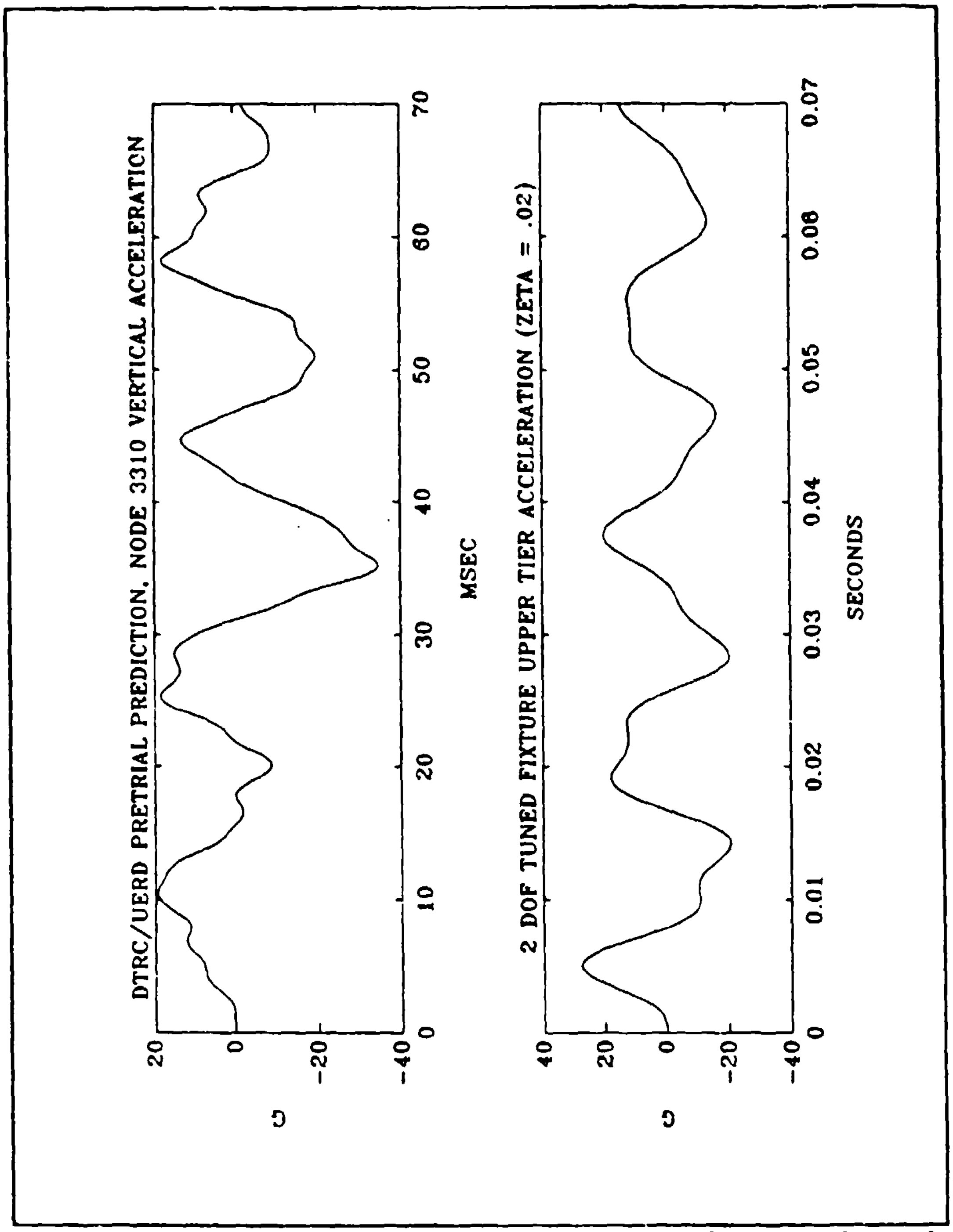

Figure 38. DTRC/UERD Predicted Acceleration Waveform for Node 3310, Radar Receiver/Transmitter, and Two DOF Tuned Mounting Fixture Upper Tier Acceleration Waveform for First 70 Milliseconds.

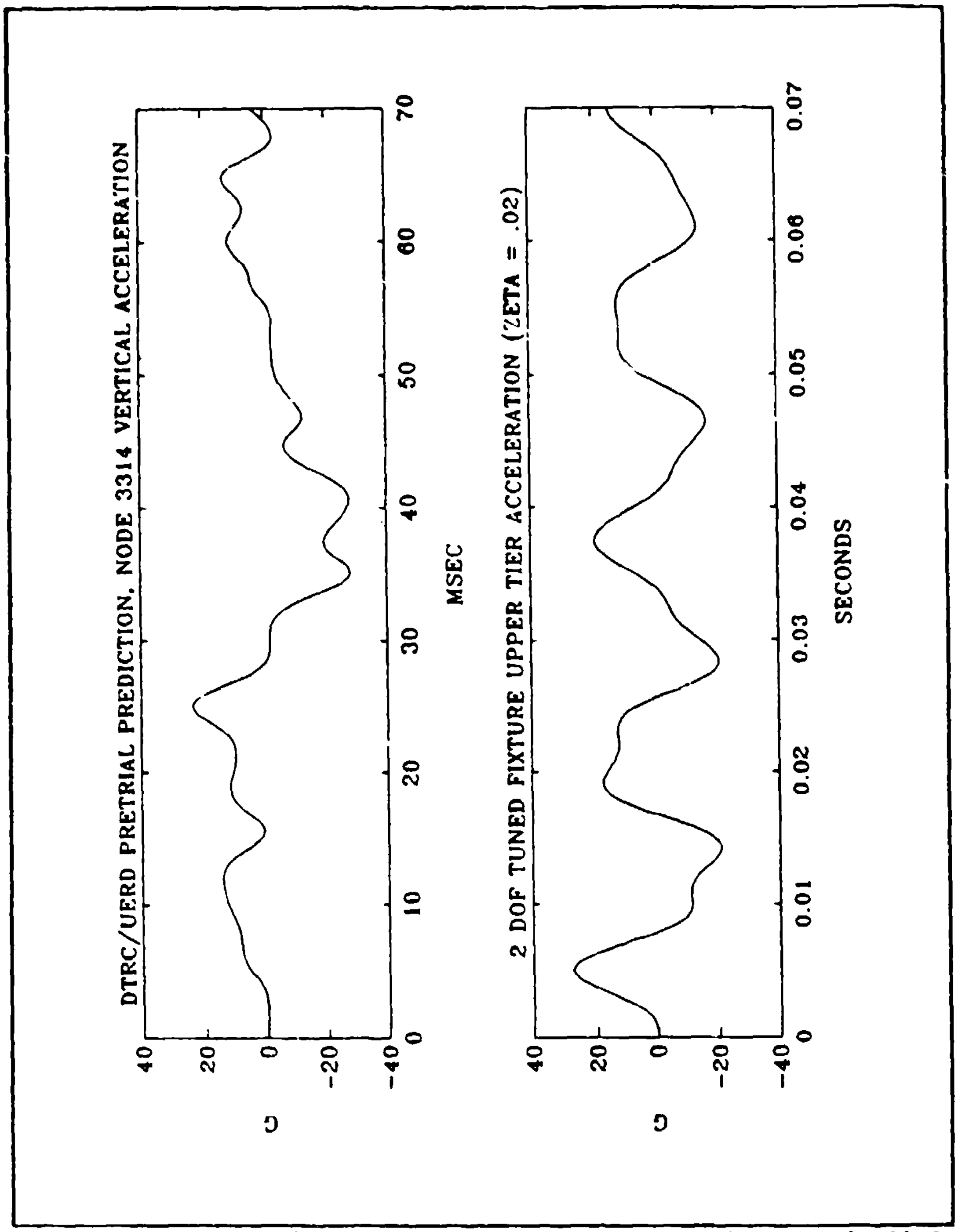

Figure 39. DTRC/UERD Predicted Acceleration Waveform for Node 3314, Beam Programmer, and Two DOF Tuned Mounting Fixture Upper Tier Acceleration Waveform for First 70 Milliseconds.

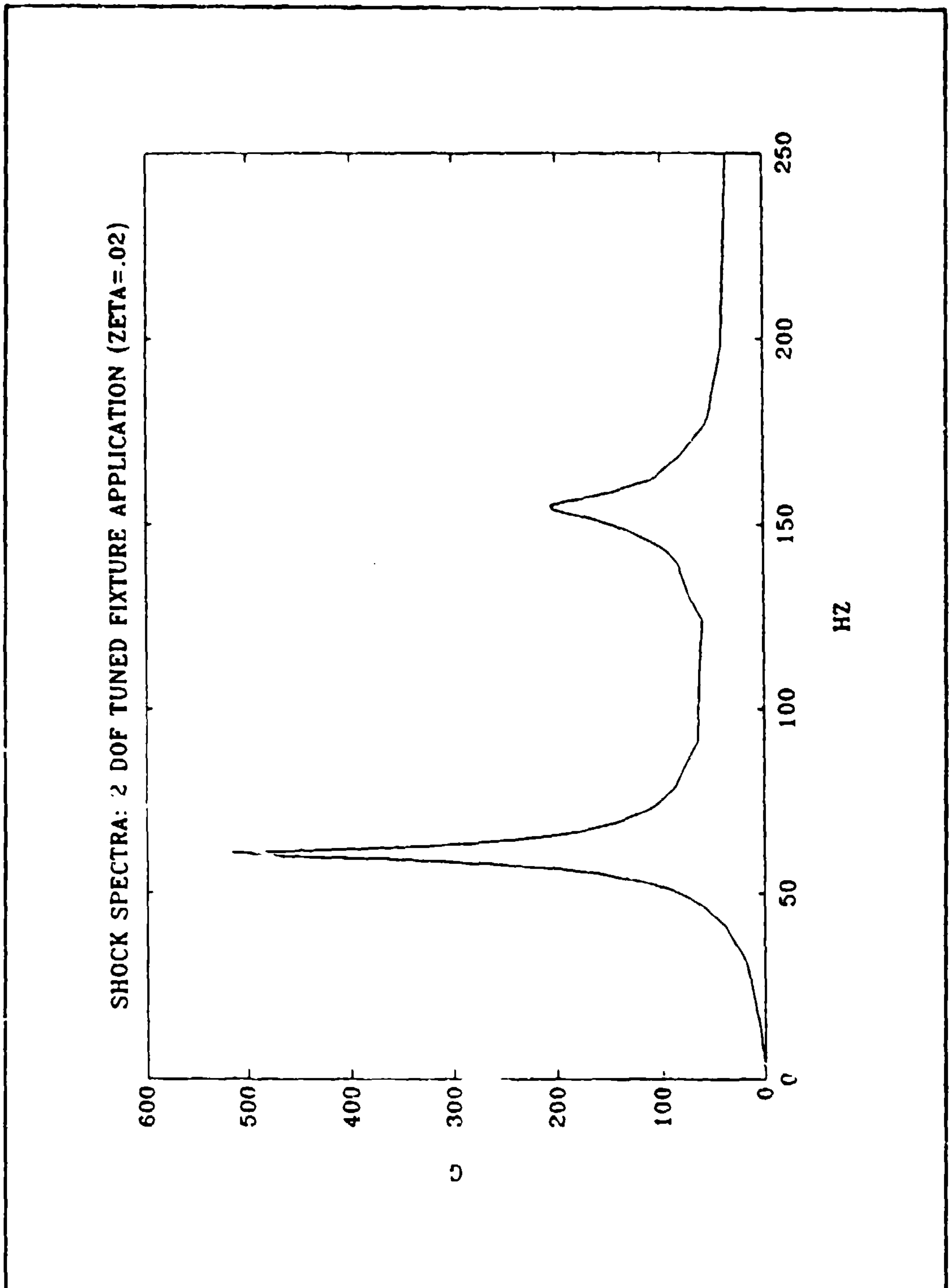

Figure 40. Shock Spectra Using Two DOF Tuned Mounting Fixture Upper Tier Acceleration as Excitation.

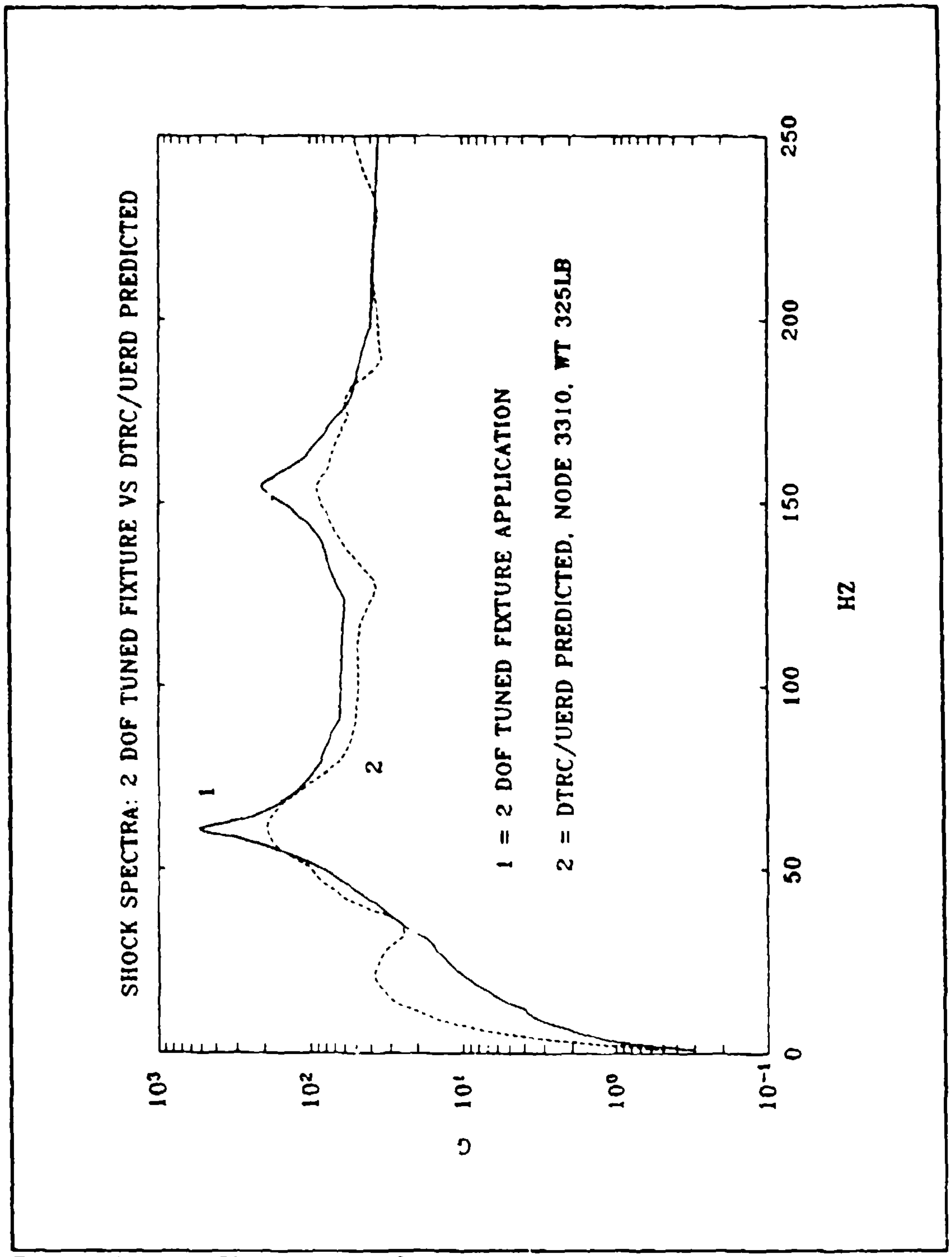

Figure 41. Shock Spectra Comparison. DTRC/UERD Predicted for Node 3310, Radar Receiver/Transmitter, Foundation Excitation and Upper Tier of Two DOF Tuned Mounting Fixture Excitation.

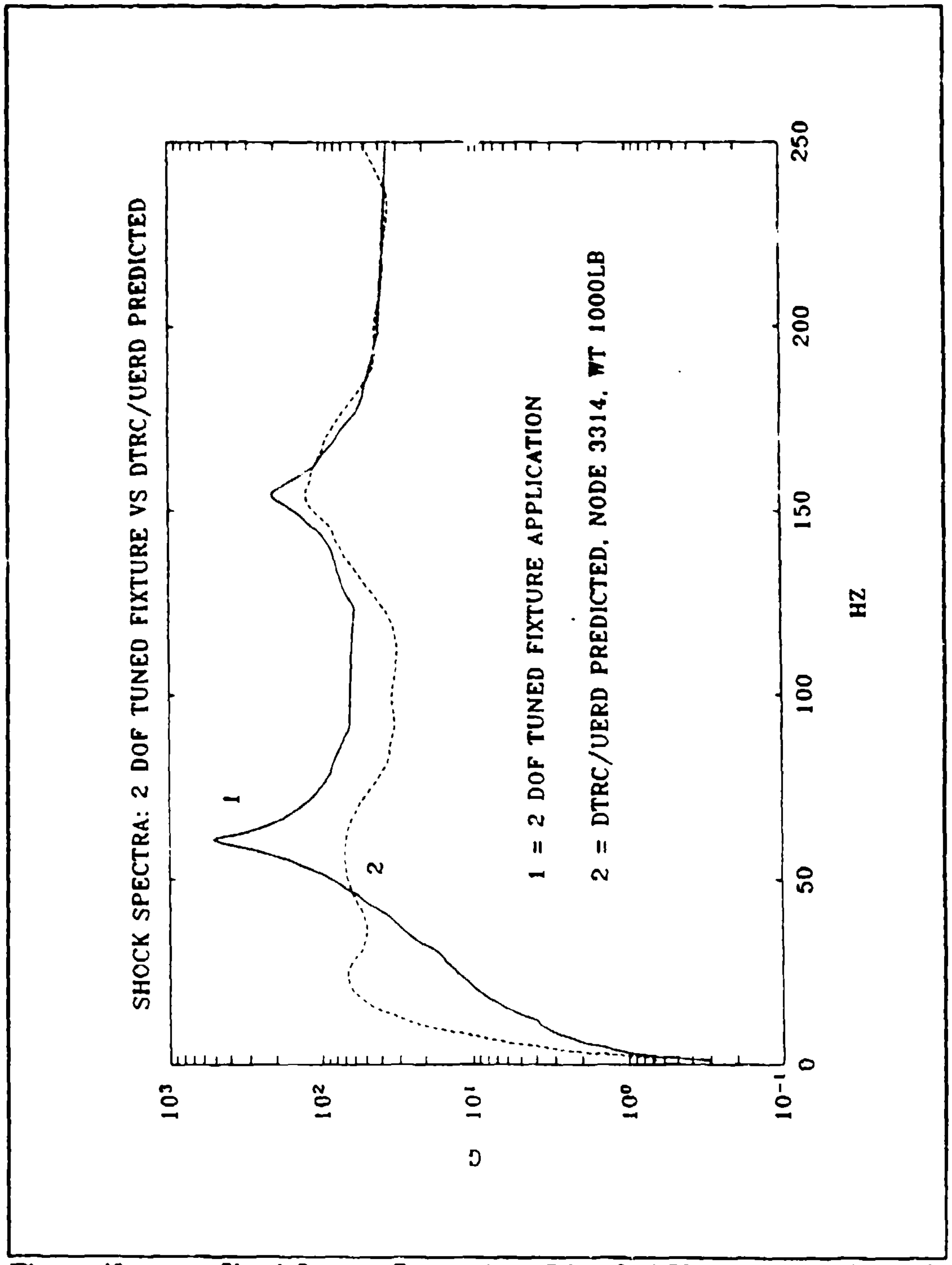

Figure 42. Shock Spectra Comparison. DTRC/UERD Predicted for Node 3314, Beam Programmer, Foundation Excitation and Upper Tier of Two DOF Tuned Mounting Fixture Excitation.

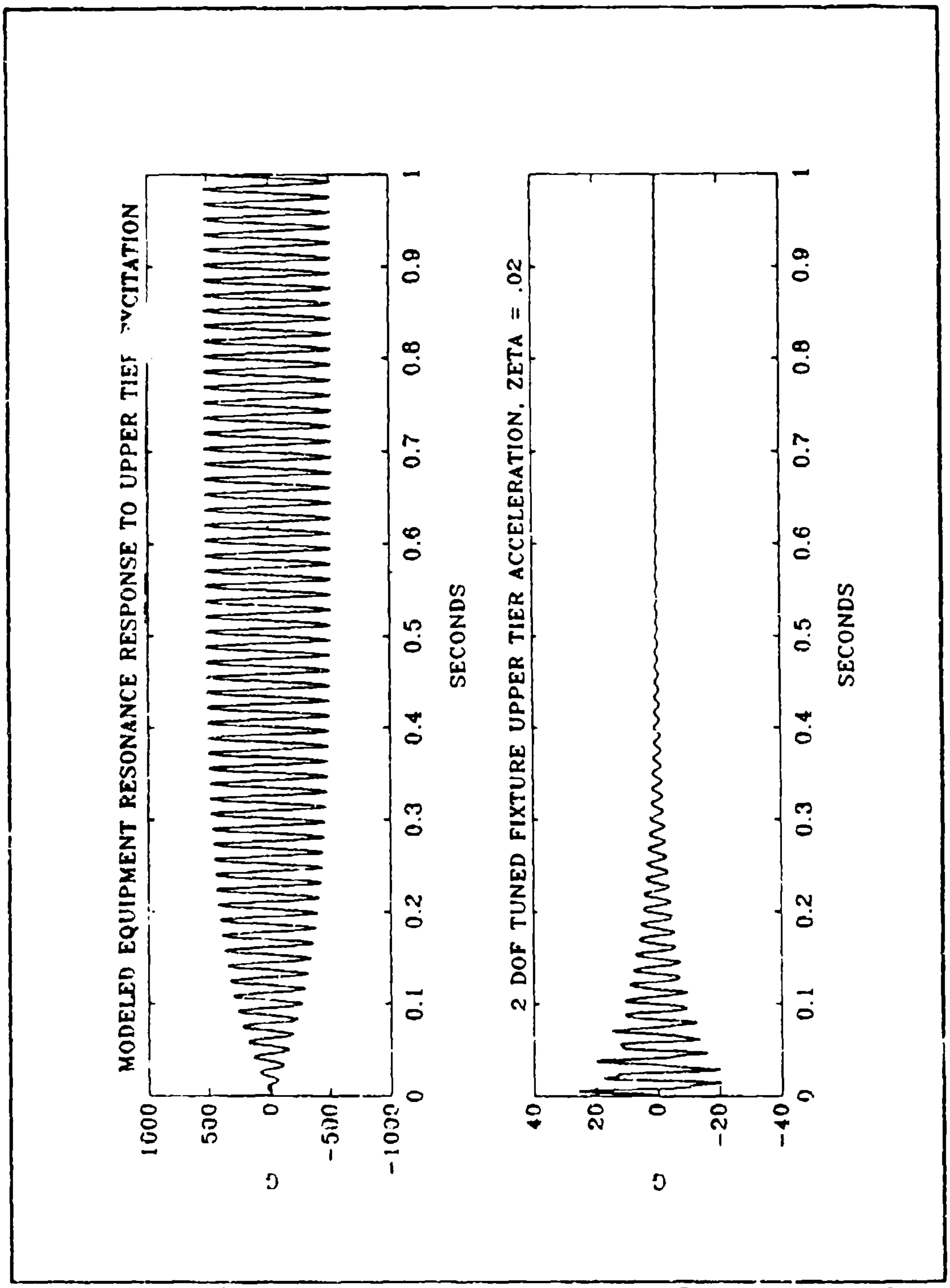

Figure 43. Resonance Response of an Equipment Subjected to the Upper Tier's A celeration Excitation.

V. CONCLUSIONS AND RECOMMENDATIONS

The DTRC/UERD DDG-51 Class Pre-Shock Trial Shock Analyses of three combat systems equipments ranging from 325 to 4600 lbs revealed shock characteristics that cannot be simulated on the MWSM unless special tuned mounting fixtures are implemented. Both the single DOF and two DOF tuned mounting fixtures can be used to produce a more realistic shock phenomena when test items are shock qualified under the simulated conditions inherent with the MWSM. The proposals below are presented for consideration in advancing the practice of shock qualifying surface ship combat systems equipments.

For heavy weight equipment, in the range of about 4600 lbs, primarily low frequency foundation excitation can be expected to dominate, yielding acceleration waveforms consisting of one dominant frequency component in the vicinity of about 23 Hertz. A single DOF tuned mounting fixture used to affix a test item to the MWSM will provide the shock characteristics observed in this situation. The Soft Deck Simulator, developed by the Naval Underwater Systems Center for the shock qualification of submarine combat systems equipments, is proposed for use in the shock qualification of heavy weight range surface ship combat systems equipments which display the foundation excitations described above. Analysis of the ship class pre-shock trial data will reveal which equipments are likely to experience support foundation excitations that can be simulated by such a device.

For low and medium weight equipments, in the ranges of 325 and 1000 lbs, respectively, the DTRC/UERD analyses revealed a more complex waveform in which a two DOF tuned mounting fixture must be used. It is proposed that the two DOF tuned mounting fixture, described in chapter IV, be used to simulate the dominant shock characteristics revealed in the ship class pre-shock trial analyses for low and medium weight range combat systems equipments. It is strongly recommended that the below proposals be implemented to advance the practice of shock qualifying surface ship combat systems equipments:

- the design and construction of a two DOF tuned mounting fixture for the MWSM
- a sensitivity/calibration study of tier weights and frequency response characteristics of the two DOF tuned mounting fixture for given MWSM hammer heights
- an investigation of damping characteristics in shock wave propagation through shipboard structures to advance the study of pre-shock trial analyses
- the development of a weight category, low and medium, combat system equipment Shock Spectra library, for various vessel classes, as a reference in parameter selection for the two DOF tuned mounting fixture

Implementation of the above proposals will produce a MWSM two DOF tuned mounting fixture with the ancillary information necessary to provide a more realistic shock phenomena when low and medium weight range combat systems equipments are shock qualified in a simulated environment, such as with the MWSM.

This overall study demonstrated that the use of tuned mounting fixtures on the MWSM can be used to accurately simulate the shock characteristics that may be observed in actual ship shock trials. Implementing these devices in the U.S. Navy Shock Qualification Program for Surface Ships will promote system reliability in times that are crucial to vessel survivability--the time of an underwater attack.

APPENDIX A

The DTRC/UERD DDG-51 Class Pre-Shock Trial Analyses Shock Spectra for three modeled combat systems equipment located on the 0-3 Level:

TABLE A-1

EQUIPMENT	FINITE ELEMENT NODE	EQUIPMENT WEIGHT LBS
RADAR RECEIVER/ TRANSMITTER	3310	325
BEAM PROGRAMMER	3314	1000
RADIO FREQUENCY AMPLIFIER	3320	4610

Figures A-1 through A-3 reveal the three Shock Spectra orientations, fore/aft, athwartships and vertical for each equipment listed above. By comparison, the vertical orientation presents the severest shock phenomena experienced by each equipment.

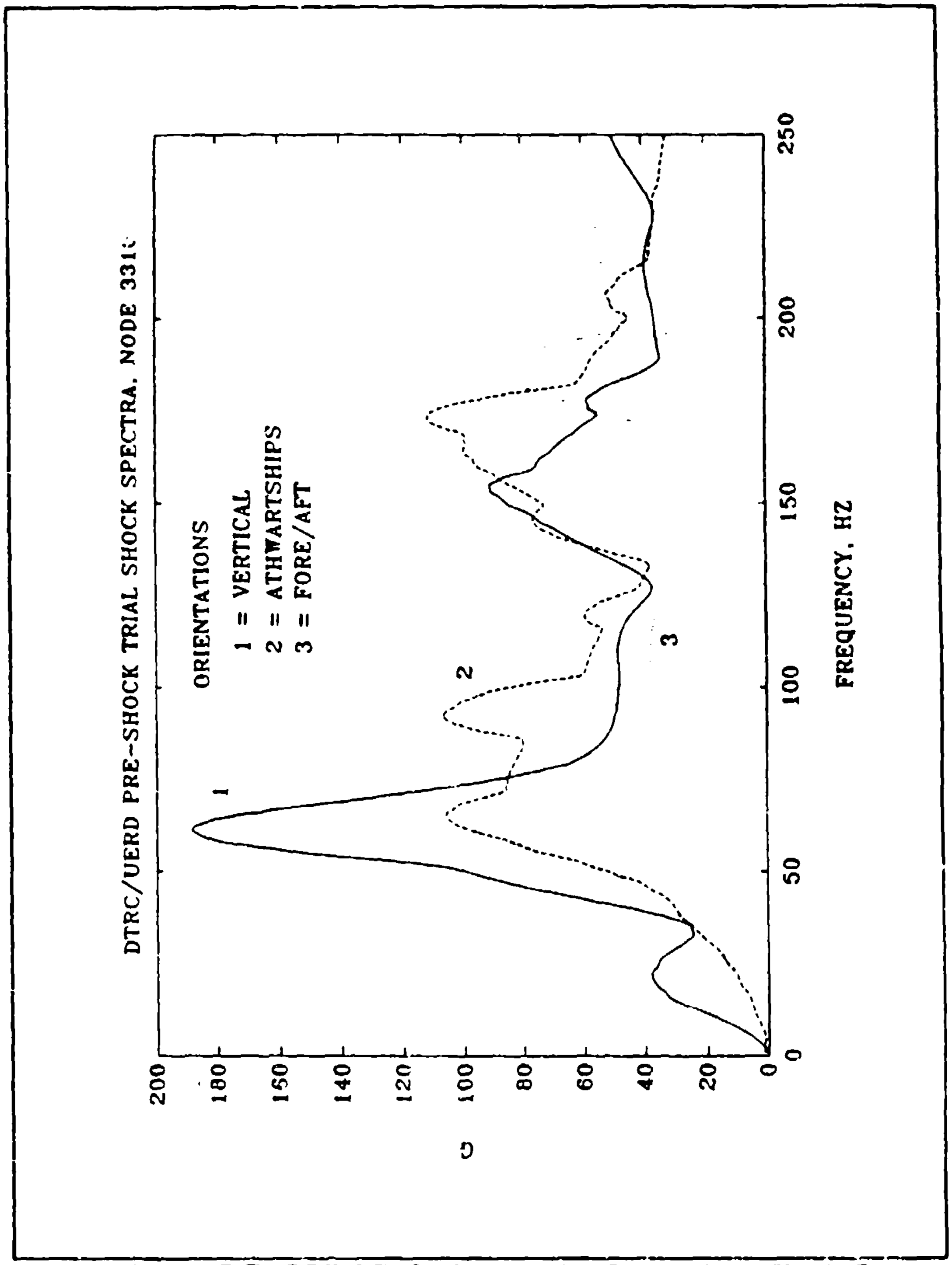

Figure A-1. DTRC/UERD Radar Receiver/Transmitter Shock Spectra. Courtesy of Costanzo and Murray (1991).

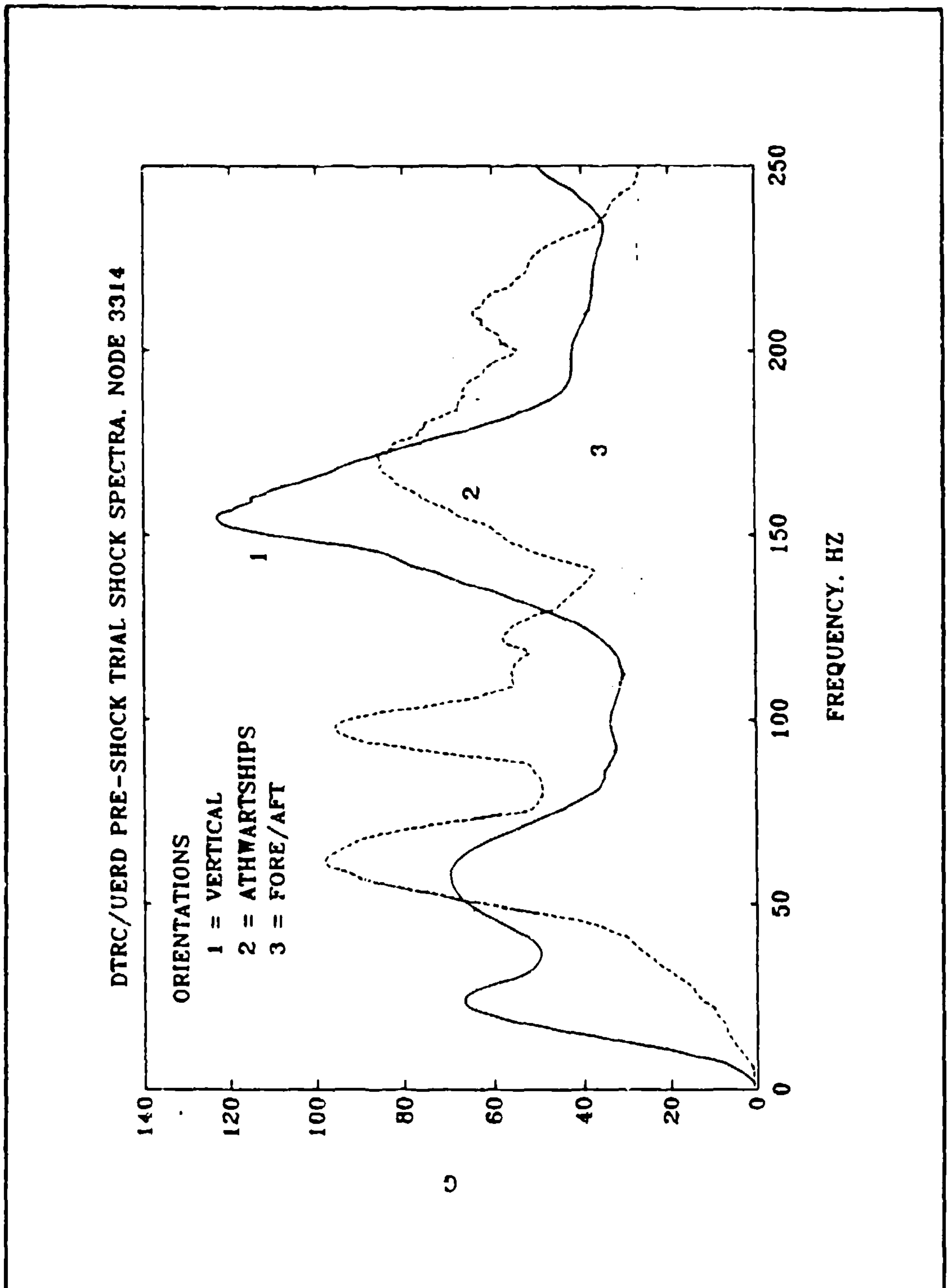

Figure A-2. DTRC/UERD Beam Programmer Shock Spectra. Courtesy of Costanzo and Murray (1991).

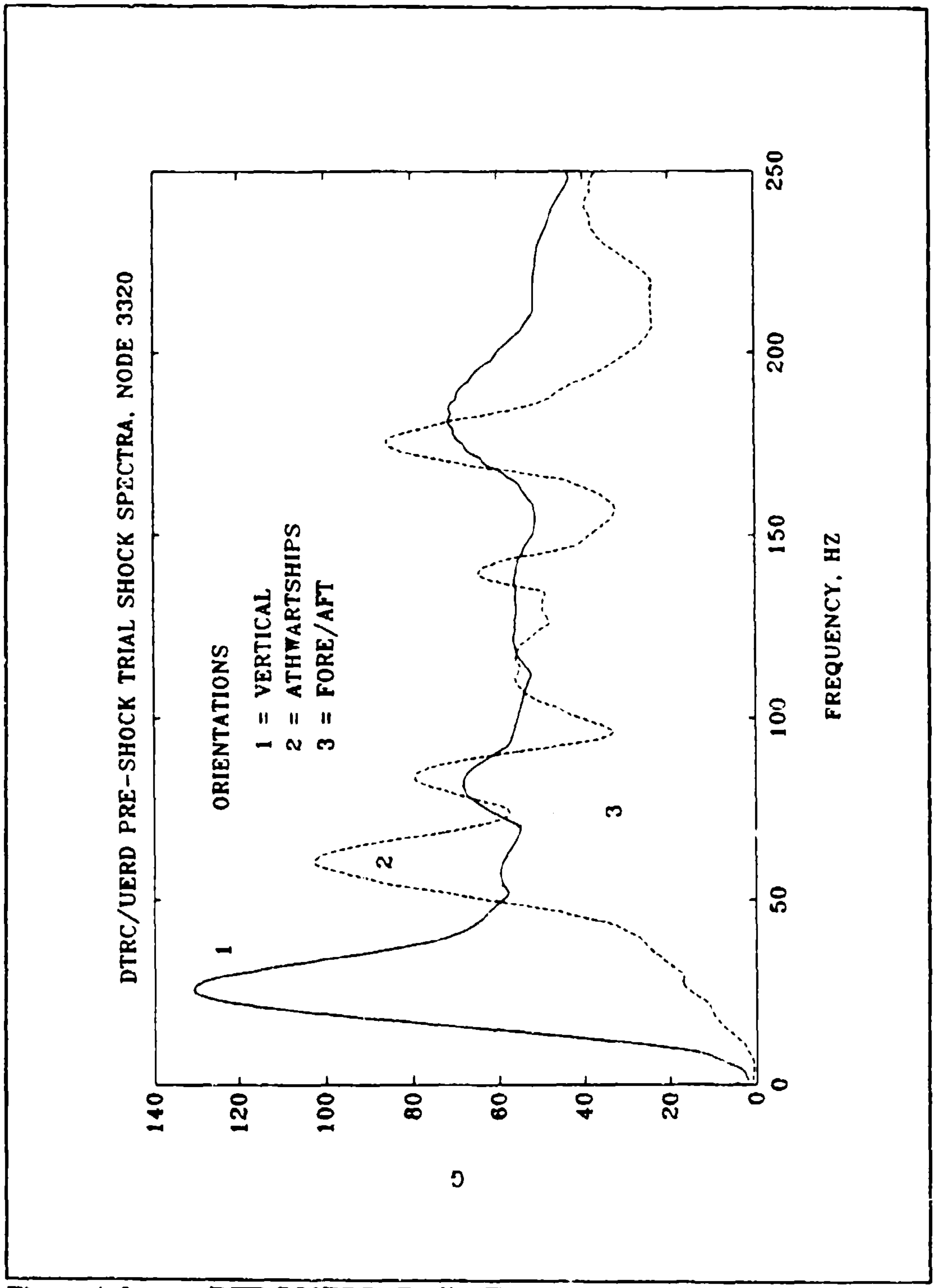

Figure A-3. DTRC/UERD Radio Frequency Amplifier Shock Spectra. Courtesy of Costanzo and Murray (1991).

APPENDIX B

```
C LT RANDALL CORBELL
C NAVAL POSTGRADUATE SCHOOL
C SHOCK QUALIFICATION OF COMBAT SYSTEMS EQUIPMENTS USING TUNED
C MOUNTING FIXTURES ON THE U.S. NAVY MEDIUMWEIGHT SHOCK MACHINE

C PROGRAM:  SDOF TUNED FIXTURE RESPONSE TO  MWSM  HALF-SINE
C           ACCELERATION PULSE

C REF:  (A) CLEMENTS,E.W.,"SHIPBOARD SHOCK AND NAVY DEVICES FOR
C           ITS SIMULATION",NRL REPORT 7396, 1972.
C       (B) CRAIG,R,"STRUCTURAL DYNAMICS, AN INTRODUCTION TO
C           COMPUTER METHODS", JOHN WILEY AND SONS, 1981.
C       (C) COSTANZO/MURRAY, "DTRC/UERD DDG-51 CLASS PRESHOCK
C           TRIAL ANALYSIS PRELIMINARY REPORT",1991.
C       (D) COSTANZO/CLEMENTS,"MWSM CALIBRATION DATA",1988.

C     THIS PROGRAM COMPUTES THE RESPONSE OF A SINGLE DEGREE OF
C     FREEDOM MASS-DAMPER-SPRING SYSTEM TO A BASE EXCITEMENT HALF-SINE
C     ACCELERATION PULSE OF 1MSEC DURATION, AS CHARACTERISTIC WITH THE
C     MWSM NOTED IN REF(A). REF(B) WAS USED FOR NUMERICAL ANALYSIS
C     PROGRAM STRUCTURE. THE ACCELERATION RESPONSE OF THE SDOF TUNED
C     FIXTURE IS THE BASE EXCITATION A WHICH COMBAT SYSTEM EQUIPMENT
C     WILL EXPERIENCE IN UNDEX, AS SIMULATED ON THE MWSM.
C
C     SELECTION OF THE NATURAL FREQUENCY OF THE SYSTEM IS
C     BASED ON ANALYSIS OF FFT/SHOCK SPECTRAL INFORMATION PROVIDED IN
C     REF (C). SELECTION OF DAMPING VALUES WERE OBTAINED AFTER ANALYSIS
C     OF MWSM CALIBRATION DATA, NOTED IN REF (D).

C     DECLARATIONS...
          PARAMETER(MAX=10000)
C     ARRAYS...
          REAL        A(3,3),L(3),UO(3),UI(3),U(3)
          DIMENSION TIME  (MAX),ZBASE (MAX),TUNFIXACC(MAX),
     :     RELDIS(MAX),RELVEL(MAX),RELACC   (MAX)
C     VARIABLES...
      REAL ZETA,DELT,FREQ,E,F,G,WN,WN2,Z
C     INTEGERS...
      INTEGER NSTEP

C     PRINT STATEMENTS FOR INPUT...
          PRINT*,'INPUT VISCOUS DAMPING COEFFICIENT,ZETA.'
          READ*,  ZETA
C     PARAMETERS USED...
```

```
      FREQ   =    23.
      DELT   =    0.0001
      NUMSTEP =  1000C
      UO(1)  =    0.
      UO(2)  =    0.
      REWIND 10
      OPEN(10,FILE='ANVILACCHV.DAT',STATUS='OLD')
      OPEN(20,FILE='SDOFTUNFIX.DAT',STATUS='NEW')

C     READ DATA FROM ANVIL ACCELERATION RECORD, NOTE ACCELERATION IN
G'S....
      DO  100 I=1,NUMSTEP
          READ(10,*) TIME(I),ZBASE(I)
100   CONTINUE

C     SOLVE EQN MOTION FOR INITIAL ACCELERATION: KNOW INITIAL
C     MASS,ZETA,FREQ,CONDITIONS...
      WN       = 2.*4.*ATAN(1.)*FREQ
      WN2      =(2.*4.*ATAN(1.)*FREQ)**2
      UO(3)    =-ZBASE(1)*32.2-(2.*ZETA*WN)*UO(2)-WN2*UO(1)
      UI(1)    =UO(1)
      UI(2)    =UO(2)
      UI(3)    =UO(3)
      RELDIS(1) =UO(1)
      RELVEL(1) =UO(2)
      RELACC(1) =UO(3)

C     COMPUTE PARAMETERS FOR OPERATOR MATRICES...
      E=WN2*(DELT**2)
      F=2.*ZETA*WN*DELT
      G=(1./(1.+(F/2.)+(E/4.)))

C     LOAD AMPLIFICATION AND LOAD VECTOR MATRIX...
      A(1,1)=G*(1.+(F/2.))
      A(1,2)=G*DELT*(1.+(F/4.))
      A(1,3)=G*DELT**2/4.
      A(2,1)=G*(-E/(2.*DELT))
      A(2,2)=G*(1.-E/4.)
      A(2,3)=G*DELT/2.
      A(3,1)=G*(-E/(DELT**2))
      A(3,2)=G*(-1.)*(E+F)/(DELT)
      A(3,3)=G*(-1.)*((F/2.)+(E/4.))
      L(1)  =G*(DELT**2)/4.
      L(2)  =G*DELT/2.
      L(3)  =G

C     BEGIN ITERATIONS...
      DO 200 J=1,NUMSTEP
      Z   =-ZBASE(J+1)*32.2
      U(1)= A(1,1)*UI(1)+A(1,2)*UI(2)+A(1,3)*UI(3)+Z*L(1)
```

```
      U(2)= A(2,1)*UI(1)+A(2,2)*UI(2)+A(2,3)*UI(3)+Z*L(2)
      U(3)= A(3,1)*UI(1)+A(3,2)*UI(2)+A(3,3)*UI(3)+Z*L(3)

C     SAVE THE DIS,VEL,ACC ARRAY VALUES FOR THIS ITERATION...
      RELDIS(J+1) = U(1)
      RELVEL(J+1) = U(2)
      RELACC(J+1) = U(3)

C     RESET THE ITERATION MATRIX FOR THE NEXT TIME
C     ITERATION...
      DO 225 K=1,3
         UI(K)=U(K)
  225 CONTINUE
  200 CONTINUE
      DO 250 I=1,NUMSTEP
         TUNFIXACC(I)=(RELACC(I)/32.2)+ZBASE(I)
         WRITE(20,*)TIME(I),TUNFIXACC(I)
  250 CONTINUE
      END
```

APPENDIX C

```
C    LT RANDALL CORBELL
C    NAVAL POSTGRADUATE SCHOOL
C    SHOCK QUALIFICATION OF COMBAT SYSTEMS EQUIPMENTS USING TUNED
C    MOUNTING FIXTURES ON THE U.S. NAVY MEDIUMWEIGHT SHOCK MACHINE

C    PROGRAM:    SHOCK SPECTRA FORMULATION USING TUNED MOUNTING
FIXTURE
C                ACCELERATION EXCITATION

C    REF:    (A) CLEMENTS,E.W.,"SHIPBOARD SHOCK AND NAVY DEVICES FOR
C                ITS SIMULATION",NRL REPORT 7396, 1972.
C            (B) CRAIG,R,"STRUCTURAL DYNAMICS, AN INTRODUCTION TO
C                COMPUTER METHODS", JOHN WILEY AND SONS, 1981.
C            (C) COSTANZO/MURRAY, "DTRC/UERD DDG-51 CLASS PRESHOCK
C                TRIAL ANALYSES PRELIMINARY REPORT",1991.

C    THIS PROGRAM COMPUTES SHOCK SPECTRA FOR AN UNDAMPED A
C    SINGLE DEGREE OF FREEDOM MASS-SPRING SYSTEM SUBJECTED TO A BASE
C    EXCITEMENT, THE TUNED MOUNTING FIXTURE ACCELERATION ON THE
MWSM
C    IN REF(A). REF(B) WAS USED FOR NUMERICAL ANALYSIS PROGRAM
C    STRUCTURE. THE RESULTS ARE COMPARED WITH DATA IN REF (C).

C    DECLARATIONS...
C    PARAMETER
         PARAMETER(MAX=10000)
C    ARRAYS
         REAL    A(3,3),L(3),UO(3),UI(3),U(3)
              DIMENSION TIME   (MAX), ZBASE   (MAX), FREQ   (MAX),
         :         RELDIS  (MAX), RELVEL  (MAX), RELACC  (MAX),
         :         EQUIPACC(MAX), EQMAXACC(MAX), ACCINPUT(MAX),
         :         ACCX1   (MAX), ACCX2   (MAX)
C    VARIABLES
         REAL ZETA,DELF,DELT,DELT1,DELT2,TLOAD,E,F,G,W,MAXEQACC
         INTEGER NUMSTEP,NUMFREQ
C    PARAMETERS USED...
         ZETA    =    0.
         DELT    =    .0001
         NUMSTEP = 10000
         UO(1)   =    0.
         UO(2)   =    0.
         DELF    =    1.
         NUMFREQ =    250
         REWIND 10
```

```
      OPEN(10,FILE='TIMEREC    .DAT',STATUS='OLD')
      OPEN(20,FILE='FREQREC    .DAT',STATUS='NEW')
C     READ AND LOAD THE BASE ACCELERATION ARRAY FROM THE DATA
C     FILES......
C     NOTE:   (1) DELT EQUAL .0001 SECONDS
C             (2) ACCELERATION IN G'S
C             (3) FOR 2DOF TUNE FIXTURE, ACCX1 IS BASE ACCELERATION
C     INPUT...
      DO 100 I=1,NUMSTEP
C             READ(10,*) TIME(I),ACCX1(I)
              READ(10,*) TIME(I),ACCX1(I),ACCX2(I)
100   CONTINUE
C     LOAD THE INPUT ACCELERATION ARRAY WITH THE BASE ACCELERATION IN
C     G'S...
      DO 150 I=1,NUMSTEP
      ACCINPUT(I)=ACCX1(I)*32.2
150    CONTINUE
C     BEGIN ITERATIONS FOR THE NATURAL FREQUENCIES BEGINNING WITH
C     DELF...
      DO 200 I=1,NUMFREQ
      FREQ(I)=I*DELF

C     SOLVE E.O.M FOR INITIAL ACCELERATION WITH KNOWN PARAMETERS...
      UO(3)  =-ACCINPUT(I)
     :       -(2.*ZETA*(2.*(4.*ATAN(1.))*FREQ(I)))*UO(2)
     :       -((2.*(4.*ATAN(1.))*FREQ(I))**2)     *UO(1)

C     LOAD INITIAL VALUES FOR ITERATION AND COMPONENT ARRAYS....
      UI(1)      =UO(1)
      UI(2)      =UO(2)
      UI(3)      =UO(3)
      RELDIS(1)  =UO(1)
      RELVEL(1)  =UO(2)
      RELACC(1)  =UO(3)

C         BEGIN TIME LOOP...
          DO 300 J=1,NUMSTEP

C         SET TIME INTERVAL FOR PHASE 1 OR 2 OF TIME HISTORY......
C         IF((J*DELT1).LE.TLOAD)THEN
C             DELT = DELT1
C         ELSE
C             DELT = 1./(20.*FREQ(I))
C                 DELT2 = DELT
C         ENDIF

C         COMPUTE PARAMETERS FOR OPERATOR MATRICES...
          E=((2.*(4.*ATAN(1.))*FREQ(I))**2)*(DELT**2)
          F=(2.*ZETA*(2.*(4.*ATAN(1.))*FREQ(I)))*DELT
          G=(1./(1.+(F/2.)+(E/4.)))
```

```
C         LOAD AMPLIFICATION AND LOAD VECTOR MATRIX...
          A(1,1)=G*(1.+(F/2.))
          A(1,2)=G*DELT*(1.+(F/4.))
          A(1,3)=G*DELT**2/4.
          A(2,1)=G*(-E/(2.*DELT))
          A(2,2)=G*(1.-E/4.)
          A(2,3)=G*DELT/2.
          A(3,1)=G*(-E/(DELT**2))
          A(3,2)=G*(-1.)*(E+F)/(DELT)
          A(3,3)=G*(-1.)*((F/2.)+(E/4.))
          L(1)  =G*(DELT**2)/4.
          L(2)  =G*DELT/2.
          L(3)  =G

C         BEGIN ITERATIONS...
          W   =-ACCINPUT(J+1)
          U(1)= A(1,1)*UI(1)+A(1,2)*UI(2)+A(1,3)*UI(3)+W*L(1)
          U(2)= A(2,1)*UI(1)+A(2,2)*UI(2)+A(2,3)*UI(3)+W*L(2)
          U(3)= A(3,1)*UI(1)+A(3,2)*UI(2)+A(3,3)*UI(3)+W*L(3)

C         SAVE THE DIS,VEL,ACC ARRAY VALUES FOR THIS
C         ITERATION.....
          RELDIS(J+1) = U(1)
          RELVEL(J+1) = U(2)
          RELACC(J+1) = U(3)

C         RESET THE ITERATION MATRIX FOR THE NEXT TIME
C         ITERATION...
          DO 350 K=1,3
               UI(K)=U(K)
      350 CONTINUE
      300 CONTINUE
C         END OF TIME ITERATION LOOP...

C         COMPUTE THE EQUIPMENT ACCELERATION NOW THAT THE RELATIVE
C         AND BASE ACCELERATION ARE KNOWN...
          DO 250 J=1,NUMSTEP
               EQUIPACC(J)=RELACC(J)+ACCINPUT(J)
      250 CONTINUE

C     SELECT THE ABSOLUTE MAXIMUM FROM EQUIPACC ARRAY AND LOAD
C     THAT VALUE INTO EQMAXACC ARRAY FOR THIS FREQUENCY ITERATE...
C     CONVERT TO G'S...
          MAXEQACC=ABS(EQUIPACC(1))
          DO 275 J=2,NUMSTEP
               IF(ABS(EQUIPACC(J)).GT.MAXEQACC)THEN
                    MAXEQACC=ABS(EQUIPACC(J))
               ENDIF
      275 CONTINUE
          EQMAXACC(I)=MAXEQACC/32.2
```

```
  200 CONTINUE
C     END OF FREQUENCY LOOP...

C     UNITS FOR FREQUENCY RECORD ARE HZ AND G'S...
      DO 400 I=1,NUMFREQ
          WRITE(20,*) FREQ(I),EQMAXACC(I)
  400 CONTINUE
      END
```

APPENDIX D

```
C    LT RANDALL CORBELL
C    NAVAL POSTGRADUATE SCHOOL
C    SHOCK QUALIFICATION OF COMBAT SYSTEMS EQUIPMENTS USING TUNED
C    MOUNTING FIXTURES ON THE U.S. NAVY MEDIUMWEIGHT SHOCK MACHINE

C    PROGRAM:   2 DOF TUNED FIXTURE RESPONSE TO  MWSM  HALF-SINE
C               ACCELERATION PULSE

C    REF:    (A) CLEMENTS,E.W.,"SHIPBOARD SHOCK AND NAVY DEVICES FOR
C                ITS SIMULATION",NRL REPORT 7396, 1972.
C            (B) CRAIG,R,"STRUCTURAL DYNAMICS, AN INTRODUCTION TO
C                COMPUTER METHODS", JOHN WILEY AND SONS, 1981.
C            (C) MEIROVITCH,L.,"ELEMENTS OF VIBRATIONAL ANALYSIS",
C                MCGRAW-HILL, 1986.
C            (D) COSTANZO/MURRAY, "DTRC/UERD DDG-51 CLASS PRESHOCK
C                TRIAL ANALYSIS PRELIMINARY REPORT",1991.
C            (E) COSTANZO/CLEMENTS,"MWSM CALIBRATION DATA",1988.

C        THIS PROGRAM COMPUTES THE RESPONSE OF A TWO DEGREE OF
C    FREEDOM MASS-DAMPER-SPRING SYSTEM TO A BASE EXCITEMENT HALF-SINE
C    ACCELERATION PULSE OF 1MSEC DURATION, AS CHARACTERISTIC WITH THE
C    MWSM NOTED IN REF(A).  REF(B) AND (C) WERE USED FOR NUMERICAL
C    ANALYSIS AND PROGRAM STRUCTURE. THE UPPER TIER ACCELERATION
C    RESPONSE OF THE 2 DOF TUNED FIXTURE IS THE BASE EXCITATION A
C    COMBAT SYSTEM EQUIPMENT WOULD EXPERIENCE IN UNDEX, AS SIMULATED
C    ON THE MWSM.
C
C    SELECTION OF THE COUPLED NATURAL FREQUENCIES OF THE SYSTEM ARE
C    BASED ON ANALYSIS OF FFT/SHOCK SPECTRAL INFORMATION PROVIDED IN
C    REF (D). SELECTION OF DAMPING VALUES WERE OBTAINED AFTER ANALYSIS
C    OF MWSM CALIBRATION DATA, NOTED IN REF (E).

C    AS PER REF(B), A 2 DOF MASS SPRING SYSTEM SUBJECTED TO BASE
C    MOTION CAN BE EXPRESSED IN TERMS OF RELATIVE MOTIONS: DISP, VEL,
C    ACC AS FOLLOWS:

C    LET:    Y1=X1-Z  WHERE X1 IS MASS1 COORDINATE AND Z IS BASE
C                     COORDINATE
C            Y2=X2-Z  WHERE X1 IS MASS2 COORDINATE AND Z IS BASE
C                     COORDINATE
C            M1 = MASS1
C            M2 = MASS2
C            K1 = SPRING STIFFNESS 1
C            K2 = SPRING STIFFNESS 2
```

```
C              F1 = UNCOUPLED NATURAL FREQUENCY 1
C              F2 = UNCOUPLED NATURAL FREQUENCY 2
C              FN1= COUPLED NATURAL FREQUENCY 1
C              FN2= COUPLED NATURAL FREQUENCY 2
C              U11= MODAL MATRIX ELEMENT, (MASS1,FREQ1)
C              U22= MODAL MATRIX ELEMENT, (MASS2,FREQ1)
C              U12= MODAL MATRIX ELEMENT, (MASS1,FREQ2)
C              U22= MODAL MATRIX ELEMENT, (MASS2,FREQ2)
C              M11= MASS  MATRIX ELEMENT
C              M21= MASS  MATRIX ELEMENT
C              M12= MASS  MATRIX ELEMENT
C              M22= MASS  MATRIX ELEMENT
C              K11= STIFFNESS  MATRIX ELEMENT
C              K21= STIFFNESS  MATRIX ELEMENT
C              K12= STIFFNESS  MATRIX ELEMENT
C              K22= STIFFNESS  MATRIX ELEMENT
C              Z11= BASE  MATRIX ELEMENT
C              Z21= BASE  MATRIX ELEMENT
C              Z12= BASE  MATRIX ELEMENT
C              Z22= BASE  MATRIX ELEMENT
C     DECLARATIONS...
         PARAMETER(MAX=10000)
C     ARRAYS...
         REAL      A(3,3),L(3),UO(3),UI(3),U(3)
         DIMENSION TIME  (MAX),ANVILACC(MAX),
     :      ZBASE1 (MAX),ZBASE2  (MAX),ZBASE  (MAX),
     :      RELDIS1(MAX),RELVEL1 (MAX),RELACC1(MAX),
     :      RELDIS2(MAX),RELVEL2 (MAX),RELACC2(MAX),
     :      Y1ACC  (MAX),Y2ACC   (MAX),
     :      X1ACC  (MAX),X2ACC   (MAX)
C     VARIABLES...
         REAL F1,F2,FN1,FN2,W1,W2,WN1,WN2,W1SQ,W2SQ,WN1SQ,WN2SQ,
     :   M1,M2,M11,M21,M12,M22,
     :   K1,K2,K11,K21,K12,K22,
     :      U11,U21,U12,U22,
     :      Z11,Z21,ZCOEFF1,ZCOEFF2,
     :   ZETA,DELT,E,F,G,Z,WEIGHT
         INTEGER NUMSTEP,I1,I2
         PRINT*,'INPUT DAMPING RATIO, ZETA'
         READ*, ZETA
         PRINT*,'INPUT WEIGHT (LB) OF ONE TIER, MASS1=MASS2'
         READ*, WEIGHT
         PRINT*, 'INPUT NUMBER OF TIME STEPS UP TO 10,000'
         READ*, NUMSTEP
C     PARAMETERS USED...
         F1       =   94.
         F2       =   100.
         FN1      =   60.691177
         FN2      =   154.88250
         DELT     =   0.0001
```

```
      UO(1)     =   0.
      UO(2)     =   0.
      M1        = WEIGHT/32.2
      M2        = M1
      W1        = 2.*4.*ATAN(1.)*F1
      W2        = 2.*4.*ATAN(1.)*F2
      WN1       = 2.*4.*ATAN(1.)*FN1
      WN2       = 2.*4.*ATAN(1.)*FN2
      W1SQ      = W1**2
      W2SQ      = W2**2
      WN1SQ     = WN1**2
      WN2SQ     = WN2**2
      K1        = M1*W1SQ
      K2        = M2*W2SQ
C     NATURAL MODES...
      U11 = 1.
      U21 = (K1-WN1SQ*M1)/K1
      U12 = 1.
      U22 = (K1-WN2SQ*M1)/K1

C     DECOUPLE SYSTEM...
      M11 = M1+U21*U21*M2
      M21 = M1+U22*U21*M2
      M12 = M1+U22*U21*M2
      M22 = M1+U22*U22*M2

      K11 = K1+U21*(-K1) + U21*(-K1+U21*(K1+K2))
      K21 = K1+U21*(-K1) + U22*(-K1+U21*(K1+K2))
      K12 = K1+U22*(-K1) + U21*(-K1+U22*(K1+K2))
      K22 = K1+U22*(-K1) + U22*(-K1+U22*(K1+K2))

      Z11 = M1+U21*M2
      Z21 = M1+U22*M2

      ZCOEFF1 = Z11/M11
      ZCOEFF2 = Z21/M22

C     READY TO PROCESS AS SDOF SYSTEMS...

      REWIND 10
      OPEN(10,FILE='ANVILACCLM .DAT',STATUS='OLD')
      OPEN(20,FILE='TDOFTIMEREC.DAT',STATUS='NEW')

C     READ DATA FROM TUNED FIXTURE FILE...ANVIL ACCELERATION IN G'S
      DO 10 I=1,NUMSTEP
         READ(10,*) TIME(I),ANVILACC(I)
   10 CONTINUE

      DO 20 I=1,NUMSTEP
         ZBASE (I) =     ANVILACC(I)*32.2
```

```
          ZBASE1(I) = ZCOEFF1*ANVILACC(I)*32.2
          ZBASE2(I) = ZCOEFF2*ANVILACC(I)*32.2
   20  CONTINUE

C     ITERATIONS...
C     1.  BEGIN WITH UPPER TIER SDOF MODEL...INITIALIZE PARAMETERS...

          UO(3)      =-ZBASE1(1)-(2.*ZETA*WN1)*UO(2)-WN1SQ*UO(1)
          UI(1)      = UO(1)
          UI(2)      = UO(2)
          UI(3)      = UO(3)
          RELDIS1(1) = UO(1)
          RELVEL1(1) = UO(2)
          RELACC1(1) = UO(3)

          OPEN(30,FILE='TEST.DAT',STATUS='NEW')
          I1 = 1
          WRITE(30,*) I1
          WRITE(30,*) F1,F2,FN1,FN2
          WRITE(30,*) W1SQ,W2SQ,WN1SQ,WN2SQ
          WRITE(30,*) M1,M2,K1,K2
          WRITE(30,*) U11,U12
          WRITE(30,*) U21,U22
          WRITE(30,*) M11,M12
          WRITE(30,*) M21,M22
          WRITE(30,*) K11,K12
          WRITE(30,*) K21,K22
          WRITE(30,*) Z11
          WRITE(30,*) Z21
          WRITE(30,*) K11/M11,K22/M22
          WRITE(30,*) ZCOEFF1
          WRITE(30,*) ZCOEFF2
          WRITE(30,*) UO(1),UO(2),UO(3)
          WRITE(30,*) RELDIS1(1),RELVEL1(1),RELACC1(1)

C         COMPUTE PARAMETERS FOR OPERATOR MATRICES...
          E=WN1SQ*(DELT**2)
          F=2.*ZETA*WN1*DELT
          G=(1./(1.+(F/2.)+(E/4.)))

C         LOAD AMPLIFICATION AND LOAD VECTOR MATRIX...
          A(1,1)=G*(1.+(F/2.))
          A(1,2)=G*DELT*(1.+(F/4.))
          A(1,3)=G*DELT**2/4.
          A(2,1)=G*(-E/(2.*DELT))
          A(2,2)=G*(1.-E/4.)
          A(2,3)=G*DELT/2.
          A(3,1)=G*(-E/(DELT**2))
          A(3,2)=G*(-1.)*(E+F)/(DELT)
          A(3,3)=G*(-1.)*((F/2.)+(E/4.))
```

```
      L(1)  =G*(DELT**2)/4.
      L(2)  =G*DELT/2.
      L(3)  =G
C     BEGIN TIME ITERATIONS UPPER TIER MOTION...
      DO 100 J=1,NUMSTEP
      Z   =-ZBASE1(J+1)
      U(1)= A(1,1)*UI(1)+A(1,2)*UI(2)+A(1,3)*UI(3)+Z*L(1)
      U(2)= A(2,1)*UI(1)+A(2,2)*UI(2)+A(2,3)*UI(3)+Z*L(2)
      U(3)= A(3,1)*UI(1)+A(3,2)*UI(2)+A(3,3)*UI(3)+Z*L(3)
C     SAVE THE DIS,VEL,ACC ARRAY VALUES FOR THIS ITERATION...
      RELDIS1(J+1) = U(1)
      RELVEL1(J+1) = U(2)
      RELACC1(J+1) = U(3)
C     RESET THE ITERATION MATRIX FOR THE NEXT TIME
C     ITERATION...
      UI(1)=U(1)
      UI(2)=U(2)
      UI(3)=U(3)
  100 CONTINUE
C  2. NOW FOR LOWER TIER SDOF MODEL...INITIALIZE PARAMETERS...
      UO(3)     =-ZBASE2(1)-(2.*ZETA*WN2)*UO(2)-WN2SQ*UO(1)
      UI(1)     = UO(1)
      UI(2)     = UO(2)
      UI(3)     = UO(3)
      RELDIS2(1) = UO(1)
      RELVEL2(1) = UO(2)
      RELACC2(1) = UO(3)
      I2 = 2
      WRITE(30,*) I2
      WRITE(30,*) F1,F2,FN1,FN2
      WRITE(30,*) W1SQ,W2SQ,WN1SQ,WN2SQ
      WRITE(30,*) M1,M2,K1,K2
      WRITE(30,*) U11,U12
      WRITE(30,*) U21,U22
      WRITE(30,*) M11,M12
      WRITE(30,*) M21,M22
      WRITE(30,*) K11,K12
      WRITE(30,*) K21,K22
      WRITE(30,*) Z11
      WRITE(30,*) Z21
      WRITE(30,*) K11/M11,K22/M22
      WRITE(30,*) ZCOEFF1
      WRITE(30,*) ZCOEFF2
      WRITE(30,*) UO(1),UO(2),UO(3)
      WRITE(30,*) RELDIS2(1),RELVEL2(1),RELACC2(1)
```

```
C          COMPUTE PARAMETERS FOR OPERATOR MATRICES...
           E=WN2SQ*(DELT**2)
           F=2.*ZETA*WN2*DELT
           G=(1./(1.+(F/2.)+(E/4.)))

C          LOAD AMPLIFICATION AND LOAD VECTOR MATRIX...
           A(1,1)=G*(1.+(F/2.))
           A(1,2)=G*DELT*(1.+(F/4.))
           A(1,3)=G*DELT**2/4.
           A(2,1)=G*(-E/(2.*DELT))
           A(2,2)=G*(1.-E/4.)
           A(2,3)=G*DELT/2.
           A(3,1)=G*(-E/(DELT**2))
           A(3,2)=G*(-1.)*(E+F)/(DELT)
           A(3,3)=G*(-1.)*((F/2.)+(E/4.))
           L(1)  =G*(DELT**2)/4.
           L(2)  =G*DELT/2.
           L(3)  =G

C          BEGIN ITERATIONS...
           DO 200 J=1,NUMSTEP
           Z   =-ZBASE2(J+1)
           U(1)= A(1,1)*UI(1)+A(1,2)*UI(2)+A(1,3)*UI(3)+Z*L(1)
           U(2)= A(2,1)*UI(1)+A(2,2)*UI(2)+A(2,3)*UI(3)+Z*L(2)
           U(3)= A(3,1)*UI(1)+A(3,2)*UI(2)+A(3,3)*UI(3)+Z*L(3)

C          SAVE THE DIS,VEL,ACC ARRAY VALUES FOR THIS ITERATION...
           RELDIS2(J+1) = U(1)
           RELVEL2(J+1) = U(2)
           RELACC2(J+1) = U(3)

C          RESET THE ITERATION MATRIX FOR THE NEXT TIME
C          ITERATION...
           UI(1)=U(1)
           UI(2)=U(2)
           UI(3)=U(3)
     200   CONTINUE

C    CONVERT RELATIVE UNCOUPLED MOTIONS TO RELATIVE COUPLED MOTIONS
C    USING MODAL MATRIX...THEN RELATIVE COUPLED MOTIONS TO ABSOLUTE
C    TIER MOTIONS...
           DO 300 I=1,NUMSTEP
                Y1ACC(I)= U11*RELACC1(I)  + U12*RELACC2(I)
                Y2ACC(I)= U21*RELACC1(I)  + U22*RELACC2(I)
                X1ACC(I)= (Y1ACC(I)       + ZBASE(I))/32.2
                X2ACC(I)= (Y2ACC(I)       + ZBASE(I))/32.2
                WRITE(20,*) TIME(I),X1ACC(I),X2ACC(I)
     300   CONTINUE
           END
```

APPENDIX E

```
C     LT RANDALL CORBELL
C     NAVAL POSTGRADUATE SCHOOL

C     SHOCK QUALIFICATION OF COMBAT SYSTEMS EQUIPMENTS USING TUNED
C     MOUNTING FIXTURES ON THE U.S. NAVY MEDIUMWEIGHT SHOCK MACHINE

C     2 DOF TUNED FIXTURE COUPLED NATURAL FREQUENCIES STUDY FOR
C     MASS1/MASS2 RATIO

C     REF: (A) SHIN,Y.S.,"NEDE REPORT NED345, CLASS II", 1981

C     THIS PROGRAM ITERATES THE EXPRESSIONS FOR THE COUPLED NATURAL
C     FREQUENCIES, FN1 AND FN2, OF A 2 DOF SYSTEM AS EXPRESSED IN TERMS
C     OF ITS UNCOUPLED NATURAL FREQUENCIES, F1,F2, AND THE MASS RATIO
C     OF THE SYSTEM, MASS1/MASS2. REF(A) WAS USED TO OBTAIN AN
C     EXPRESSION FOR THIS RELATION.
C     FN1 = FIRST  MODE COUPLED NATURAL FREQUENCY
C     FN2 = SECOND MODE COUPLED NATURAL FREQUENCY
C     F12 = FIRST  MODE UNCOUPLED NATURAL FREQUENCY SQUARED
C     F22 = SECOND MODE UNCOUPLED NATURAL FREQUENCY SQUARED
C     R   = MASS1/MASS2

C     DECLARATIONS...
            REAL F1,F2,F12,F22,FN1,FN2,R,DELF,C
            PRINT*,'INPUT MASS1/MASS2 RATIO, R'
            READ*, R
C           PRINT*,'INPUT F1'
C           READ*, F1
            DELF=1.
            C   =(.5)**.5
            OPEN(20,FILE='NATFREQDAT.DAT',STATUS='NEW')
C           DO 100 I=1,250
                  F1  = I*DELF
                  F12 = F1**2
C                 DO 200 J=1,400
C                       F2  = J*DELF
                        F2  = 100.
                        F22 = F2**2
                        FN1 = C*((F12+R*F12+F22)
      :                       -((F12+R*F12+F22)**2 - 4.*F12*F22)**.5)**.5
                        FN2 = C*((F12+R*F12+F22)
      :                       +((F12+R*F12+F22)**2 - 4.*F12*F22)**.5)**.5
C                             IF(((FN1.GE. 20.).AND.(FN1.LE. 25.)).AND.
C     :                       ((FN2.GE.153.).AND.(FN2.LE.156.)))THEN
```

```
                    WRITE(20,*) F1,F2,FN1,FN2,R
C                 ENDIF
C    200        CONTINUE
     100   CONTINUE
     END
```

APPENDIX F

The preliminary design for a Two Degree of Freedom MWSM Tuned Mounting Fixture is presented below. The beam loading and configuration determines the system stiffness. For this model, each tier is simply supported with uniform loading over the span, L, between the two simple support beams of each tier. Figure F-1 depicts the model and Table F-1 lists the parameters.

TABLE F-1

PARAMETER	UPPER TIER	LOWER TIER
NATURAL FREQUENCY HZ	94	100
WEIGHT LBS	1700	1700
MASS LB*SEC^2/FT	52.8	52.8
STIFFNESS LB/FT	1.84E7	2.08E7
BEAM LENGTH W5x18.5 FT	5	5
NO. STIFFNESS BEAMS	3	3
NO. SIMPLE SUPPORT BEAMS	2	2
SUPPORT BEAM SPACING, L FT	4.0	3.8
LOWER TIER MOUNTING PLATE THICKNESS IN		1.2

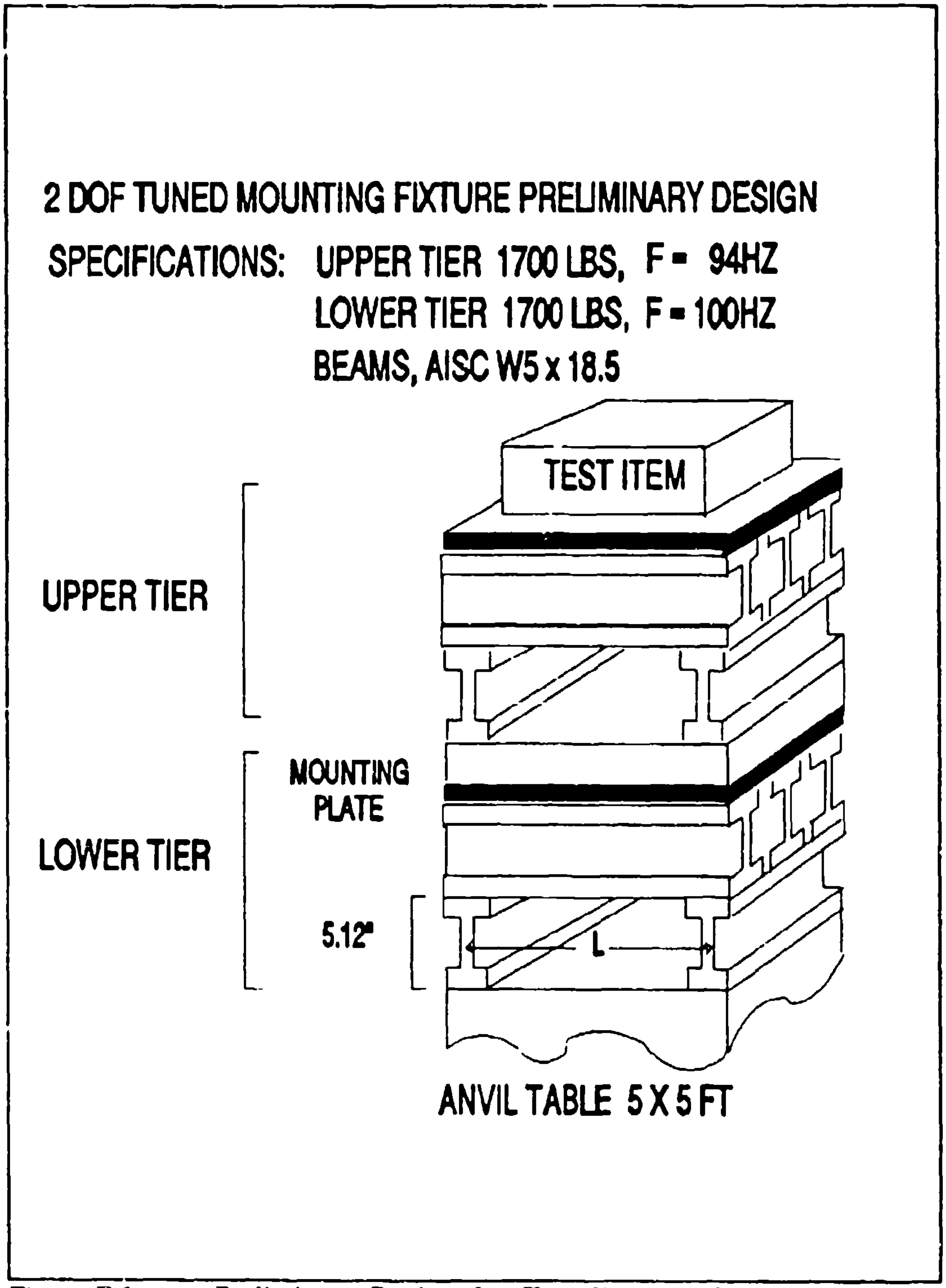

Figure F-1. Preliminary Design for Two Degree of Freedom Tuned Mounting Fixture.

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