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DEVELOPMENT OF A SHOCK WAVE SIMULATOR FOR SHELTER EQUIPMENT TESTING

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ABSTRACT

A series of studies was initiated by NCEL to determine the feasibility of constructing an equipment test machine capable of producing motions predicted to occur in shelters exposed to nuclear weapons effects. To establish the machine performance criteria, Agbabian-Jacobsen Associates contracted to investigate the response of typical equipment support elements to environments resulting from weapon yields ranging from 1 to 20 megatons with overpressures ranging from 15 to 500 psi. The Ralph M. Parsons Company contracted to develop a machine concept capable of producing the criteria motions. To establish the performance capabilities of the machine components, it was proposed to design and construct a Model machine using the operating principles and critical components of the basic concept. To this end, The Ralph M. Parsons Company performed the preliminary design of a complete test facility with a specimen capacity of 30 by 30 inches by 89 inches high and 400 pounds weight. Cost estimates of both the full scale and the Model machines established the feasibility of this proposed program.

equipments representative of highly coupled linear and nonlinear systems. Such fundamental parameters of motion in a shelter as peak displacement, velocity and acceleration depend on the respective amplification ratios, which can be accurately determined only if the shape of the waveform is known.

Verification of the acceptability of the spectrum as a criteria of damage potential and of the validity of sweep frequency and peak acceleration tests is usually established by comparing test results with the equipment response to the actual service environment. However, in this case, unlike most other problems, no data are available on the responses of the equipment to the actual environment for correlation with test results. Because of the vital nature of the equipment to be tested, to confidently interpret test results, it is necessary to construct a simulator which can duplicate, and vary systematically, all significant input wave-form parameters.

GROUND MOTIONS

To date, because of (1) the lack of an inclusive theory, (2) the shortcomings of field instrumentation used for measuring of ground motions, and (3) the unpredictability of ground irregularities at the test sites, there is no rational basis for predicting the exact shape of the waveform and the time phasing of the ground motion directional components. Nevertheless, careful review of nuclear detonations test data, with the aid of simple one-and-two dimensional elastic theory, has identified some distinct waveform characteristics whose ranges of magnitude can be predicted with fair accuracy.

Ground motions induced by a nuclear detonation may be divided into two classes by virtue of the process that produces them:

(1) Crater-induced ground motion, resulting from the direct conversion of the nuclear detonation energy into mechanical energy at ground zero, and

(2) Air blast-induced ground motion, resulting from the rapidly moving airblast wave produced from nuclear detonation.

The effects of a typical surface nuclear detonation and the relative significance of the crater-induced and airblast-induced ground motions is shown in Figure 1. Neither of the two motions can be well defined in a circle around ground zero, whose radius R_1 is equal to approximately one and one-half of the apparent crater radius R, because the soil in this region is completely altered due to cratering. At a given distance from ground zero for a surface detonation, both the crater-induced and airblast-induced ground motions will be present. As the range from ground zero increases, however, the crater-induced motion is attenuated rapidly and the airblast-induced motion becomes dominant.

Starting at distance R_1 from ground zero, the airblast peak overpressure, duration and arrival time can be expressed as a function of weapon yield and range. The strength of the airblast-induced ground shock at any point is determined by the magnitude of overpressure in the blast wave immediately above it. For large overpressures with long positive phase durations, the shock will penetrate some distances into the ground, but blast waves which are nearer and of shorter duration will be attenuated more rapidly. For uniform soil condition, the principal stresses in the soil will be vertical and about equal in magnitude to the blast overpressure. For nonuniform soils, however, peak soil stresses may actually be significantly higher than the blast overpressure.

The composition of most soils is generally nonuniform Thus the seismic velocity in the ground will vary both with depth and range. In a layered soil, a disturbance generated at the surface may be quickly transmitted to a lower layer of higher seismic velocity where it will move ahead of the wave in the surface layer. Further, refractions and reflections from the higher velocity wave may be transmitted back into the surface layer reaching a given point on the surface prior to the arrival of the original wave. Figure 2 shows the paths of crater-induced waves and of the airblast-induced waves that contribute to the ground motions at the buried protective shelter.

The phasing and arrival of the airblast wave and the ground wave may be divided into three regions, as shown in Figure 3. If the airblast wave arrives at a given distance from ground zero prior to the arrival of any wave transmitted through the ground, the condition is said to be super-seismic. This condition can occur only if the velocity of the airblast wave exceeds the seismic velocity of the soil and if no reflections or refractions from the lower layers outdistance the blast wave. In the second or trans-seismic region, the airblast and the ground wave arrive at the given point nearly simultaneously. In the third region, the ground is set initially in motion by an outrunning wave. This condition is referred to as sub-seismic. Thus, the motion of a particle of soil in a free-field around a protective shelter is dependent on the characteristics and phasing of (1) the direct airblast slap, (2) the outrunning airblast-induced waves, and (3) the crater-induced waves.

WAVEFORMS

To simulate the nuclear detonation induced motions at the equipment support points within a protective shelter, it will be necessary first to synthesize the free-field ground motion waveform at the given location. To this end, several prediction methods based on experimental data have been developed.

Newmark[1] has formulated a relationship for soil particle peak velocity, displacement and acceleration of crater-induced ground motions due to surface detonation in granite, and ranges at which airblast-overpressure is between 100 and 600 psi. Newmark's method is based on one-dimensional elastic theory with scaling criteria obtained from underground detonations of small nuclear weapons and high explosive charges.

The prediction method developed by Sauer[2] is based upon a correlation of field measurements at the Nevada Test Site and at the Pacific Proving Grounds, using near-elastic scaling. The peak values of motion parameters at the surface are calculated first from the equations and then attenuated for depth by the aid of curves scaled for weapon yield. Using this method, it is possible to predict ground motions in both superseismic and subseismic regions.

Several waveforms for ground motion predictions have also been developed. The two most widely accepted of such waveforms are shown in Figures 4 and 5.[2] The waveform in Figure 4 representing a superseismic airblast-induced ground motion due to direct airblast slap is referred to as "Type I Pulse." This waveform has been based primarily on measurements made during Operation Tumbler and in the high-pressure region of the Shot Princilla. It can be defined as an impulsively generated decaying sinusoidal oscillation with varying frequency. Most of the significant features of this waveform are relatively easy to predict at nominal depths with estimated soil properties. Parameters such as peak acceleration, velocity and displacement, velocity rise time, and residual displacement can be calculated by methods outlined in Reference 1.

The waveform in Figure 5, referred to as "Type II Pulse," represents a combination of all other ground motions including those due to reflections and refraction. This waveform was developed by comparing the outrunning Tumbler data from which the Type I Pulse motion was subtracted, with the Shot Koa data, from which the Type I motion was similarly filtered out. As it consists of many waves, the strength and phasing of each being dependent on many parameters, the characteristics of this component of the ground motion cannot be predicted with much confidence. This fact is borne by the poor correlation between the idealized Type II waveform of Figure 5 with that recorded during Shot Ivy - Mike, shown in Figure 6. For purposes of this study, the Type II motion is considered to have the general waveform given by Sauer,[2] modified in accordance with Newmark's[1] recommendations for peak values.

Horizontal ground motions generally are assumed to have waveforms similar to the vertical with peak values somewhat reduced. Newmark suggests ratios of horizontal to vertical motions varying from 0.3 to 1.0 depending on soil properties and wave phasing.

While presently it is not possible to accurately predict all of the parameters of these waveforms, it is expected that refined computer programs for tracing wave propagation in layered media will increase the level of confidence in predicting many of these features in the very near future. The availability of a machine to systematically vary these parameters will provide a means for evaluating the importance of the variations, and possibly, of eliminating the need for clearer defining some of them.

SOIL-STRUCTURE INTERACTION

The problem of determining motions of underground protective-shelter structure produced by engulfing free-field ground motions have proven to be extremely difficult. At first approximation, however, it can be assumed soil-structure interaction has no significant influence on either the spectra, nor on the motion waveforms transmitted to the interior of an underground shelter. This assumption is based on the results of numerous field experiments. In the case where motion was transmitted only by the ground without additional airblast effects, such as in the case of the Plowshare Experiment, no difference between motion measured in the ground and on the structure was observed. In the case where the structural specimen were directly subjected to the airblast, as in the Buster-Jungle Tests, no conclusive evidence was found that foundation movements in the ground influenced the structural response to any great extent.

AIRBLAST EFFECTS

There is one further source of motion to which equipment in a protective-shelter may be exposed. Aboveground portions of some shelters must bear the full brunt of the nuclear detonation airblast. The airblast will tend to move the shelter horizontally and at the same time, press it into the soil. This type of motion is referred to as the "airblast-induced rigid-body response of the structure." Bending and shearing of the aboveground portion of the shelter can also be expected as the result of high surface wind velocities. In addition, local deformations of exposed portions of the shelter structure must be anticipated due to overpressure in the airblast wave.

SHELTER RESPONSE

The free-field ground motion in all cases will be modified by the dynamic response of the protective shelter structure. The shelter structure in general tends to attenuate the high-frequency components of the free-field motion in certain ratios designated as "influence coefficients." If the equipment is mounted on a flexible structural support within the shelter, the oscillations of the support in its characteristic modes, similar to those shown in Figure 7, will alter the modified free-field ground motion at the support points of the equipment, with possibility of amplification of the motion at the resonance frequencies of the support elements.

The waveform of the motion at the equipment support points in an underground shelter might be considered for all practical purposes to comprise three components:

1. The regular pulse induced by direct airblast slap.

2. The quasi-oscillatory motion introduced by the airblast-induced outrunning seismic wave, and

3. The oscillatory motion characterizing the elastic equipment support members.

The detailed nature of each of these components is a function of many independent parameters, and the test machine to simulate the complete motion must be capable of generating independently each of these components in any desired form, ratio and phasing.

GROUND MOTION STUDY

To establish the output envelope for the proposed simulator at NCEL, a study of ground motions and the response of equipment in protective shelters was completed by Agbabian-Jacobsen Associates (AJA) under contract NBy-62198 with the Laboratory.[3] The study considers nuclear detonation induced motion inputs to five typical items of equipment, listed in Table 1. The equipment was assumed to be mounted either at the lowest horizontal level or on the side in six typical shelters as shown in Figures 8 and 9. For analytical purpose, equipment was considered to consist of idealized single-degree-of-freedom systems. Homogenous soils and two types of layered soils, shown in Figure 10 were considered. The shelters were located at points where airblast overpressures induced by weapon yields of 1 to 20 megatons ranged from 15 to 500 psi.

The shock motion at the equipment support points was evaluated in terms of shock response spectra and a qualitative description of velocity time histories.

The analysis included the following nine steps:

1. A preliminary structural design of the six shelters was performed to establish approximate stiffness and mass characteristics of each structure.

2. The peak free-field ground motion intensities were calculated for the direct airblast slap at all specified conditions over a range of depths underground in which each structure was located.

3. The air-slap induced seismic motions were considered by calculating arrival time of refracted and reflected waves. The peak intensities were related to the peak intensities of the direct airblast slap motion.

4. The velocity-time history characteristics, for the motion defined in Step 2, were based on the compatibility of acceleration, velocity and displacement.

5. The velocity-time history for the motion calculated in Step 3 was assumed to be an initially upward Type II Pulse with time characteristics empirically related to field test data.

6. The peak displacement, velocity and acceleration at the critical wall and floor equipment supports induced by the airblast impact on aboveground shelters were calculated.

7. The peak motions calculated in Steps 2, 3 and 6 were combined as a single resultant motion imparted to each shelter structure by taking the square root of the sum of the squares of each peak value.

8. The flexibility of the structure was considered in assessing the input to the equipment. The wall or floor supporting the equipment was modeled to a single-mass-spring system with the equipment and attachment as a second mass-spring system. The equipment mass was found to be small with respect to the mass of supporting elements. Therefore, the response of supporting elements was calculated, assuming a single degree-of-freedom system, using a shock-spectra response envelope.

9. The response calculated in Step 8 became the input to the equipment and was used to obtain modified shock spectra describing response of the equipment.

The study concludes that the simulator must be capable of producing, simultaneously in the vertical and horizontal directions, two types of motion:

1. A velocity jump similar to Type I Pulse combined with 3 or 4 cycles of alternating motion similar to Type II Pulse, and

2. A quasi-steady vibration with a frequency range of approximately 5 to 500 cps. Combination of the Type I and Type II waveforms should be possible in any manner that results in peak motion values less than or equal to the maxima shown in Figure 11.

MACHINE PERFORMANCE CRITERIA

The initial performance criteria for the test machine were developed from the reported results of the ground motion study.[4] Subsequent to the publication of the report, AJA conducted a refined analysis based on a recently developed computer program which incorporates one-dimensional dynamic stress wave propogation theory in layered, damped media. This analysis (1) identified structures 3, 4, and 5 as critical cases, and (2) reduced the peak motion requirements.[5] The revised machine performance criteria developed from this subsequent analysis are summarized below.

Maximum specimen weight	1,500 lb
Maximum specimen size	6 ft x 6 ft x 7.5 ft high
Maximum acceleration	146 g's vertical -- 133 g's horizontal
Maximum velocity	198 ips vertical -- 108 ips horizontal
Maximum displacement	46 in. vertical -- 16 in. horizontal
Maximum frequency	600 cps desired -- 200 cps acceptable

To produce the specified motion, the simulator must be capable of generating very large forces in a very short period of time and over significant displacements. In some cases where the outrunning motion occurs prior to the arrival of the blast wave, the simulator will already be in motion when the requirement for a high acceleration, and therefore, maximum force arises. Furthermore, the motion must be imparted to test items simultaneously in vertical and horizontal directions, incorporating all frequency components within the full range of the anticipated bandwidth. Also, the time histories of the motions must be adjustable throughout the established range of waveform parameters. It is evident that such a machine will represent an advancement in the state of the art of shock testing. On the other hand, if feasibility is to be clearly established, it must be designed only of proven components and materials and constructed by proven techniques.

The difficulty of achieving performance specification of this magnitube has discouraged, in the past, the development of ground motion waveform simulators. So far as is known, only one testing machine has ever been constructed to simulate a nuclear ground shock waveform. This machine, installed at the Air Force Special Weapons Center, Kirtland Air Force Base, New Mexico, was designed specifically to generate one type of motion, a Type I waveform. Slight variations can be made in the strength of the shock, but ranges of adjustability are narrow. Further, the machine lacks the capability to simulate the Type II ground shock components.

FEASIBILITY STUDY

To determine the feasibility of a simulator of the type under consideration, a study was undertaken by The Ralph M. Parsons Company for NCEL under contract NBy-62201. The study was divided into two phases: (I) an evaluation of existing and proposed machine concepts, and (II) a preliminary design of a complete test facility based on the most promising machine concept.

CONCEPT EVALUATION

The evaluation of the machine concepts was separated into two relatively independent areas, power generation and kinematics.[6] Detailed investigation into the possible types of motion generators revealed that the required motions are beyond the limits of currently available closed loop (continuous control) actuator systems, and that open loop systems do not have the desired versatility. It was concluded, therefore, that a combination and open loop-closed loop system concept would be necessary to generate the specified motions. The major kinematic concerns were that the machine be able to provide two independent degrees of freedom, transmit undistorted high frequency motions and that the mechanical "noise" due to bearing tolerance be a minimum. Further, the study indicated that, although not employing elements beyond the current state of the art in the construction of the simulator, the concepts envisaged involved certain components and configurations whose performance could best be established by experiment. To this end, the construction of a full-scale simulator would be preceded by fabrication and test of a smaller machine or Model, as it was designated, employing the same principles of operation.

The smaller machine would be capable of testing mechanical and electrical equipment and shock isolation systems for use in critical communications facilities. After verification, the Model components could either be incorporated in the full-scale simulator, or the Model could be retained intact as a research tool.

This course was adopted and the performance criteria amended as follows:

Maximum specimen weight	400 lbs
Maximum specimen size	2.5 ft x 2.5 ft x 7.5 ft
Maximum displacement	24 in. vertical
	8 in. horizontal

The ranges of weapon yields and overpressures that could be simulated by a machine built to the initial performance criteria and the effect of displacement criteria reduction is shown in Figures 12, 13 and 14. It can be seen that only the ranges for soil profile (a) are affected. This is not considered a serious limitation.

OPERATING PRINCIPLE

The operating principle of the concept is shown on Figure 15. The hydraulic servo is programmed to introduce the sum of all the low frequency components of the desired motion. The impacting device, also hydraulic servo controlled, is preset to impact with the moving test platform at the proper time intervals, generating the high frequency pulses needed to complete the direct airblast-induced and outrunning motions. If significant structural responses occur at frequencies higher than the limiting frequency of the servo, elastic elements of the proper characteristic may be inserted between the platform and the test specimen.

As the servo which drives the impacting device is a low frequency element, and a finite time is required for the impacting head to reach the desired impact velocity, there is a minimum time interval between successive impacts. However, the limiting intervals are much shorter than those typical of the outrunning motion.

PRELIMINARY DESIGN

The preliminary design phase of the feasibility study involved the design of a complete test facility including the basic machine, its supporting features such as data recording instrumentation, foundation and building, a performance evaluation and the cost and time estimates. A summary of the results of the study, as reported by The Ralph M. Parsons Company, are presented in the following sections.[7]

MACHINE ELEMENTS

The basic machine consists of three subsystems: structural, hydraulic and control. The ability of the structural elements to transmit the motions, undistorted, is determined principally by their natural frequencies, a function of rigidity and weight. High structural weight, however, increases the load on the hydraulic power system. A high tare load on the hydraulic system, in turn, affects its ability to respond to control system signals. Thus, to obtain optimum machine performance, the basic machine elements were designed for maximum rigidity with minimum weight.

Friction losses and clearance noise were also items of concern. Moderate amounts of friction in the mechanisms are not critical as the force outputs are much higher than the friction forces. However, friction losses do represent nonlinearities in the system and can cause instability in servo-controlled feedback. Nominal clearances can become the sources of considerable output noise.

STRUCTURAL SUBSYSTEM

The structural subsystem consists of the platform, the vertical and horizontal load transmitting elements, and the guidance mechanisms. These elements are shown in Figures 16, 17 and 18.

The platform, made of a magnesium alloy casting, provides the load application and specimen attachment points. Its area is determined by the maximum specimen size and is limited by the natural frequency requirements. At the bottom of the platform are three sets of dovetail plates which, with the roller way bearings mounted on the vertical base frame, provide the horizontal guidance for the platform. This guidance mechanism also transmits the vertical load from the base frame to the platform. A similar arrangement is provided on one side of the platform so its vertical motion is guided by the horizontal base frame. The horizontal load is also transmitted from the base frame through the guidance mechanism to the platform.

The load transmitting elements include two independent sets: vertical and horizontal. Each set consists of the hydraulic actuators and the mechanical linkages that apply the load to the platform. The vertical load transmitting system consists of the vertical base frame, the vertical impact frame, and the vertical hydraulic actuators. The vertical base frame serves to transmit the force from the main hydraulic actuator and impact device to the platform. It also provides a base for the horizontal guidance of the platform. The frame is connected in the center section to the main actuator on the bottom and to the main platform on the top. Four webs extending from the center section connect with four impact blocks which are guided vertically by four rods through ball bushing bearings. The vertical impact decelerators are attached to the impact blocks and are arranged so that they are impacted by the vertical impact frame in either the upward or the downward direction. Again, because of weight and rigidity considerations, the vertical base frame is made of a magnesium alloy casting.

When the platform is in an off-center position or the specimen center of gravity does not coincide with the vertical main actuator center line, a moment is produced. This moment is reacted by the foundation through the guidance rods of the vertical base frame.

The frame of the vertical impact device is supported by two impact hydraulic actuators. The frame, made of aluminum alloy, connects each impact actuator to a steel impact head assembly which also slides on the vertical guidance rods. The two impact heads are connected physically by two aluminum beams to insure they travel together. The vertical impact frame is designed so that it can be easily disassembled and the impact actuators connected directly to the base frame. By this means, the total performance capability of the low frequency shock in the vertical direction can be increased.

There is one vertical main hydraulic actuator and two vertical impact hydraulic actuators. The main actuator is driven by a 1,000 gpm hydraulic servo valve. A single 1,000 gpm hydraulic servo valve is used to drive both impact actuators. One 80 gpm compensating hydraulic servo valve is also attached to one of the impact actuators as a means for adjusting the relative position of the two vertical impact frames. To achieve good dynamic efficiency, the vertical actuators are based directly on the foundation. The actuator columns are made of aluminum alloy with hard anodized surface, providing excellent wear properties at minimum weight.

The horizontal load transmitting system also consists of a base frame, impact frame and hydraulic actuators. The base frame, a box-like structure of welded magnesium alloy tooling plate, performs the same function as the vertical base frame, but in the horizontal direction. Its front face provides an attachment for guiding the main platform in the vertical direction. The horizontal impact decelerators are mounted on the inside surfaces of the box structure. The impact shock is transmitted from the back surface of the base frame to front through four steel rods which also serve as guides for the horizontal impact frame.

The steel horizontal impact frame is attached at its center to the horizontal impact hydraulic actuator and is guided at its corners by the four steel rods specified above. Plates are provided on both the front and rear faces at the points of impact with the horizontal base frame.

In the horizontal direction, the main hydraulic actuator and the impact hydraulic actuator are mounted in tandem. The piston rod of the main actuator goes through the hollow piston and rod of the impact actuator and the impact frame, and connects to the base frame. The impact frame is connected directly to the impact actuator column. Each actuator is driven by two 200 gpm hydraulic servo valves ganged to function as a single unit.

Guidance mechanisms are used to react bending moments on the platform due to off-center loading. In this application, they also serve as load carrying elements. The methods employed to guide the main platform were described previously. The "Roller Way" bearing, manufactured by Beaver Precision Products Incorporated, was specified as the primary platform guiding element because it has a high load carrying capacity, close dimensional tolerances and it can be preloaded to eliminate clearance. For secondary guidance where no load is transmitted, the ball bushing bearing, manufactured by Thomson Industries, Inc., was selected. This bearing has diametrical clearance of less than one thousandth of an inch. Very close tolerances in the guidance mechanism demands precise control of the dimensions of the simulator parts. To minimize the machining cost, shims will be used during the assembly and alignment of the machine.

HYDRAULIC SUBSYSTEM

The hydraulic subsystem provides the power input to the machine. Electrical energy is converted, through a motor pump set, to high pressure hydraulic fluid. The high pressure fluid, in turn, is stored in accumulator tanks until released by a servo control valve to drive an actuator piston. All the components of the subsystem are commercially available items designed to operate with MIL-5606 hydraulic fluid at 3,000 psi and 100 degrees F.

A 50 gpm, variable volume, pressure compensated type hydraulic pump, driven by a 100 horsepower motor, is sized to charge the accumulators in 1½ minutes while compensating for leakage flow through the servo valves. The pump and motor are mounted on a 100 gallon reservoir fitted with a suction filter.

The 30 gallons fluid capacity of the accumulator bank provides for a maximum travel of six times the full actuator piston stroke displacement during a test. To achieve rapid system response, bladder type transfer accumulators are used. The accumulators' gas chambers are connected to a 40 cubic foot high pressure gas tank to minimize adiabadic expansion pressure drop. Identical fluid capacity low pressure surge accumulator tanks are located on the return side of the system.

The most critical element in the hydraulic subsystem is the servo valve. As previously stated, there are seven of these valves, two 1,000 gpm and one 80 gpm rated for the vertical motion and four 200 gpm rated for the horizontal. Each utilizes a low voltage electrical signal from the control system to regulate the direction and rate of flow of the fluid, thereby regulating the hydraulic system power input. The performance of these valves limits the performance capability of the complete simulator. The servo valve performance characteristics on which the preliminary design is based, therefore, are presented in the section on machine performance. As their performance is very sensitive to fluid contamination, a 10 micron filter is provided at the inlet to each servo valve. A hydraulic fluid cooler, safety valves, isolating valves, and miscellaneous fittings complete the hydraulic system equipment list.

CONTROL SUBSYSTEM

The function of the control subsystem is to control the motion of specimen mounting platform by regulating the hydraulic system power input to the machine. It was designed, therefore, to have a high degree of operational stability and reliability, and to have adequate features to minimize machine damage due to component failure or incorrect control input. The complete subsystem consists of an input unit and five hydraulic control units, three for vertical motion and two for horizontal.

The test program input to the machine consists of a four-track pre-recorded magnetic tape with an analog output in the form of velocity-time function. The channels are recorded and played back simultaneously to assure synchronization. The channels (tracks) are:

1. horizontal impact
2. horizontal platform motion
3. vertical platform motion
4. vertical impact

The output of the four read amplifiers are conditioned for direct input to the control subsystems. The vertical and horizontal platform motion signals drive the main actuators and the impact actuators. The impact signals are superimposed on the platform motion signals to the impact actuators. Thus the primary motion of the platform is followed by the impactors and the impact is effected by adding or subtracting from this motion.

The basic platform vertical control system consists of the 1,000 gpm servo valve, the main hydraulic actuator and a linear variable differential transducer (LVDT). The servo valve drives the hydraulic actuator to produce the output motion at the platform. The LVDT, with a range of ± 12 inches, measures and converts the output platform motion to an electrical feedback signal. The difference between the feedback and input signals is used to drive the servo valve. Electronic position limiting ensures that a unit malfunction and/or incorrect input does not overdrive the machine.

The relative positional accuracy required of the two impact actuators magnifies the complexity of their control systems. The two actuators must act as a single input and must maintain the same position, prior to impact, with relation to the platform. This is accomplished by using a single 1,000 gpm servo valve for both actuators. Two ± 12 inch LVDT's are used in parallel to provide the gross position feedback control.

By comparing the feedback signal of these two transducers, any positional error between the two impactors is detected. This relative displacement error is used as the input to a supplemental 80 gpm servo valve connected in parallel with the 1,000 gpm output to one impact actuator only. Four 1-inch transducers are mounted on the impactors so as to measure the final inch of upward or downward travel relative to the platform. The output of the transducers is compared and any difference is used as an error feedback signal to the 80 gpm servo valve. The positional error is thereby reduced to about 0.002 inches.

These subsystems provide the degree of differential correction required for vertical impact input to the platform. They do, however, add a requirement for special feedback control electronics and make the alignment calibration to the test machine a major pretest operation.

The horizontal control subsystems are similar, but not as complex as the vertical. The ganging of two 200 gpm servo valves to provide the input to each horizontal actuator does not introduce any significant problems.

As with the hydraulic system, to insure reliability, the control system is designed using commercially available components. However, the feedback control unit and the feedback transducers will require special fabrication. The ± 12 inch transducers, while within the state of the art, are longer than commercially available "off-the-shelf" units. The one inch fine position transducers will require special mounts to operate in their impact environment.

MACHINE PERFORMANCE

The performance capability of the simulator is defined by the strength of the motions applied to the test specimen and by the accuracy with which these motions compare with the predicted service environment. The factors which play a major role in machine performance are the power and frequency limitations of the actuators, the capability of the control system to detect and to correct errors, and the amount of distortion introduced by the hydraulic fluid and the mechanical elements of the system.

The performance characteristics of three major elements, the actuators, the controls, and the elastic components of the machine, were estimated and the capability of the entire assembly compared with the design criteria. As previously stated, the most critical elements of the machine are the servo valves. The performance evaluation of the hydraulic actuators and of the closed-loop response of the simulator were based on the limited engineering and performance data supplied by the Pegasus Laboratories, Inc., a principal manufacturer of high flow rate, rapid response servo valves.

The performance of the actuator can be specified by its ability to drive a rigid mass in a sinusoidal vibration or to produce a half sine acceleration pulse. Figures 19 and 20 show the performance limitation of the vertical and horizontal actuators to produce a sinusoidal vibration on a 400 lb specimen. The lower lines show the performance of each main actuator alone, while the combined effect of the main actuator and the impact actuator with the impact frame removed is shown by the upper lines. Similarly, the performance limitation of the vertical and the horizontal actuators to produce a half sine shock pulse are shown in Figures 21 and 22. It can be seen that the performance of the closed-loop, servo-controlled hydraulic actuators is limited to motions with relatively low frequency and acceleration.

For motions with higher frequency and acceleration, the impact method must be used. The frequency of an impact pulse is a function of the stiffness of the decelerator. The generation of a particular pulse frequency depends on the selection of the proper size and material to achieve the required decelerator pad stiffness. Once the frequency of the pulse is fixed, the pulse magnitude is strictly a function of the relative impact velocity. The pulse magnitude and the decelerator size determine the maximum stress in the decelerator. This stress should be kept within the linear, elastic region of the decelerator material if a half sine type shock pulse is to be produced. For high frequency, high acceleration pulses, hard rubber and plastics have been found to be the most suitable materials.

The motions at the main platform will be the same as the actuator output only if the load transmitting elements are infinitely rigid. Since this is not possible, some distortion of the output will always be present. The magnitude of the distortion is a function of the ratio of the lowest natural frequency of the transmission path elements to the highest input frequency. The maximum input frequencies will be about 50 cps at the attachment of the main actuators (low frequency system) and 200 cps at the point of impact on the main platform (high frequency system). Estimates predict a probable minimum frequency of about 750 cps in both the vertical and horizontal base frames. With a frequency ratio of 15, the distortion in the low frequency system output will be fairly low, with residual oscillations of less than two percent. Similarly, the frequency ratio for impacts of 3-3/4 is considered acceptable for minimum distortion.

During impact, both in the vertical and horizontal directions, the motion of the base frame is resisted by the main hydraulic actuators which are low frequency elements of about 50 cps. Some of the 50 cycle oscillation will appear at the table as distortion. However, once the characteristics of the machine during the impact are determined by calibration, the impact pulse can be modified so that, together with the distortion, the proper shock pulse is achieved. Also, the compressional flexibility of the base frames will generate residual oscillations. Although these oscillations will be at a frequency above the range of interest, they do constitute an unwanted environment.

In Figures 23 and 24, the maximum motions which can be generated by the Shock Simulator are shown and compared with the criteria. Except for displacement where a reduction was made for the Model, it is seen that the machine capability exceeds the criteria requirements.

Two vital considerations in evaluating the machine performance and its suitability for acquiring useful data are its accuracy in simulating the desired waveform and in repeating the test environment. As the servo system is the basic component both in the low frequency and in the impact modes, and as only preliminary estimates of the servo valve performances are available, the accuracy and repeatability can only be estimated at this time.

The low-frequency system comprising just the hydraulic servos without impact poses no unusual problems and is typical of many (although smaller) systems now in service. It is estimated that by careful calibration, the repeatability of the Simulator during closed-loop servo controlled operation can be held to ± 5 percent.

It is estimated that the pulse shape due to impact alone can be closely controlled. Again, however, the response of the servo actuator system during impact and its inter-relationship with the impact mode is dependent on the high frequency capability of the servo system and on the more complex control network which is required. As noted earlier, by careful calibration and adjustment many of these effects can be minimized or even eliminated. However, as a basis for comparison, it is estimated that the repeatability of the impact mode will be on the order of ± 10 percent.

DATA ACQUISITION AND RECORDING

The design of the data acquisition and recording system does not present problems of the same magnitude as the development of the actual machine. The measurement instruments and the data recording methodology required are well within the state of the art. The developed data system is capable of measuring performance of the test machine, responses of the test specimen, and performance of the specimen under test conditions. It is a direct digital recording system which is compatible in format and message content with the NCEL IBM data reduction system. All the components may be procured on an "off-the-shelf" basis from many vendors. Sufficient transducers are included to acquire the following data:

- Platform motion in two axes
- Specimen response in three axes
- Specimen functional operation
- Machine operation

Intermediate steps between the transducer and recording system include signal conditioning to normalize the data, excitation as necessary, and conversion of the data form. As the output varies with the type of transducer, and as the input to the analog-digital converter must be of standard level, amplification and some filtering is required.

The output of the initial system is digital data recorded on ½-inch IBM compatible tape. Data evaluation is accomplished through use of the NCEL 1620 computer system. Although "quick look" data are not required, a dual trace oscilloscope is required for monitoring the operation of the machine.

The data system is completely modular and can be expanded both in scope and capability. Conversion of the data input to determine other parameters can be accomplished automatically with little additional equipment. The signal conditioning equipment can include an integration, double integration, or differentiation function. This would permit direct recording of the parameter that is to be analyzed such as real time evaluation of velocities. In addition, the following equipment will eventually be added:

- alpha numeric, single channel readout capability
- 10 channel oscillograph, 500 cps response
- 2 axis high speed camera system

The use of off-the-shelf components with proven performance records will assure reliability. Further, to obtain a total system performance guarantee, it is planned to procure the data acquisition and recording system on a total system basis.

FOUNDATION AND BUILDING

The performance evaluation of the simulator was based on the assumption that the machine would be supported by a rigid unbounded mass. The effect of flexibility or resilience of the foundation will be to increase platform motion distortion. The design of the foundation, therefore, is an important component of the complete facility.

The foundation design, based on data derived from soil borings at the proposed site, consists of a concrete mat foundation supported by a 3-foot thick compacted gravel fill pad located 7 feet below grade.

The weight of the foundation is estimated at 206,000 pounds, while the backfill directly above the mat is about 226,000 pounds. Using these weights, conservative estimates predict peak motions at the actuator support points will be approximately 0.10 inches vertically and 0.024 inches horizontally. These motions will decay very rapidly, of course, due to the high damping between the foundation and the sand.

To protect the machine, its controls and its instrumentation from the salt spray and sand common to the proposed site, a small sealed building was designed to house the equipment. The 30- x 50-foot sheet metal building contains a concrete wall between the control and test areas to protect personnel from shrapnel.

FEASIBILITY

Figure 25 presents an estimate of the time required to complete the detail engineering design, to verify by experiment the performance of certain critical items, to fabricate the Model Simulator, its accessories and housing, and to calibrate the completed assembly. It is estimated that the design through installation of the 1,500-pound capacity machine will require 24 months.

Estimates of the cost of designing, fabricating, and installing the 400-pound and 1,500-pound capacity Simulator, exclusive of cost of program administration, are given in Table 2. These costs include only those associated with installing the equipment and demonstrating its capability to meet the performance specification. The engineering effort includes drafting the experimental program, observation of the tests and reduction and interpretation of the data, but does not include the costs of the test gear, nor performing the tests.

It should be noted that there are several areas where equipment common to both systems is included in both cost estimates. If the Model machine would continue to be used after the 1,500-pound machine was put into operation, by using instrumentation already available at the Model, a saving of about $70,000 could be made in the cost of the 1,500-pound machine, and at the same time, the added instrumentation presentation equipment would be available for both systems. If, on the other hand, the Model machine were dismantled and its components incorporated into the full size machine, the cost of the 1,500-pound machine would be reduced by approximately $100,000.

CONCLUSIONS AND RECOMMENDATIONS

On the basis of this preliminary design analysis, it is believed that the construction of a test machine to generate a wide variety of programmed motions similar to those predicted to occur in elastic structures exposed to nuclear blast-induced ground shock is feasible and can be accomplished by the use of proven components. Machine outputs simulating a wide variety of direct airblast-induced and outrunning motions and their combinations can be achieved throughout a range which encompasses many facilities of interest. To generate these motions, however, the machine must incorporate these significant features:

1. Independent vertical and horizontal motions
2. Infinite variability in low frequency waveforms with moderate accelerations
3. Superposition of high frequency high acceleration pulses as desired
4. Pulses to be either positive or negative

The advantages to be gained by the availability of such a machine considered to be of vital importance to the hardened facility programs are: First, the machine will provide a research tool for validating the current testing techniques. By varying waveform but maintaining essentially the same response spectrum, the error in accepting the spectrum as a criteria of damage potential can be evaluated quantitatively for a wide variety of equipment now in use in hardened facilities. Similarly, the acceptability of the sweep frequency test and the error involved in introducing vertical and horizontal shocks independently can be determined. In this manner, standardization of shelter equipment test techniques can be established. Second, by virtue of its flexibility of output, the machine can be used as a means for upgrading equipment, investigating the feasibility of various types of shock isolators, optimizing their transmissibility and damping characteristics, and developing individual equipment and personnel hardening criteria. Further, the machine will be the only one in existence which can be used to proof test beyond the limit of doubt all types of equipment with or without the isolating elements for installation in critical facilities. In summary, it is believed that a strong need exists for a machine with these features, and that the concept, as described, is entirely feasible. Therefore, it is recommended that a program leading to the design and construction of the Model machine, be initiated as soon as practicable.

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Table 1. Typical Equipment Characteristics

Equipment Name	Number	Weight (lb.)	Horizontal[1] Natural Frequency
Microwave R and T	1	130	50
Base Station Transceiver	2	300	25
Tape Unit	3	800	10
Air Compressor	4	300	40
Generator Set	5	1,170	20

1. Vertical frequencies are twice the horizontal frequencies (cps)

Table 2. Test Facility Cost Estimate

Test Facilities	400 lb.	1,500 lb.
Simulator Building	$ 20,000	$ 30,000
Simulator Foundation	13,000	25,000
Simulator Assembly	40,000	90,000
Hydraulic System	57,000	116,500
Control System	31,700	46,000
Instrumentation	72,500	132,500
Installation	19,000	35,000
Engineering	35,000	67,000
	$ 288,200	$ 553,000

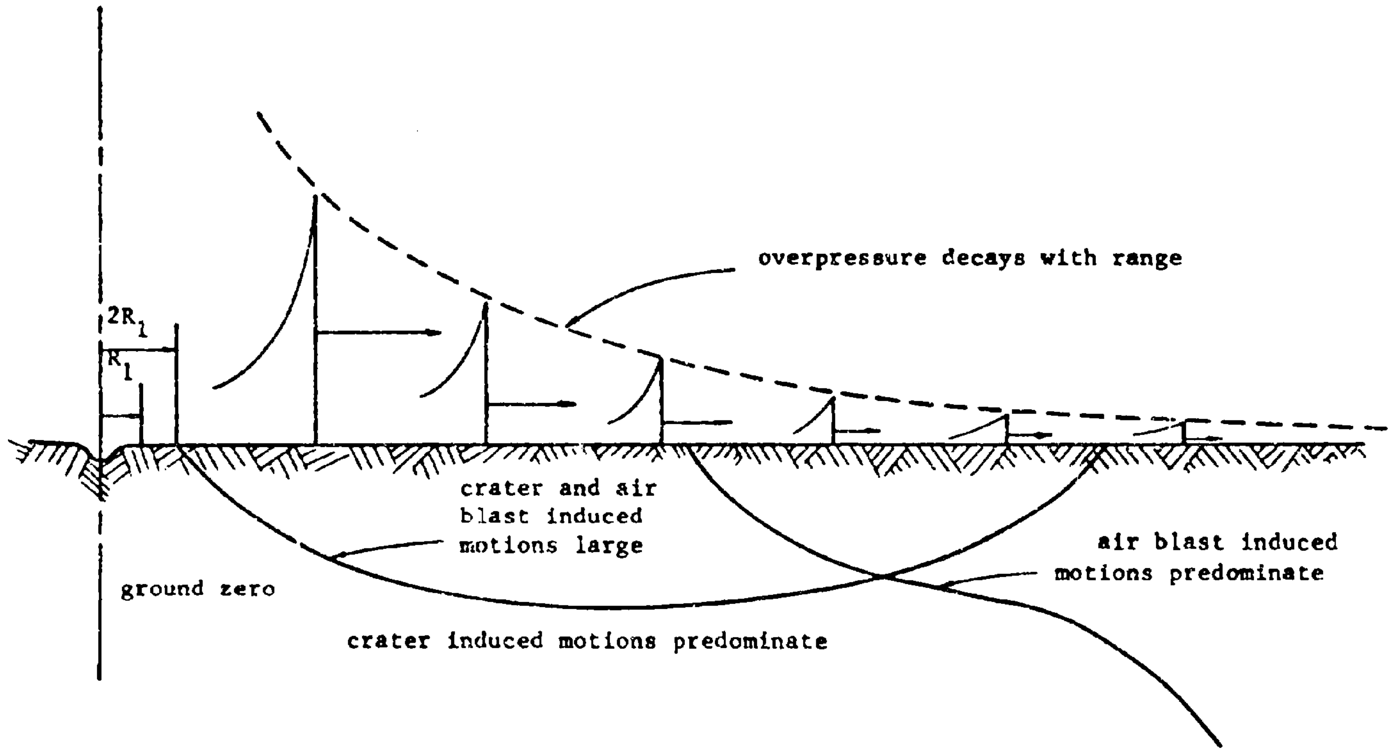

Figure 1. Effects of surface nuclear detonation.

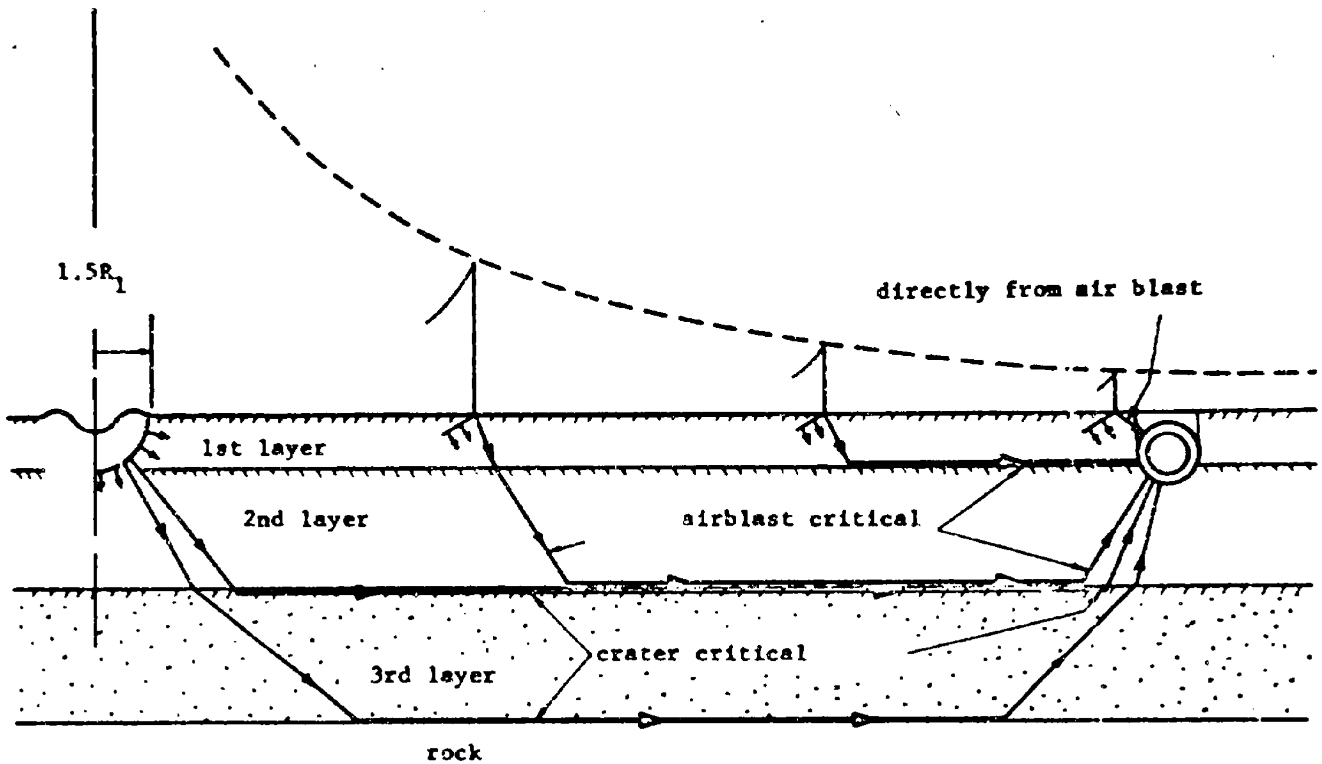

Figure 2. Propagation of crater and airblast seismic waves.

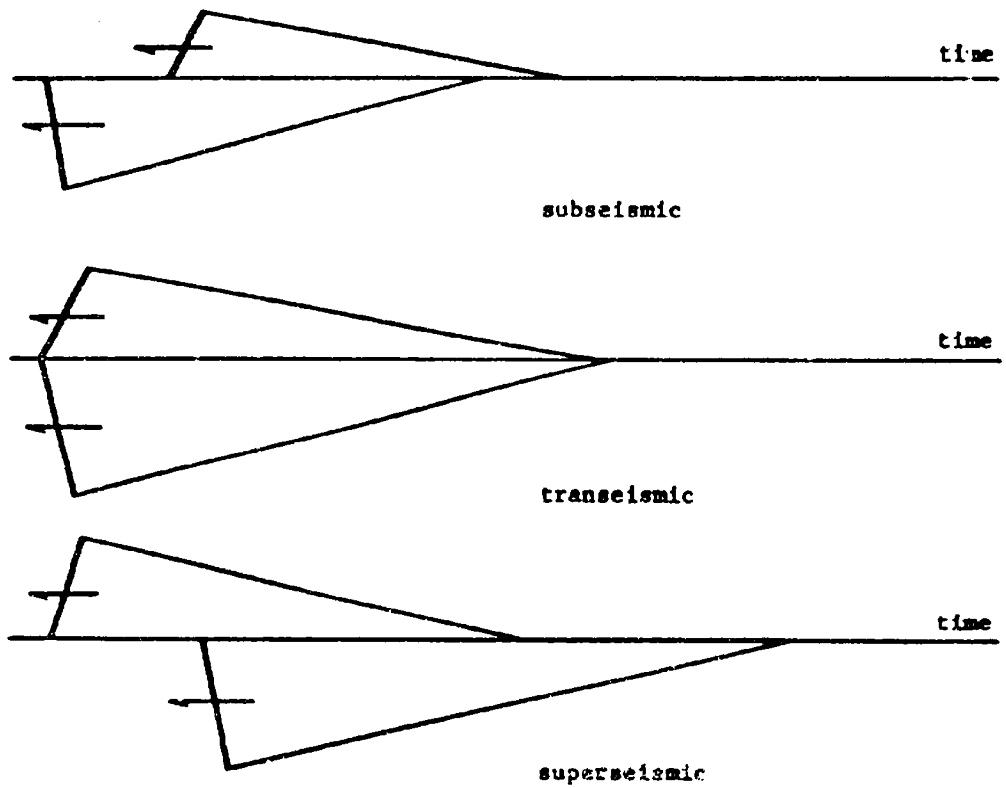

Figure 3. Phasing of triangular waveforms.

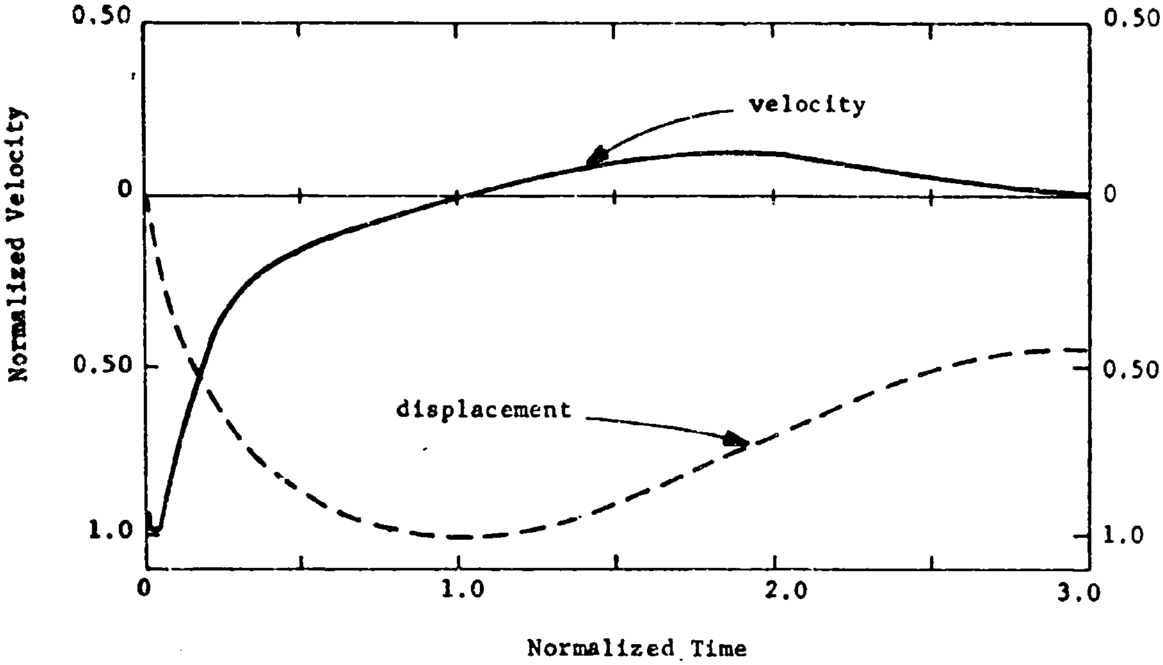

Figure 4. Idealized Type I superseismic airblast induced ground shock waveform.

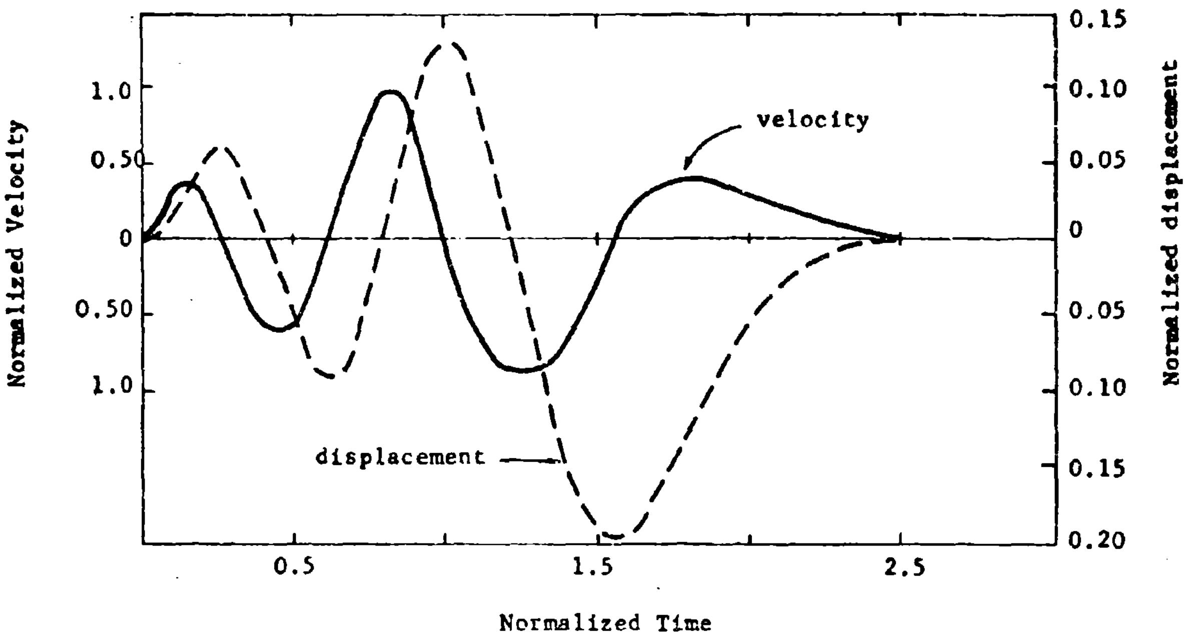

Figure 5. Idealized Type II outrunning ground motion waveform.

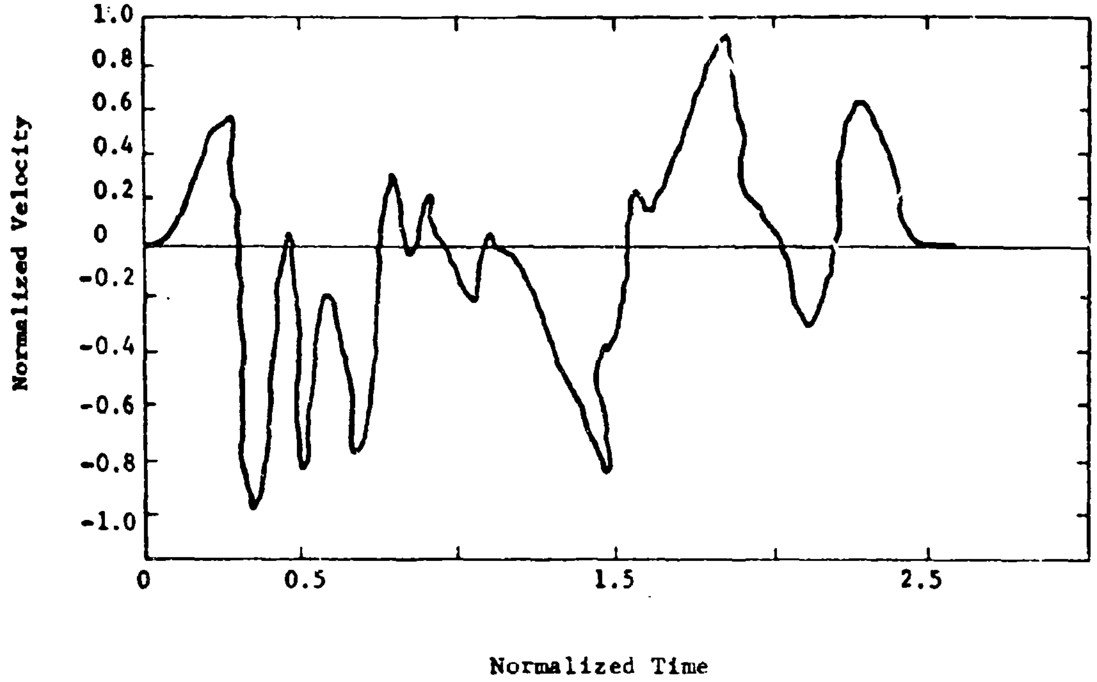

Figure 6. Vertical outrunning ground motion recorded at Shot Ivy-Mike.

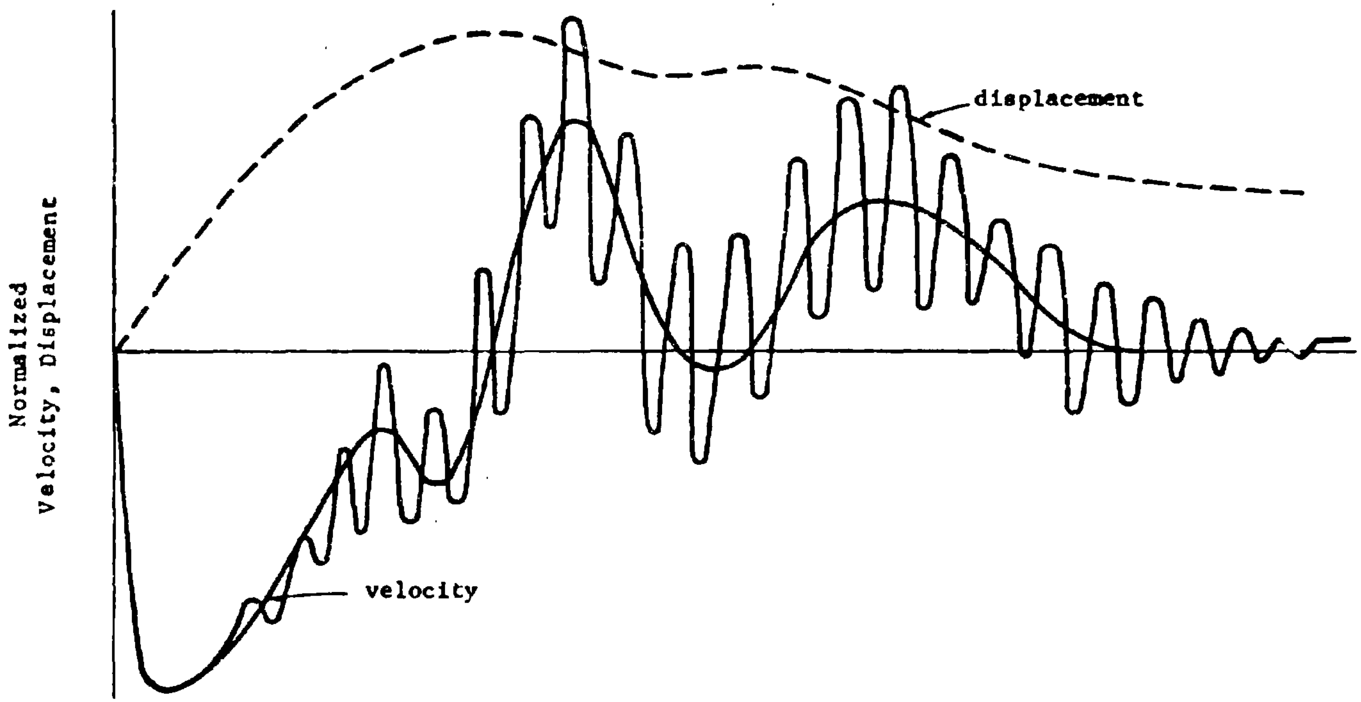

Figure 7. Combined shock and vibration waveform of equipment support.

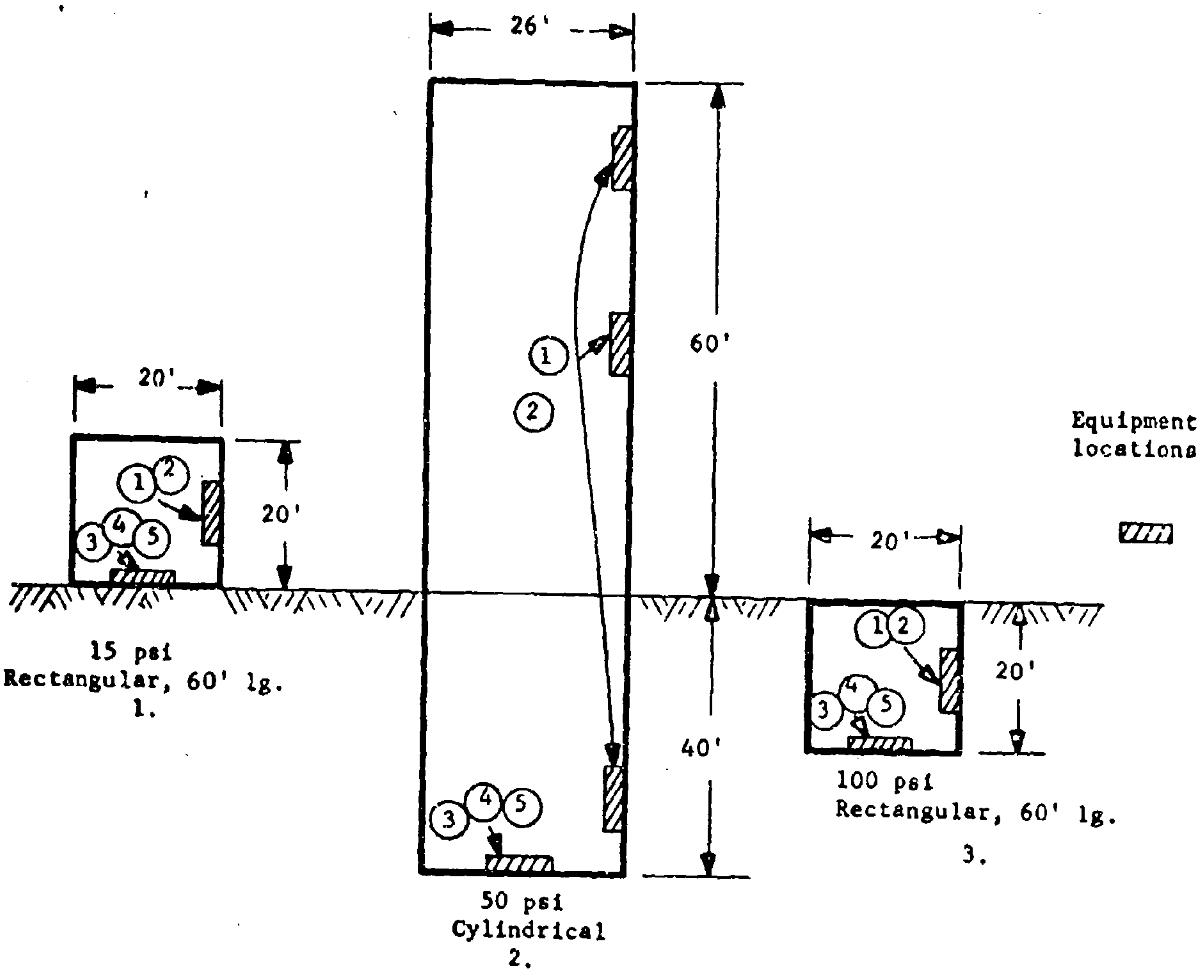

Figure 8. Typical protective shelter configurations.

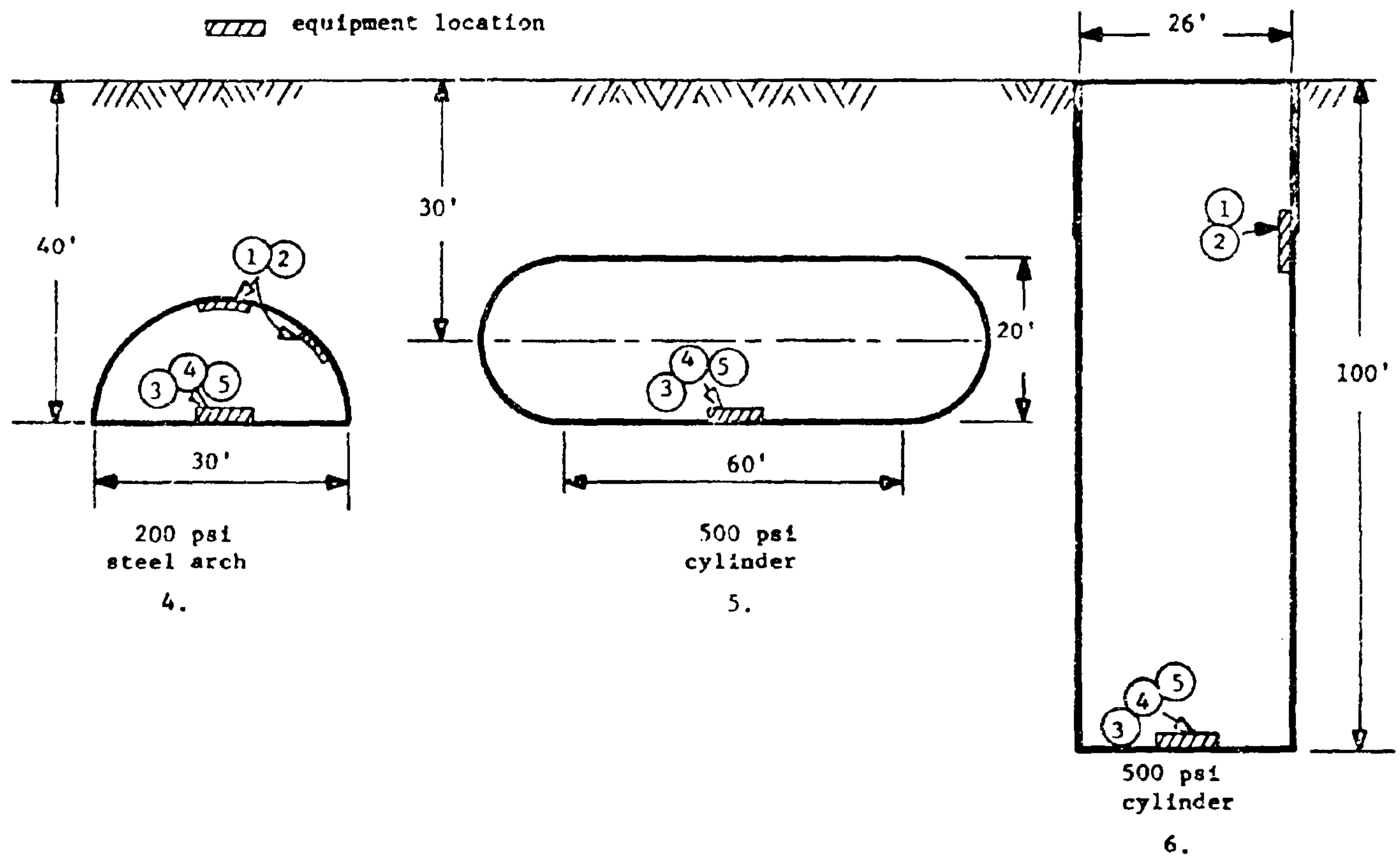

Figure 9. Typical protective shelter configurations.

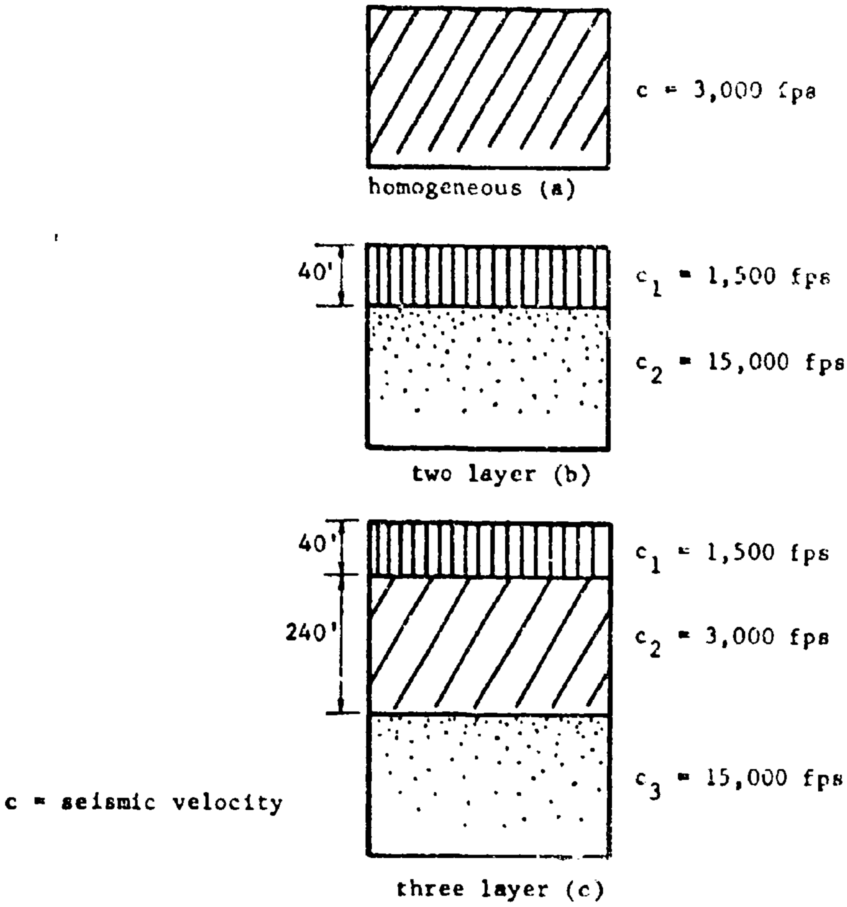

Figure 10. Soil profiles.

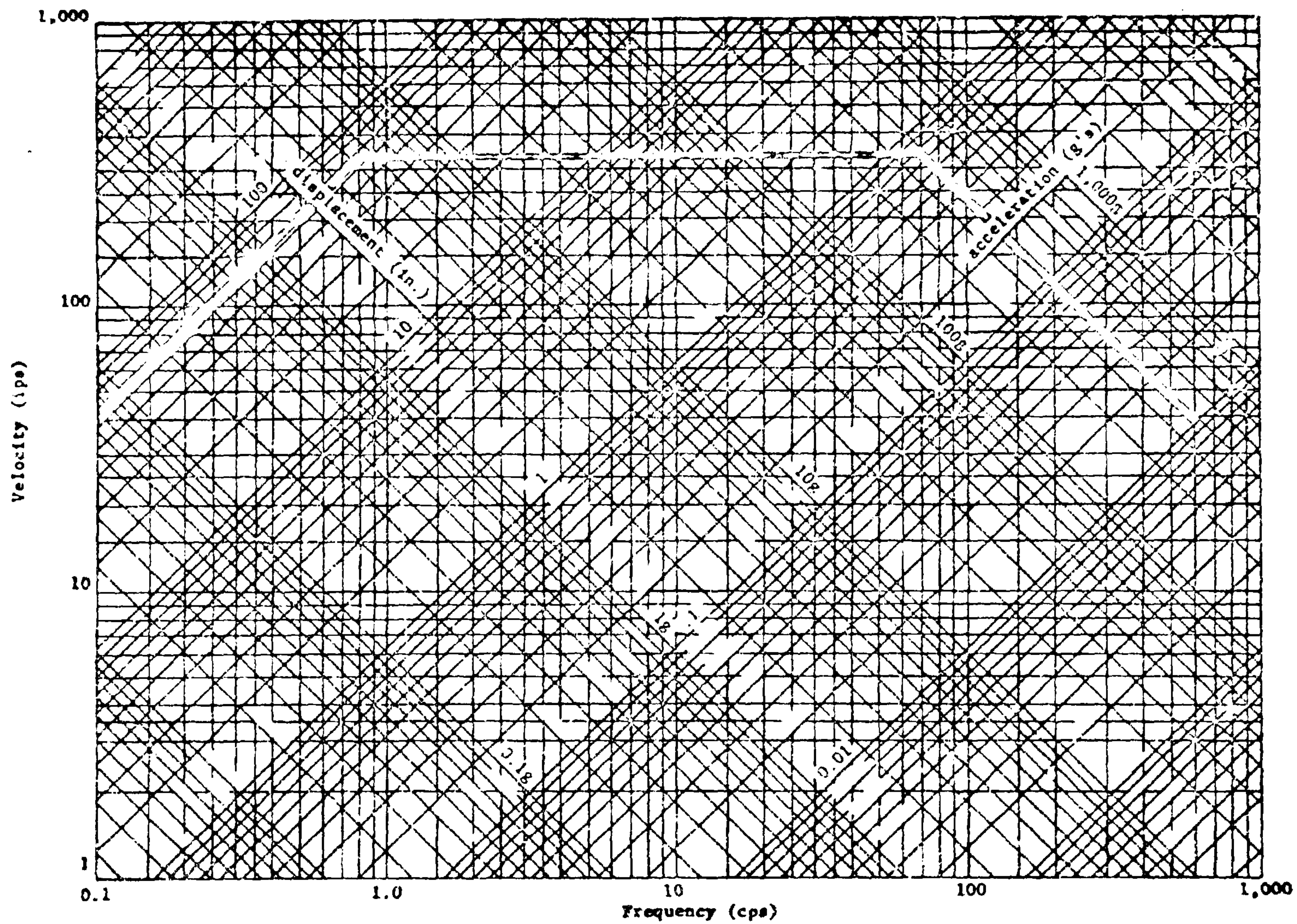

Figure 11. Peak motion spectrum.

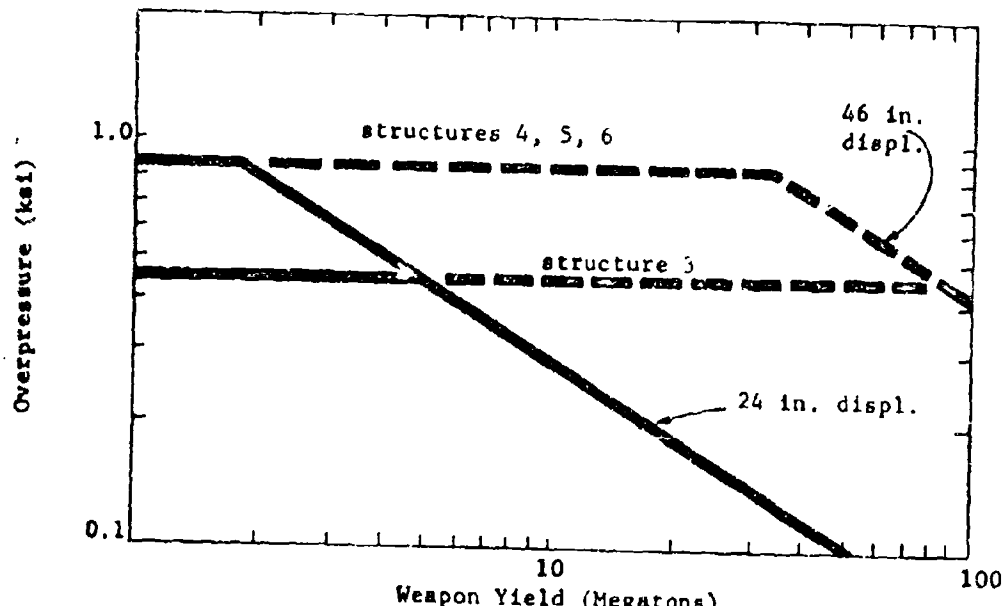

Figure 12. Site profile (a) simulation ranges.

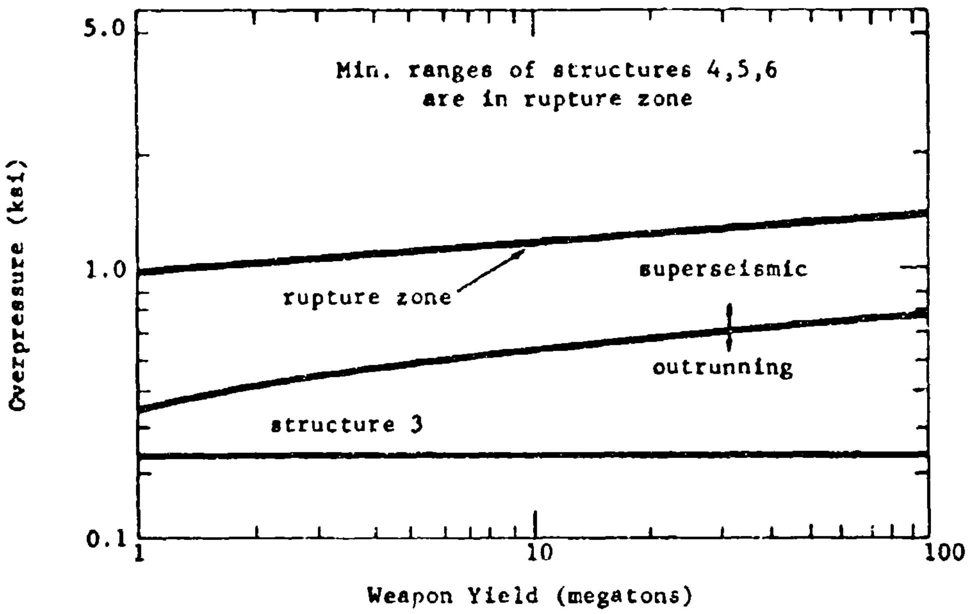

Figure 13. Site profile (b) simulation ranges.

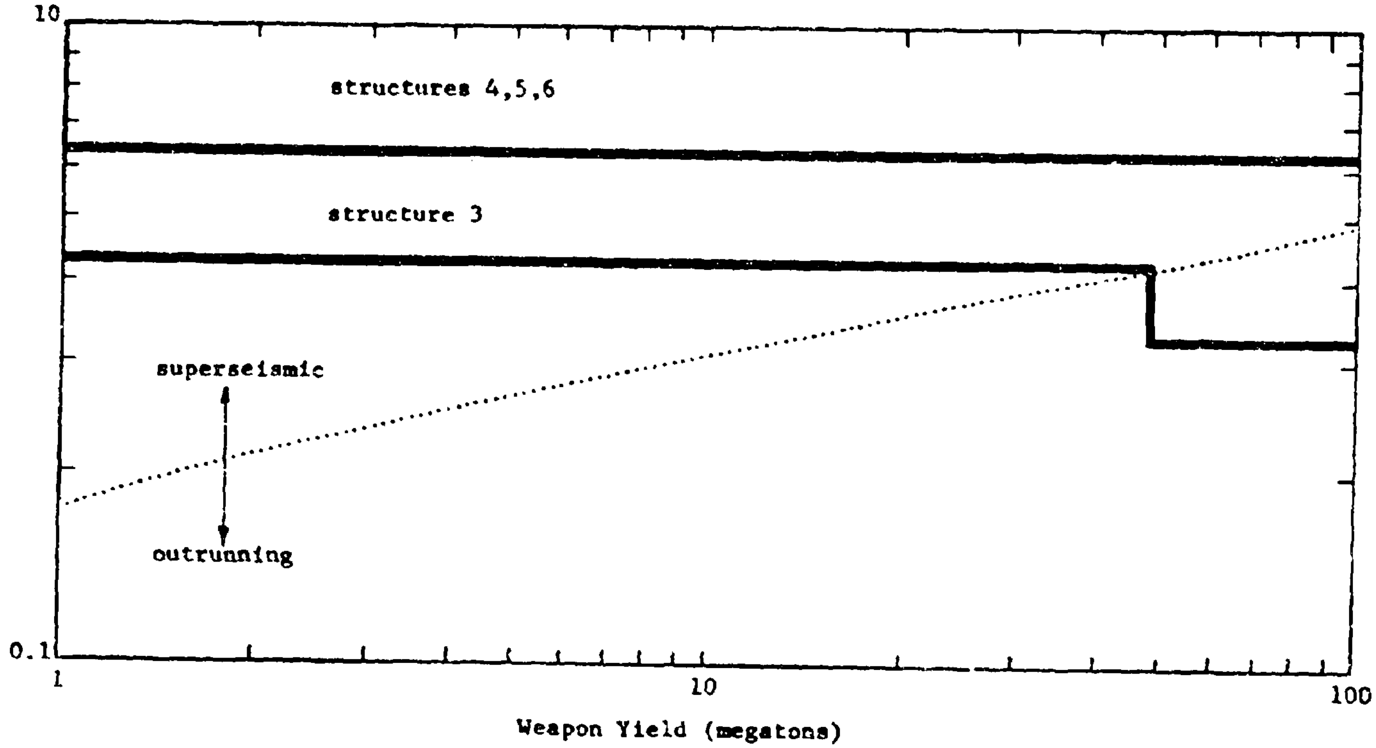

Figure 14. Site profile (c) simulation ranges.

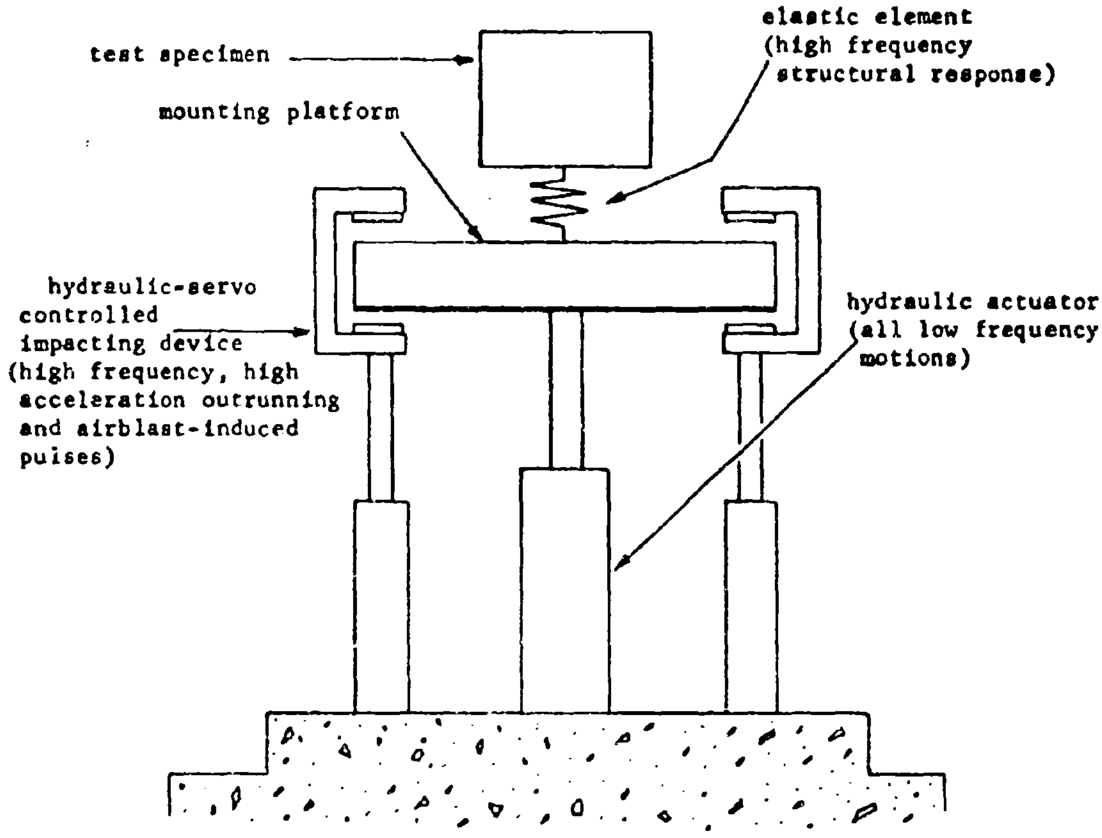

Figure 15. Shock simulator concept schematic.

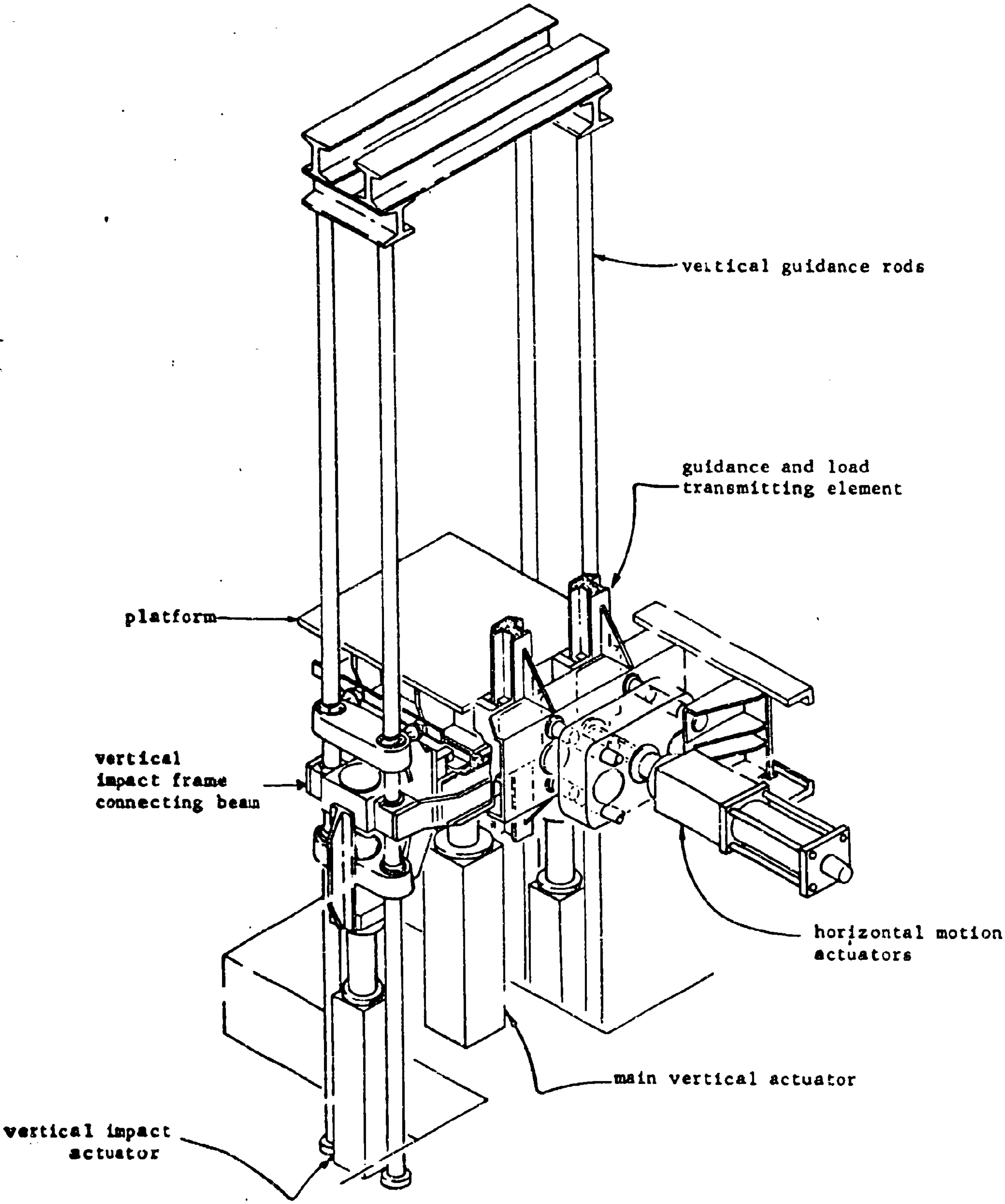

Figure 16. Isometric of simulator machine elements

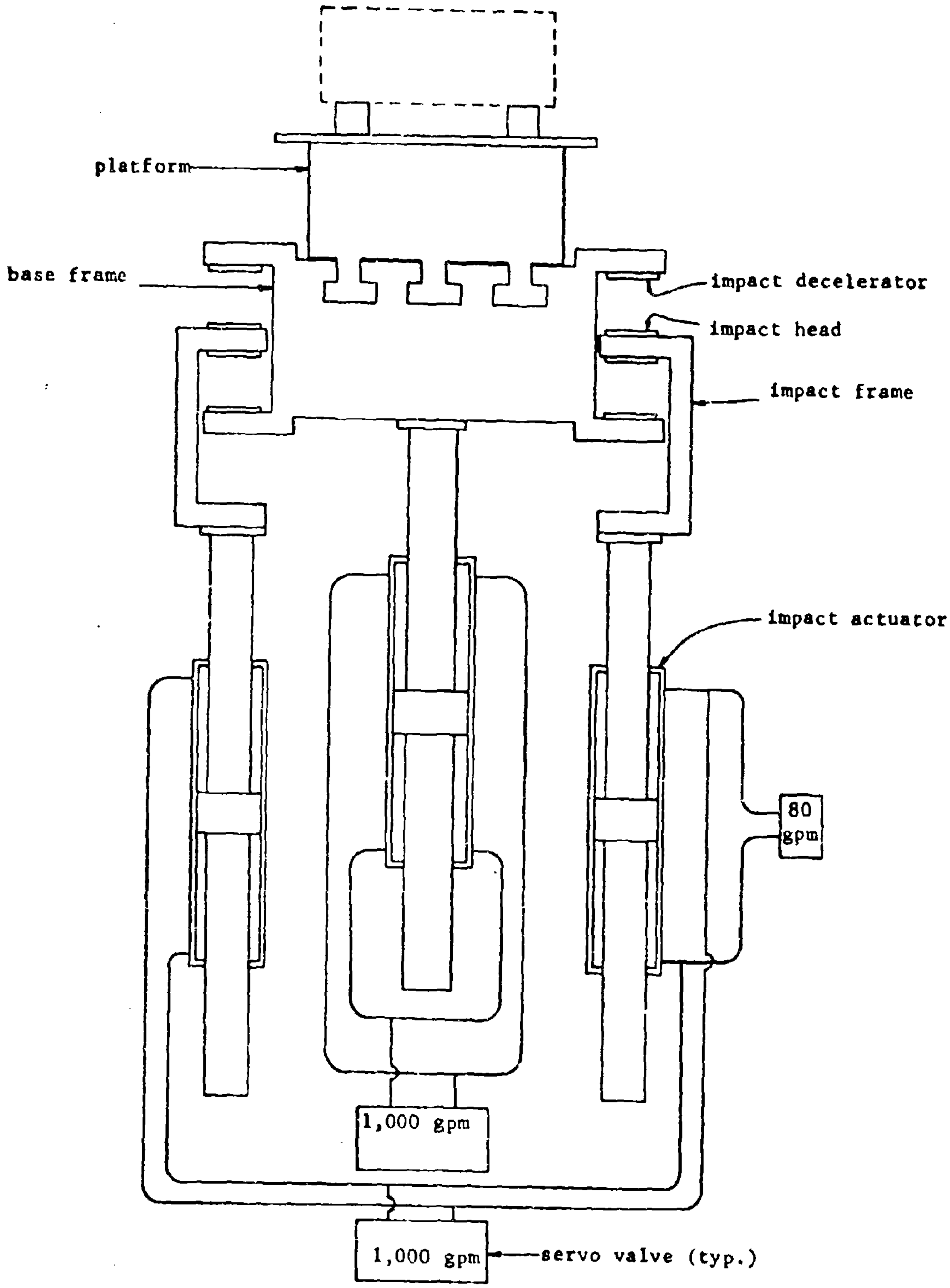

Figure 17. Schematic of principal vertical components of simulator.

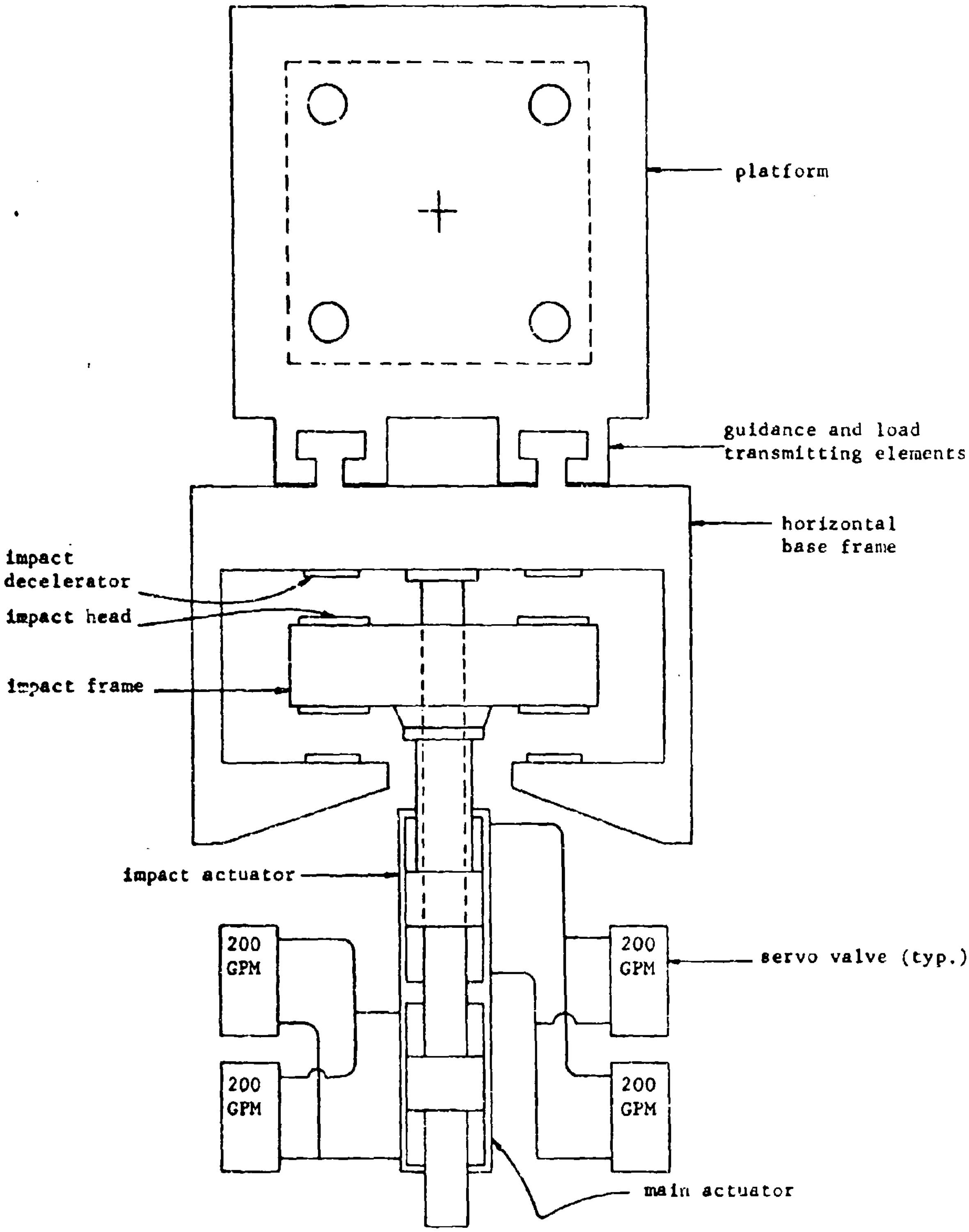

Figure 18. Schematic of principal horizontal components of simulator.

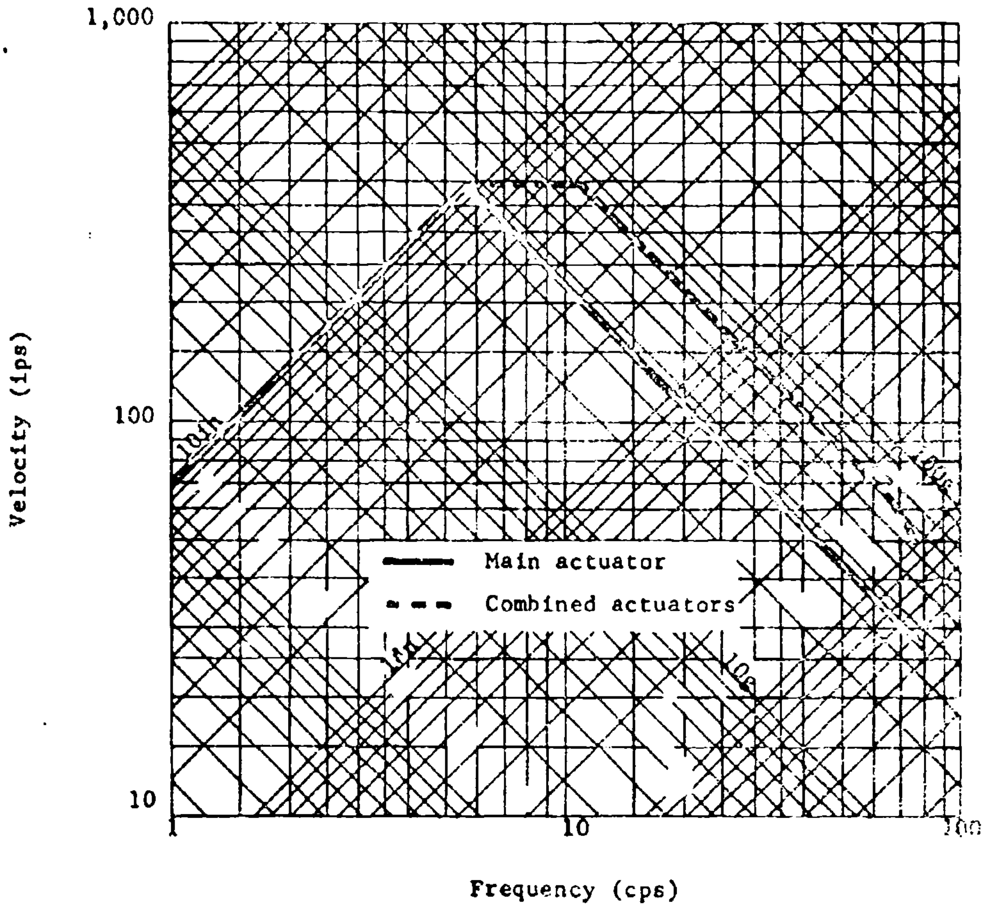

Figure 19. Simulator capability to produce a vertical sinusoidal vibration of a 400 pound specimen.

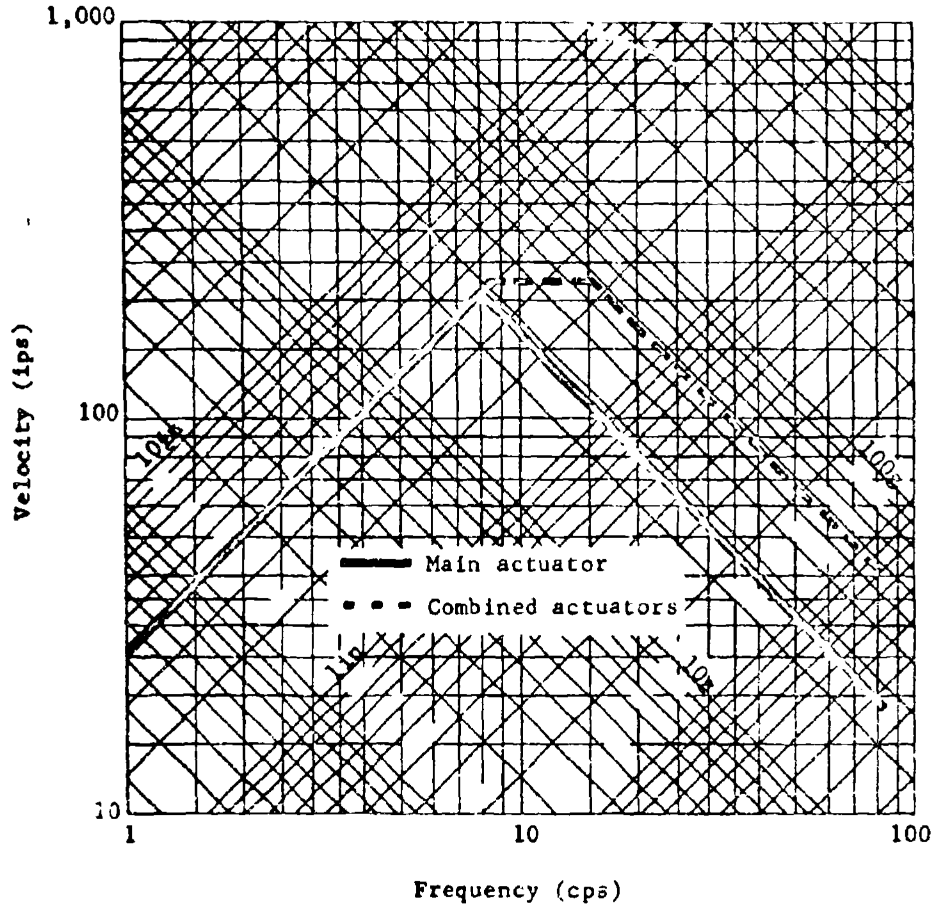

Figure 20. Simulator capability to produce a horizontal sinusoidal vibration of a 400 pound specimen.

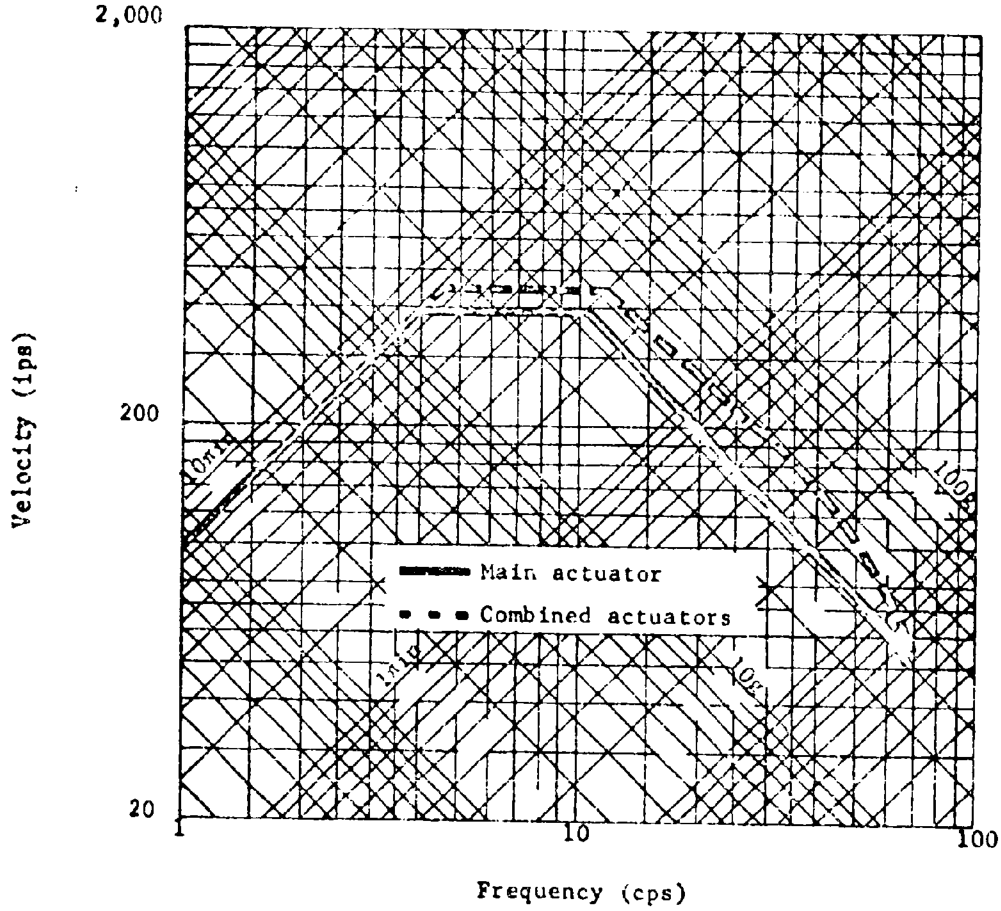

Figure 21. Simulator capability to produce a vertical half-sine acceleration pulse on a 400 pound specimen.

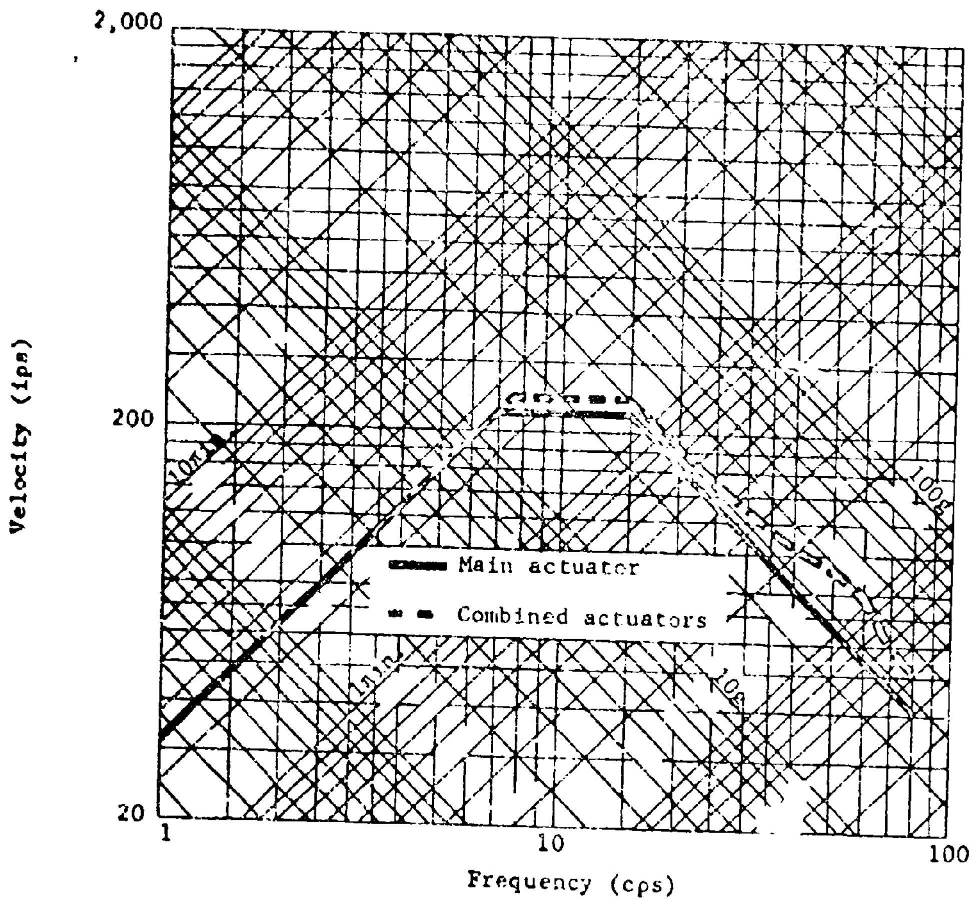

Figure 22. Simulator capability to produce a horizontal half-sine acceleration pulse on a 400 pound specimen.

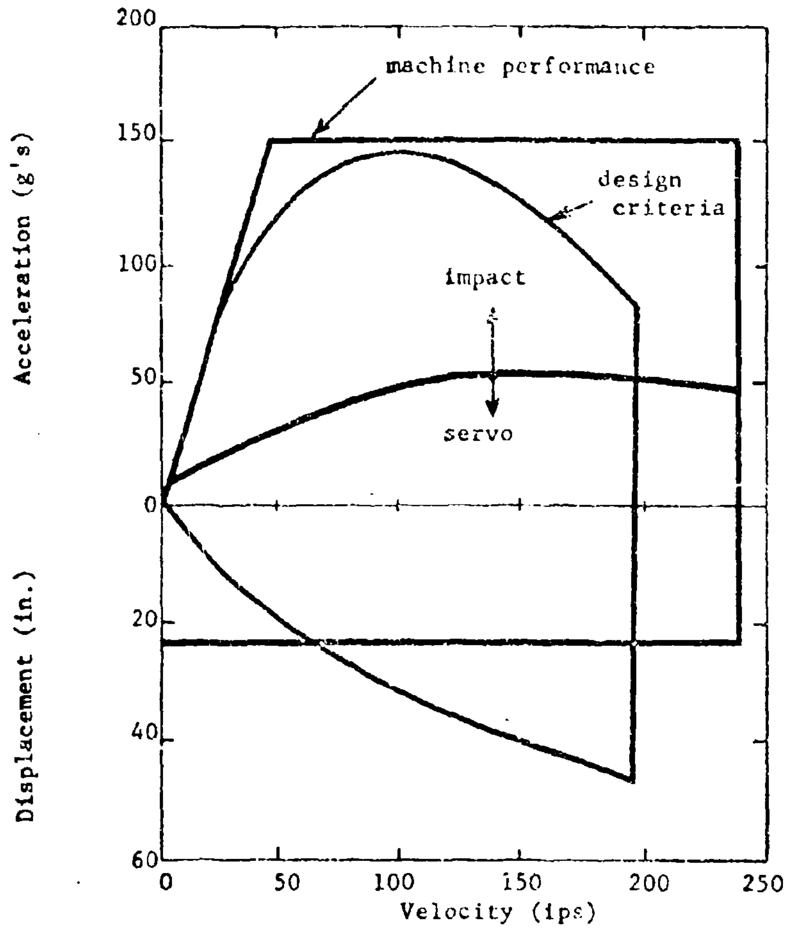

Figure 23. Vertical performance envelope comparison.

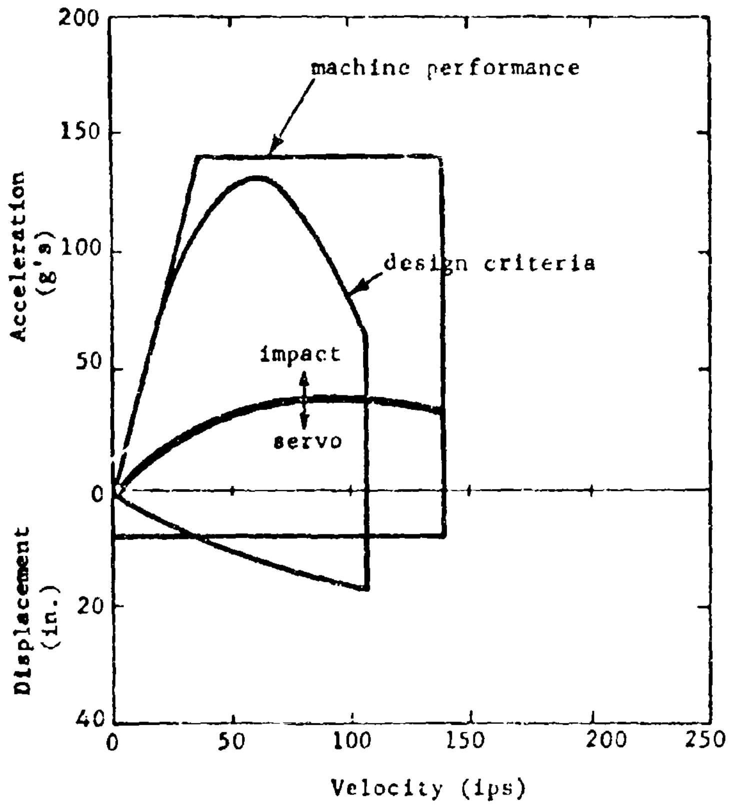

Figure 24. Horizontal performance envelope comparison.

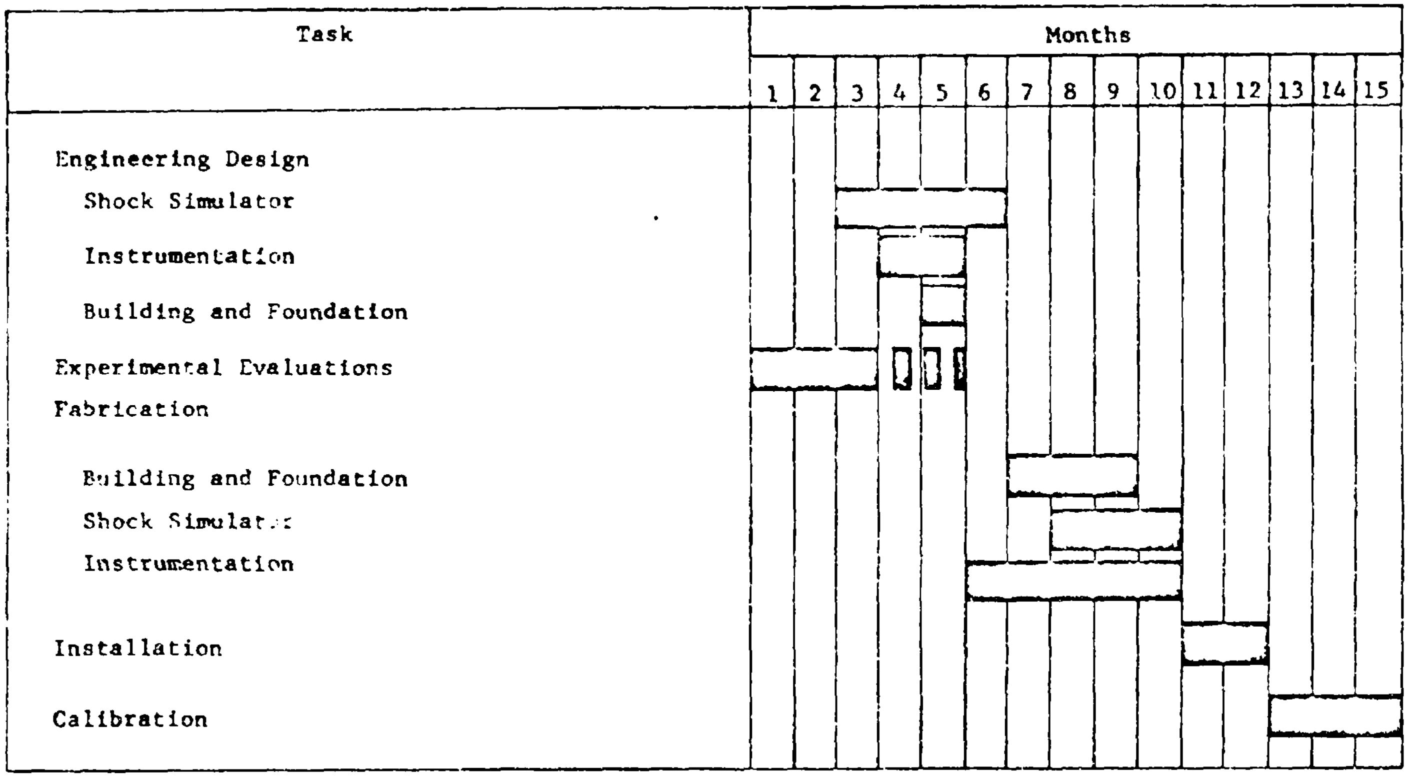

Figure 25. Estimated time schedule.