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NAVY HIGH-IMPACT SHOCK MACHINES FOR LIGHTWEIGHT AND MEDIUMWEIGHT EQUIPMENT

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ABSTRACT

Descriptions are given of the Navy HI shock machines for lightweight and mediumweight equipment. Shock motions are given for standard loading conditions. These are illustrated by acceleration-, velocity-, and displacement-time relations. Maximum values of velocities and displacements, and of accelerations passed by various low-pass filters, are presented. Shock spectra are presented for selected conditions. Equivalent displacement- and velocity-shock, together with maximum values of acceleration, can be established for their respective effective frequency ranges from observations of the shock spectra.

Concepts relative to the specification of shock tests are considered. These include brief considerations of analyses of shock motions, methods of specifying a shock test, and what is meant by simulation of field conditions. It is indicated that shock tests should not be specified in terms of shock motions, or spectra, unless the values specified be considered only as nominal values.

PROBLEM STATUS

This is an interim report on one phase of the problem; work is continuing.

AUTHORIZATION

NRL Problems F03-02 and F03-08
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INTRODUCTION

Object

This report will consolidate information contained in previous publications, many of which are out-of-print, relating to the characteristics and use of Navy HI (High-Impact) class shock machines. In addition the report will present recent views relating to shock tests and test procedures.

There are presently two principal classes of Navy HI shock machines: the HI Shock Machine for Lightweight Equipment (1-5) and the HI Shock Machine for Mediumweight Equipment (5-7). Equipment for use on naval ships is classified as lightweight if its weight does not exceed 250 lb, and as mediumweight if the weight is between 250 and about 5000 lb. These two classes of machines were designed primarily to simulate shocks probable on shipboard. Their characteristics and performances will be considered in detail. Other shock machines, such as the Shock Machine for Electronic Devices (8,9), the JAN-S-44 shock machine (10), various air guns (11-13), and drop-tables (11,14-16) are also used by the Navy in common with other services, but will not be specifically considered here.*

Early History

Prior to the early stages of the Second World War the major causes of shock to equipment aboard ships were direct hits by enemy shells, torpedoes, and the firing of the ships' own guns. It was then generally conceded that the only practical protection of equipment against enemy action was to mount the equipment as far as possible from the hull plating and to use as much armor as possible. A "3 ft-lb" (17), a "250 ft-lb" (18), and a combination roll, shock, and vibration machine (19) were developed during this period to simulate ship environments caused by the action of its own machinery and ordnance and to generally improve equipment reliability.

During the Second World War the use of underwater mines that exploded some distance from a ship often resulted in little structural damage to the ship but caused considerable shock damage to equipment in the ship. This was in consequence to the large area of the hull that was exposed to the underwater pressure pulse. In addition, the greater quantity of, and the greater reliance on, electronic and other complex equipment for the operation and control of the functions of a ship required greater reliability for this equipment.

In 1939 the British developed a shock testing machine for lightweight equipment that produced damage to items under test that was similar to that caused by shipboard shock. The U.S. Navy shock machine for lightweight equipment was then designed similar to that of the British, and the first such machine was built in 1940. In order to perform shock

*More complete general descriptions of shock machines are given in Chapter 26 of Ref. 11 and in Ref. 12.

tests on heavier equipment the shock testing machine for mediumweight equipment was designed and built in 1942. The shock outputs of the two classes of machines were such as to have about the same maximum values of acceleration, velocity, and displacement for an equivalent shock condition.

EQUIVALENCE OF SHIPBOARD SHOCK AND ITS LABORATORY SIMULATION

One of the most characteristic features of shock motions is their infinite variety. It is neither desirable nor practical to construct a shock machine which produces a shock motion equivalent to that of a given field condition. Rather the shock motion generated should have a damage potential at least as great as any probable field shock for which protection is required. The shock machine, therefore, is not designed to simulate a given field condition; and the question frequently posed, as to how accurate this simulation is, cannot be given a sensible answer.

Nevertheless it is presently the objective of shock tests to provide an accurate simulation of field conditions – not a given field condition, but a shock motion that possesses the important characteristics of all probable field shock motions. This objective requires that sufficient field data be obtained so that they can be treated in a statistical manner. The field data must be analyzed so that their damage potentials can be assessed. An envelope of all of the values of shock intensity, or their damage potentials, obtained from field measurements is drawn. A shock machine must then be devised which will provide a shock motion represented by the envelope. This simulated shock motion then has, for the methods of analyses used, a damage potential equal, at least, to the maximum values encountered under any probable field condition for which protection is required.

This statistical approach, at first glance, appears straightforward and valid. And so it is for equipment that is relatively light. However, it is known that equipment reacts to foundation motions in a manner similar to dynamic vibration absorbers (20), and that the equipment will reduce the amplitude of frequency components of foundation motions that are equal to "fixed-base"* natural frequencies of the equipment (21-23). It is also known that the frequency components of shock motions that are the most damaging to an equipment are those that correspond to the equipment fixed-base natural frequencies. Thus in a statistical study of damage spectra (shock spectra), where an envelope of the maximum shock spectra values of all appropriate field shock motions are normally used to indicate the spectra of a suitable simulated shock motion, the simulated motion would be oversevere for relatively heavy† equipment. For in the statistical approach the envelope is determined by the maximum values. But equipment reactions cause values to be systematically at their minimum at only the frequencies that are important for damage considerations. The statistical approach often results in impossibly high design requirements which may be an order of magnitude above the true requirements. Such factors as these indicate that the science of shock testing is still in an youthful stage and that considerable judgment may be necessary in formulating tests, and that considerable changes may be expected in shock machines and procedures in the future.

*The natural frequencies of the equipment when the foundation to which it is attached is infinitely rigid and heavy, i.e., the foundation is fixed, or does not move.

†An equipment is relatively heavy when its effective weight is sufficient to considerably influence the shock motions. This is dependent upon the relative mass of the equipment and the foundation as well as upon the internal damping and the modes of vibration of the equipment. It may be that equipment having weights less than 100 lb might sometimes be termed "heavy" under this definition.

ANALYSES OF SHOCK MOTIONS

Some examples of commonly idealized shock motions, and a complex shock actually experienced, are shown in Fig. 1. It is difficult to use such information as is given for complex shock, it is even difficult to tell when one type of shock is more severe than another. In order that the damage potential of one shock motion can be compared with that of another some quantitative means of measuring the damage potential must be devised. Various means have been used but, perhaps, the following predominate: shock spectra, velocity-shock, and simple shock pulses.

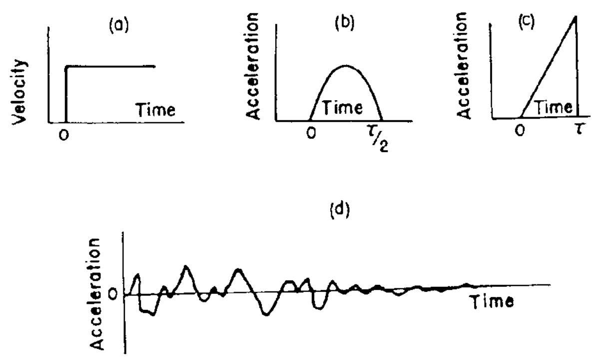


Fig. 1 - Different types of shock motions: (a) velocity-shock, a step-velocity change; (b) half-period sine acceleration pulse; (c) sawtooth acceleration pulse; (d) complex shock wave. The HI shock machines provide complex shocks.

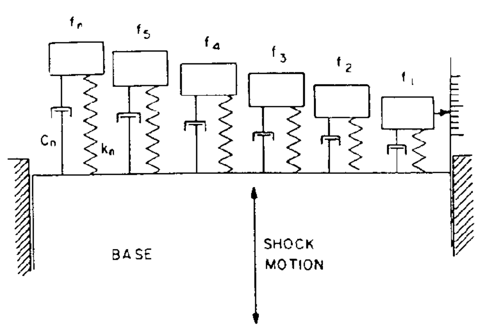


Fig. 2 - Series of single-degree-of-freedom systems of different natural frequencies. The maximum responses of these elements to a motion (shock) of the base, plotted as a function of their natural frequencies, is a shock spectrum of that shock.

Shock Spectra

A series of mass-spring systems (single-degree-of-freedom, or simple systems) is shown in Fig. 2. Each system has a different natural frequency so that together they cover a frequency range of interest. Each system has the same fractional part of critical damping. They are all attached to a common base which is made to partake of a shock motion that is to be analyzed as to its damage potential. A curve representing the maximum responses of each of the systems to the shock motion, expressed as a function of the natural frequencies of the systems, is the damage potential of that shock to the series of simple systems. This curve is called a "shock spectrum." The use of shock spectra, besides having other valuable applications, has become an acceptable means of expressing in a quantitative and standard manner the intensity and nature of the damage potential* of shock motions (24). Shock spectra can be expressed as the maximum relative displacement of the masses (Fig. 2) with respect to their base, as the maximum absolute values of the accelerations of the masses, or as pseudo maximum relative velocities. If the maximum relative displacement response is given as X, then the displacement, velocity, and acceleration spectra (or responses) have the relation $X:2\pi fX:(2\pi f)^2X$. This relation is only approximate and applies accurately only if the damping is negligibly small. In specifying a shock spectrum the amount of damping should be given; if it is not, then the amount is assumed to be zero. The spectra are usually plotted as the maximum values irrespective of the sign of the response. However, somewhat more information is available if both the positive and negative maximum values of responses are plotted to provide positive and negative shock spectra.

Examples of shock spectra are shown in Figs. 3 through 6. In Fig. 3 the acceleration of the shock motion are shown by the inset curve. The ordinate represents acceleration in units of gravity. The maximum accelerations experienced by the masses (Fig. 2) for several given amounts of damping, when the base is subjected to the accelerations shown in the inset, are shown by the three principal curves. It is thus noted that the shock spectra do not represent the shock motion, rather they show what the shock motion does to a standard set of simple systems. The shock spectra of the simple pulses of Fig. 1 are shown in Figs. 4, 5, and 6.

The four sets of coordinates of Fig. 4 are of particular interest. The three ordinate values, velocity, acceleration, and displacement, represent amplitudes of sinusoids and so are all dependent upon each other. If the velocity amplitude, V, is assigned, then the displacement amplitude is $X = V/2\pi f$, and the acceleration amplitude is $2\pi fV$. These will appear as straight lines with 45-degree positive and negative slopes respectively for the method of plotting employed. This coordinate system is not only a convenient way to simultaneously represent the three types of shock spectra, but also represents directly the shock spectra of velocity-shock. Velocity-shock is discussed in the following section. The numerical value of a velocity-shock spectra is equal to the magnitude of a step-velocity change (for conditions of negligible damping) and is independent of frequency. The corresponding acceleration and displacement shock spectra are given by the sloping coordinates.

Figure 5 illustrates shock spectra for a half-period sine pulse. A curve representing the maximum positive values would be the positive shock spectrum, as previously defined, and a curve representing the maximum negative values would be the negative shock spectrum. These are shown respectively by the curve labeled "primary" and by the curve below the zero axis. However, as illustrated, the responses during the time of the pulse are separated from those following this time and are respectively identified as "primary" and "residual." In Fig. 6 the same separation is provided for the shock spectra of a sawtooth pulse. A primary shock-spectrum is defined as the maximum response of the simple systems (Fig. 2) during the time of the shock. The residual shock spectrum is the maximum response after the completion of the shock. Positive and negative shock spectra may exist for both primary and residual shock spectra.

Figure 6 illustrates why a pulse of a sawtooth shape has become popular for shock specifications. Its positive and negative shock spectra are equal and rise smoothly to a maximum value, after which they remain close to a constant value. All symmetrical pulses, on the other hand, have different positive and negative spectra with the residual spectra periodically becoming very small.

*The shock spectra represent damage potential for simple systems. The relative values may be different for nonlinear and more complicated systems.

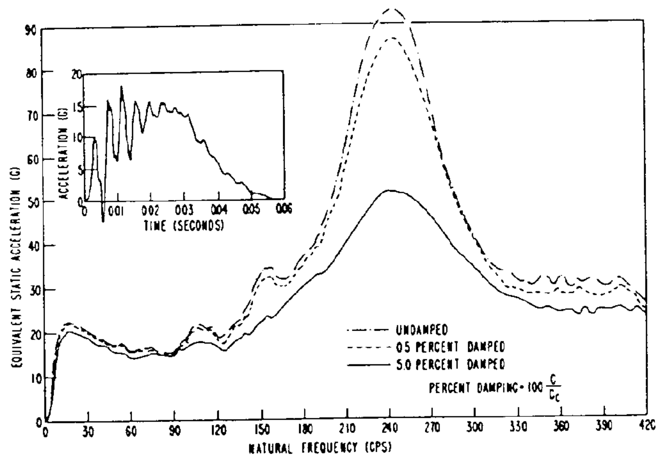


Fig. 3 - Acceleration shock spectrum for shock motion shown by inset figure. Both damped and undamped elements are considered. (After Ellett (25).)

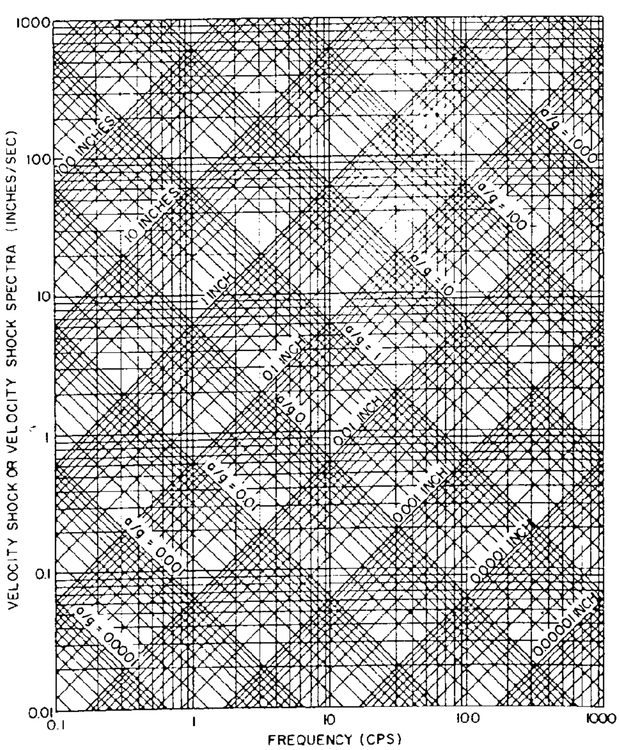


Fig. 4 - Shock spectra for velocity-shock. The velocity shock-spectra are numerically equal to the values of the velocity-shock (the magnitude of step-change of velocity). The displacement shock-spectra are given by the lines of 45-degree-positive slope and the acceleration shock-spectra are given by the lines of 45-degree-negative slope.

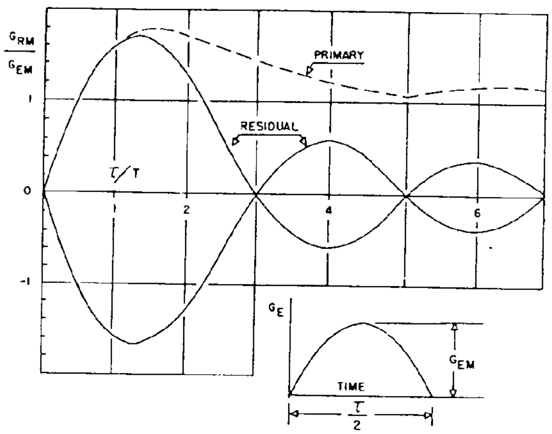


Fig. 5 - Shock spectra for a half-period-sine-wave acceleration-pulse. The pulse (inset) has an amplitude of G_{EM} and a duration of $\tau/2$. The maximum response is G_{RM}. The primary shock spectrum is the maximum response during the pulse interval and is positive (the same direction as the pulse). The residual spectra have equal positive and negative responses. See Ref. 26.

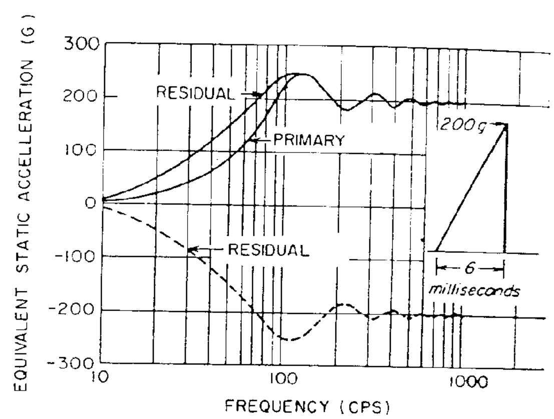


Fig. 6 - Shock spectra for inset sawtooth acceleration pulse. Primary and residual refer, respectively, to maxima that occurred during and after the pulse period. (After Morrow and Sargeant (16).)

Velocity-Shock

The most important characteristic of many types of shock motions can be expressed as a step-change in velocity (27). This concept is sufficiently accurate for a large class of shock motions and, outside of purely static considerations, is the most simple in its practical application. The velocity-shock is expressed quantitatively as equal to the magnitude of a step-change of velocity of the excitation as shown by Fig. 1(a). Frequently an equivalent velocity-shock is taken as the maximum change of velocity of the center-of-mass of an equipment subjected to a shock. It is to be noted that the velocity-shock spectrum of a velocity-shock (no damping), as shown on Fig. 4, is numerically everywhere equal to the value of the velocity-shock.

The use of a single number to describe the intensity of shock over given ranges of frequencies can be useful and sufficiently accurate for many applications. The term velocity-shock is one such number. Obviously under real conditions, where displacements and accelerations remain finite, the spectra for velocity-shock involving a finite step-time must decrease below and above certain frequency limits. Shock spectra, shown later in Figs. 28-29 and 42-43, give a middle range of frequencies where the velocity-shock spectra is about constant. The single numbers corresponding to these values can be given as the equivalent velocity-shocks for these frequency ranges.

For some lower frequency range one observes a constant-displacement shock spectra. The value of this displacement is equal to the value of "displacement shock" over that frequency range, where displacement-shock is defined as a step-change of position and its magnitude is equal to the magnitude of the step-change. The shock spectrum in this region is equivalent to the displacement-shock, and is equal to the maximum displacement involved in the shock motion.

Above a certain frequency the acceleration shock spectra becomes constant. For this frequency range the peak, or maximum, value of acceleration present in the shock excitation is equal to the acceleration shock spectra. This condition exists when the highest significant frequencies associated with the shock motion are low compared with the shock-spectra frequencies considered. Thus the accelerations can be considered as equivalent to static values.*

Simple Shock Pulses

When the time for the velocity change associated with a shock motion cannot be considered short (compared with periods of significant modes of vibration of an item subjected to the shock), then it is sometimes sufficient to construct a shock motion of mathematically simple shape which will be equivalent in important aspects to some shock motion of interest. Some of these pulses are shown in Fig. 1. Responses of simple systems to a variety of type of pulses can be found in Ref. 28. An application of this type is to obtain a simple shock pulse which has a shock spectrum approximately equal to an envelope of maximum values of a statistically significant quantity of shock spectra obtained from field measurements. A shock machine, built to provide this pulse, would then provide a simulation of the damage potential of the field environment. In addition the responses of an item to the pulse excitation can be theoretically determined more easily. As has been mentioned, there are valid objections to the statistical procedure involved in establishing the shock spectrum, because of the effects of equipment reactions in modifying the shock motions.

Miscellaneous

It is acceptable under certain conditions (29) to give maximum values of accelerations or velocities and associated frequencies which one observes in a shock motion. Fourier integral and series techniques have also had limited use. The Fourier integral method has considerable potential value.

SPECIFYING A SHOCK TEST

Shock tests can be specified by three methods (12): First, a shock motion can be specified. A shock test then consists of causing the points of attachment of the item under test to partake of this motion. Second, a shock spectrum can be specified. A shock test then consists of causing the points of attachment of an item under test to partake of a motion that has this spectrum. Third, a shock machine can be specified together with a procedure for its operation. A shock test then consists of mounting an item under test to the machine in a prescribed manner and of operating the machine according to the given procedure.

*It would be interesting to include "acceleration-shock" for the upper range of frequencies, where acceleration-shock would be defined as the magnitude of a step-change of acceleration; however, the shock spectrum of an acceleration-shock would be equal to twice the value of acceleration-shock, which is not analogous to the similar situation for displacement and velocity.

The first and second methods of giving specifications are somewhat similar, although the second places more burden on the test engineer, as he may be required to devise a shock motion corresponding to a given shock spectrum. However both methods are impractical of achievement unless the items under test are perfectly rigid or relatively light. The cause of the difficulty is the reaction of the load on the test machine. This reaction causes the applied shock motions to become dependent upon the nature of the equipment under test, so that, unless large tolerances are permitted, the test cannot practically be made as prescribed. The only practical solution to this problem is to consider that the numerical values of shock motions, or spectra, given in the specification be considered as nominal values. They should be complied with for rigid loads rigidly attached to the machine, and calibrations of the machine should be made to assure that this is so. Tests of real equipments should then be made according to prescribed procedures with no further concern as to the specified motions or spectra. However, care should be taken that sustained natural frequencies, not typical of field conditions, are not introduced by the shock machine and the mounting arrangements. This procedure is preferred, even though it would be possible to generate the specified motions, as this procedure may prevent the overtesting which would result if prescribed shock motions were maintained in spite of equipment reactions. If this procedure is followed, the shock machine should have a mechanical impedance, as seen by the equipment, at least as great as that of the structure to which the equipment will eventually be attached.

The third method of specifying shock tests requires that the agency responsible for the test provide a machine design which, when the machine is built and used in a prescribed manner, will provide a suitable shock motion. This reduces the trials and tribulations of the test engineer to relatively small proportions. The Navy HI machines are in this category. The design of the HI Shock Machine for Lightweight Equipment has been standardized (4) by the American Standards Association. Complete working drawings of the machine together with operating instructions are available from this source. Because of the small number of mediumweight machines in existence, it has been subjected only to Navy standards.

When a machine is specified for a shock test (the third method) it is the responsibility of those who specify the test to provide information as to its shock motions and their spectra. These are normally given only for rigid loads rigidly attached to the shock machine.

NECESSITY FOR PRECISION AND ACCURACY FOR TESTS ON SHOCK MACHINES

Great accuracy can seldom be justified for shock tests on the basis of field information. However three factors require that shock machines be constructed so that they can reproduce shock conditions with considerable precision: (a) Shock tests may be legal requirements for the acceptance of an equipment. Whether a test has been performed according to specifications within acceptable limits is of great concern because it involves whether or not the equipment is of acceptable quality for contract fulfillment and payment. (b) In developmental work it is not possible to tell whether or not a significant improvement has been made unless the magnitude and nature of the shock can be accurately repeated. (c) A shock machine at one location should be able to provide, within reasonable tolerances, the same test to a given equipment as would be provided by the same type of shock machine at another location.

The motion output of shock machines (that do not suffer plastic deformation of semipermanent parts) should not differ by more than 5 percent for frequency components below about 200 cps. Greater variation can be expected in the shock-spectra peak values, as these values are determined by damping losses in the machine. As the damping depends largely on bolt tightness and friction between surfaces this factor may show considerable variations. Considerable variations may exist at higher frequencies. If machine parts (such as the anvil of the HI Shock Machine for Lightweight Equipment) gradually deform with use, then greater variations in performance can be expected. The HI Shock Machine for Mediumweight Equipment suffers no appreciable permanent deformation of its part.

THE NAVY HI SHOCK MACHINE FOR LIGHTWEIGHT EQUIPMENT

Description

The HI Shock Machine for Lightweight Equipment (Fig. 7) consists of a welded frame of standard steel sections, two hammers, one of which drops vertically and the other swings in a vertical arc, and an anvil plate which may be placed in either of two positions. The combination of two hammers and two anvil-plate positions permits blows to be delivered in each of three mutually perpendicular directions without remounting the test equipment. Each hammer weighs 400 lb and may be raised to a maximum height of 5 ft above its impact position, to deliver a maximum of 2000 ft-lb of energy at impact.

The anvil plate consists of a steel plate measuring $34 \times 48 \times 5/8$ in., reinforced across its back surface by I-beam stiffeners. Steel shock-pads are welded to the top and side edges and at the center of the back face over the stiffeners at the points of hammer impact. For back and top blows the anvil plate is positioned across the main frame and rests on a pair of enclosed helical springs. It is constrained to a vertical position by a set of springs and through bolts bearing against the main uprights. Washers and spacers prevent binding of the anvil plate during top blows. For edge blows, the anvil plate is rotated 90 degrees around a vertical axis and is supported by rollers bearing against steel tracks. It is positioned by a set of springs mounted on the forward support-brace edge. Edge blows are delivered by the swinging hammer against the rear anvil-plate edge. In each of the three directions of hammer impact the anvil plate constraining springs are adjusted to permit 1.5 in. of forward motion against the springs before bottoming occurs against limit stops. Rebound springs for back and edge blows are also provided with limit stops, although these springs reach their solid height before the limits are reached. There are no rebound springs for top blows, the maximum spring extension being governed by a captive bolt.

Several standardized mounting plates have been devised which simulate the mounting conditions aboard ship. These are interposed between the anvil plate and the test equipment and provide a degree of flexibility and isolation to the shock motions in a manner similar to normal shipboard bulkheads and decks. Two mounting plates are used predominantly for specification shock tests (4,5): the 4A plate for bulkhead-mounted equipment and the shelf mounting plate for platform-mounted equipment. The former derived its name from its figure number in shock-test specifications (5a) and is a flat steel plate $27 \times 34 \times 1/2$ in., while the latter is a similar plate to which a reinforced shelf has been welded. Reinforced 4-in., 13.8-lb car-building channels along the vertical edges space the mounting plates away from the anvil plate. Holes are drilled in the mounting plates as required to mount the test equipment centrally. The plate is discarded when the holes from previous tests become too numerous.

Loading Arrangements for Reported Characteristics

Reference 26 gives the results of an investigation of mechanical shock on the machine. Detailed drawings of the load apparatus used are given in this reference. The total weight capacity of the machine was covered by two separate load assemblies, the lighter covering

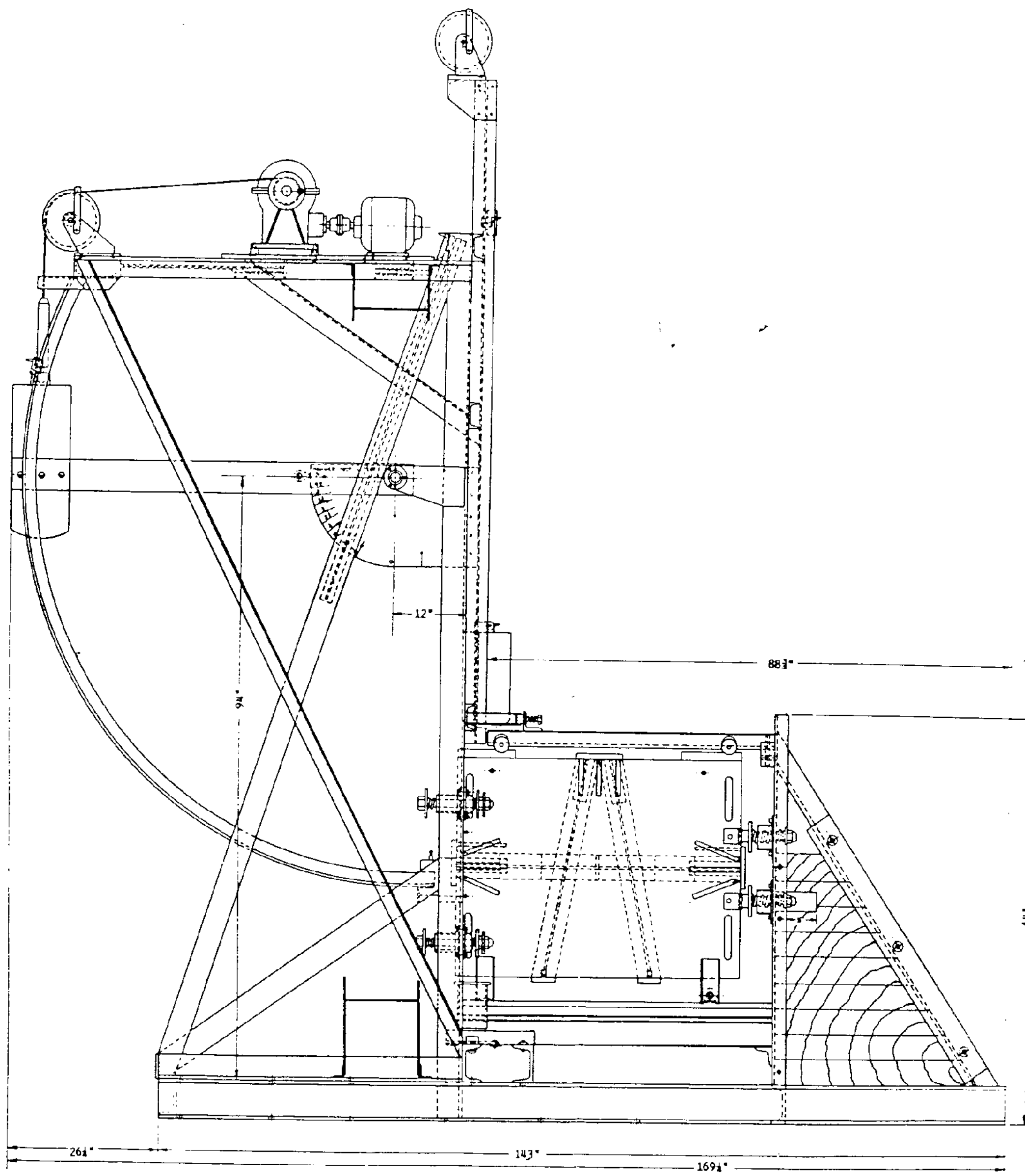


Fig. 7 - HI Shock Machine for Lightweight Equipment.
The anvil plate is oriented for an edge blow.

the range up to 200 lb, and the heavier from 200 to 400 lb.* A rugged welded frame comprised the base assembly in the light range to which additional steel plates were bolted to increase the load weight in small increments of approximately 25 to 50 lb. In the heavier range the load consisted of two sections of solid steel plating, one weighing approximately 200 lb and the other 125 lb. These sections could be used singly or in pairs to alter the load weight. Each load was drilled to accommodate the same rectangular mounting bolt pattern so that the load distribution on the mounting plate remained the same for all loads. Two-inch cylindrical pedestals spaced the load weight away from the mounting plate to prevent binding. Examples of the loading arrangements are shown in Figs. 8 and 9. The loads and their mounting supports can be considered as relatively rigid compared with the mounting plates (4A or shelf plate) of the shock machine.

Instrumentation and Measurements

Measurements were made of velocity and acceleration as a function of time, and of shock spectra as indicated by a multifrequency reed gage. Instruments for these measurements were bolted directly to the load assembly and oriented to measure the shock motions in the direction of the hammer impact. A second accelerometer was maintained in a fixed position on each mounting plate, and provided an indication of the magnitude of shock motions at these specific locations.

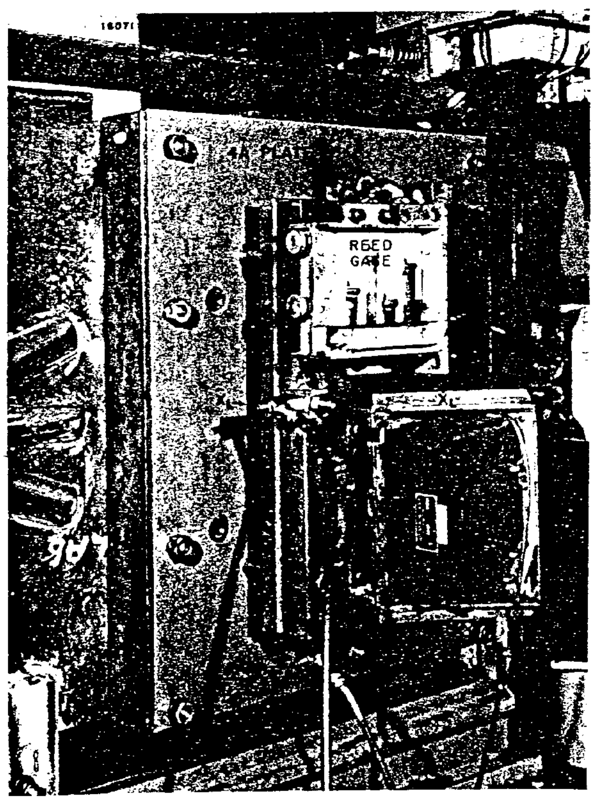


Fig. 8 - 4A plate with 389-lb load. The instruments are oriented for an edge blow.

*Normal specifications limit the load to 250 lb.

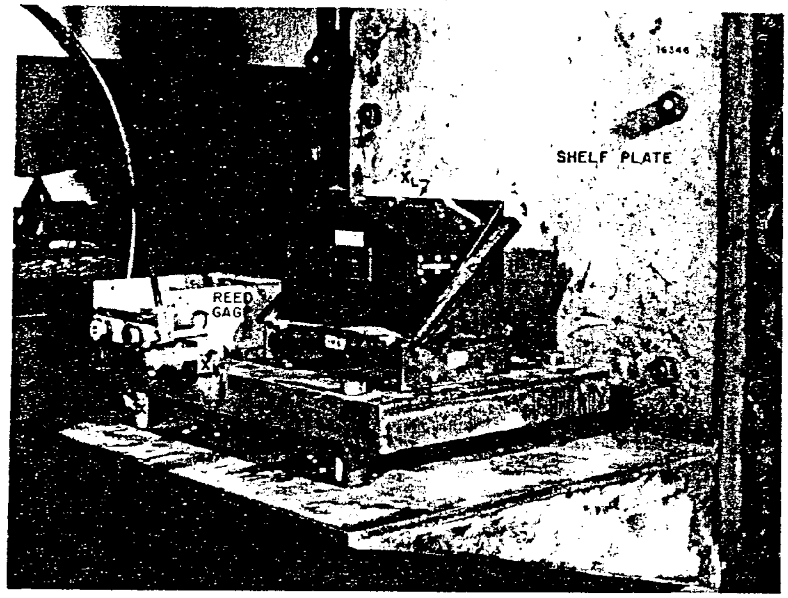


Fig. 9 - Shelf plate with 261-lb load. The instruments are oriented for a back blow.

With the possible exception of the reed gages, the instruments were standard types whose characteristics and limitations are well known and which have proven satisfactory for shock measurements. Instrument locations and details of their mounting adaptors may be seen in Figs. 8 and 9. Standard 300-, 1000-, and 5000-cps low-pass filters, incorporated in the accelerometer preamplifiers, limited the upper frequency response of the acceleration signals by removing accelerometer resonances and the higher frequency acceleration components which are of little importance, i.e., have little damaging value, yet predominate in the unfiltered records. The filtered output signals were displayed simultaneously on a multichannel cathode-ray oscillograph and recorded photographically by a moving-film camera as shown in Fig. 10. Sufficient recording channels were available to permit recording each accelerometer signal on two channels at the same time, using different sets of filters. Thus, the 1000-cps filtered record was recorded for every blow, while the paralleled channel alternated between a 300- and a 5000-cps filter. The velocity pickup signal was unfiltered and included an acceptable frequency range from about 6 to 2000 cps.

Accelerations, Velocities, and Displacements - The shock-motion waveforms, produced by the Shock Machine for Lightweight Equipment exhibit the same general characteristics for different heights of hammer blow delivered to a particular load arrangement in a given direction but greatly modify their characteristics with changes in direction of blow, load weight, load orientation, and mounting plates. Previous history of the anvil and mounting plates also affect the shock waveforms, but to a much lesser extent, by its influence on the plate stiffnesses and shape. These change because of work hardening and plastic deformation.

The acceleration traces for both the load and mounting plates are shown on the typical test record of Fig. 10. In general, the maximum value of acceleration occurs shortly after impact; it is followed by irregular vibratory modes. Characteristics of the motion remain similar, except for amplitude, if only the height of hammer drop is varied. The magnitude and frequencies of the accelerations which follow the maximum value are greatly influenced

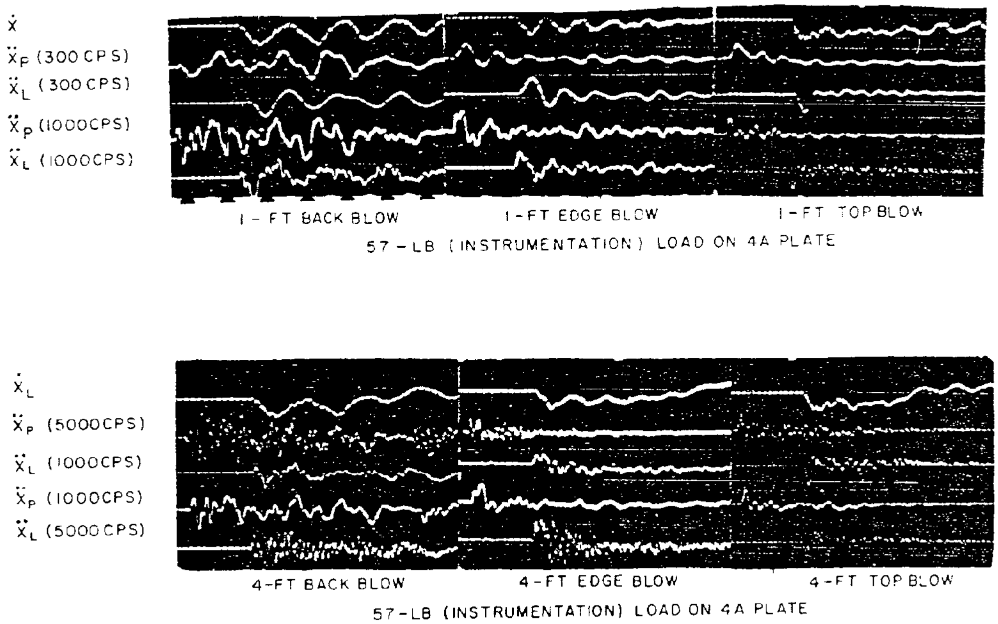


Fig. 10 - Typical recordings of shock motions: $\dot{X}_L$-velocity of load; $\ddot{X}_L$-acceleration of load; $\ddot{X}_P$-acceleration of 4A plate. 5000, 1000, and 300 cps refer to the low-pass filter through which the acceleration signals were passed.

by a large number of factors such as bolt tightness, mass distribution, and energy dissipation. Slight variations in any of these factors produce large changes in the waveform after the first pulse. Records taken using the 5000-cps filter are worst in this respect, since the high frequency components are most easily changed.

With information given as to the general waveform of the shock motions, it is sensible to plot maximum values of acceleration and velocity in order to provide comparative values for the shock motion. Figures 11 through 13 give maximum values of acceleration of the load and 4A plate and Figs. 14 and 15 give acceleration values when the shelf-plate mounting arrangement is used. As the load was essentially rigid the location of the accelerometer on the load was not of great importance. However, the value of the acceleration of the 4A plate was strongly dependent upon the accelerometer position. The apparent erratic trends of 4A plate acceleration curves are caused by the changes of mode shapes of the plate for different loads. For a complete description of results and the factors involved see Ref. 2.

Maximum values of load velocity are plotted in Figs. 16 through 21. For any given load these values increase approximately linearly with the hammer impact velocity, or as the square root of the height of drop. This approximation is also roughly true for the maximum values of acceleration.

The displacement-time motions of the load can be calculated from the velocity records. These motions have been plotted and are given in Fig. 22 for two different loads mounted on the 4A plate.

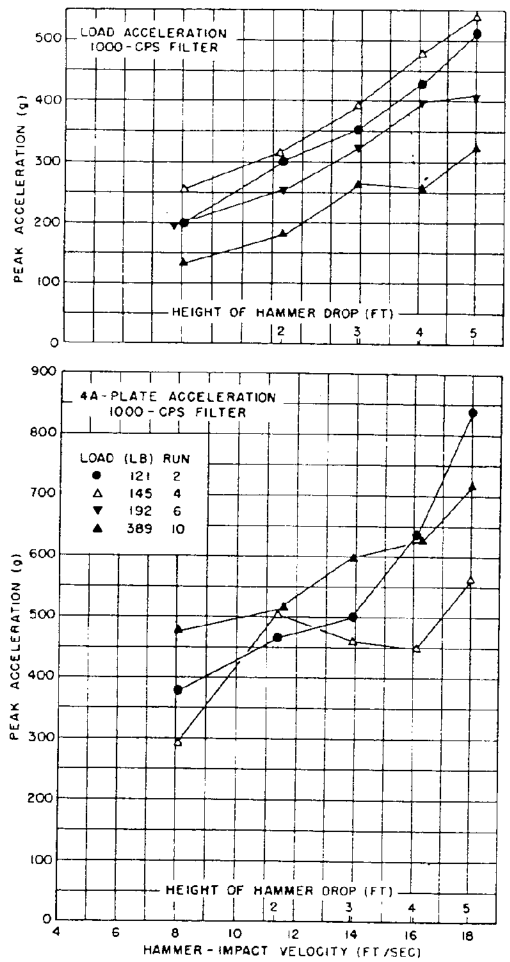


Fig. 11 - Maximum accelerations of load and of 4A plate for back blows

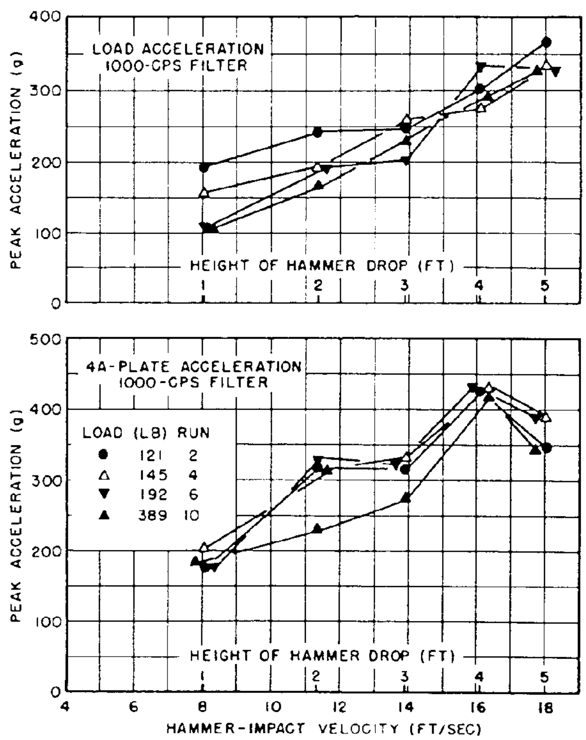


Fig. 12 - Maximum accelerations of load and of 4A plate for edge blows

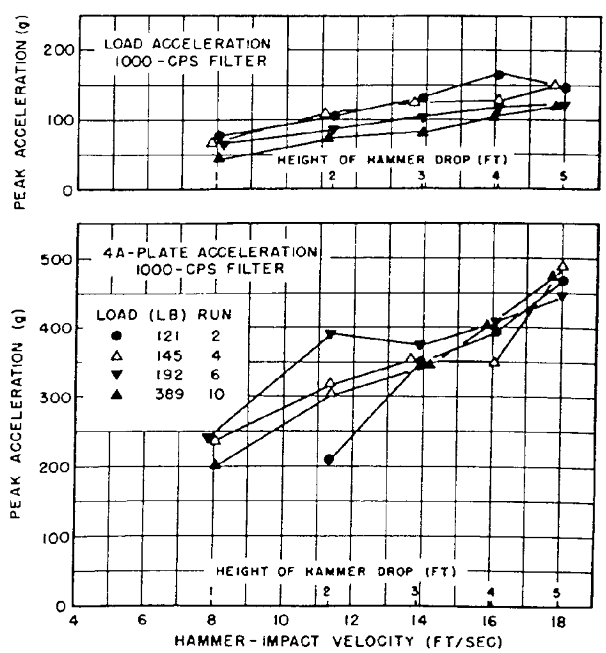


Fig. 13 - Maximum accelerations of load and of 4A plate for top blows

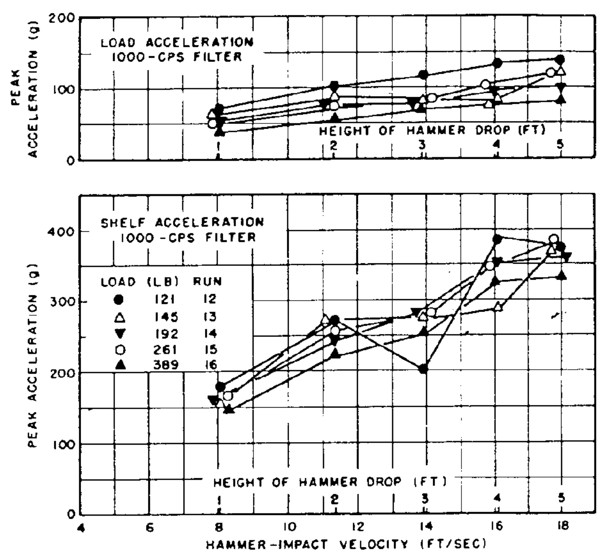


Fig. 14 - Maximum accelerations of load and of shelf plate, for back blows, for load on the shelf plate

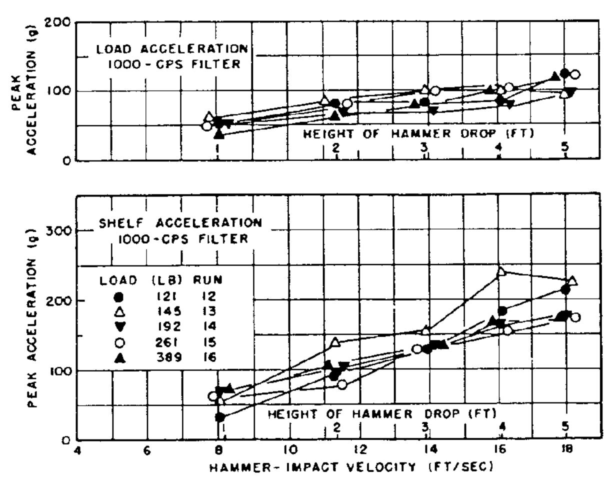


Fig. 15 - Maximum accelerations of load and of shelf plate, for edge blows, for load on the shelf plate

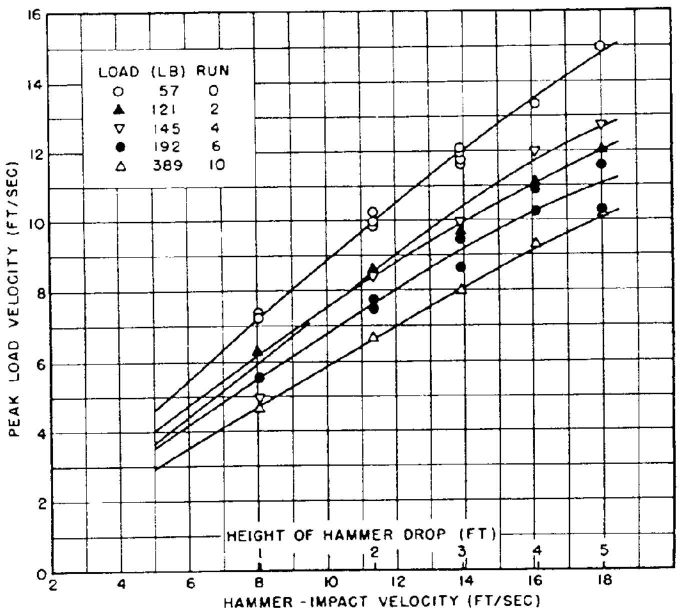


Fig. 16 - Maximum load velocities for back blows for loads on the 4A plate

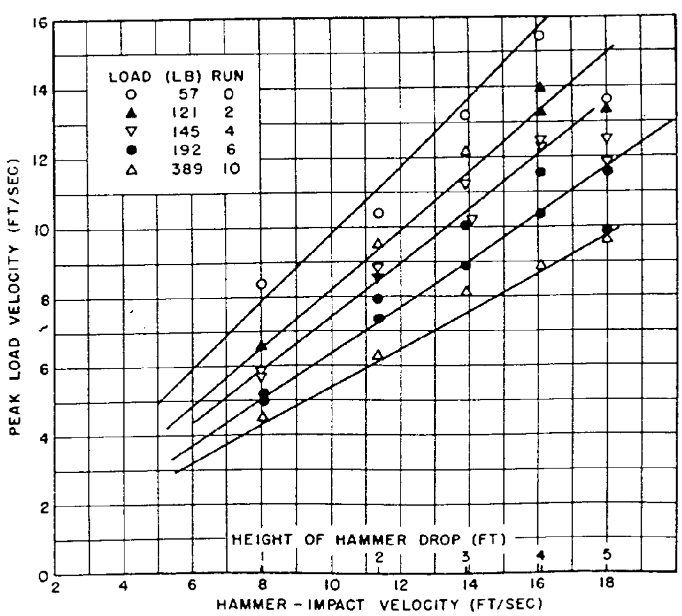


Fig. 17 - Maximum load velocities for edge blows for loads on the 4A plate

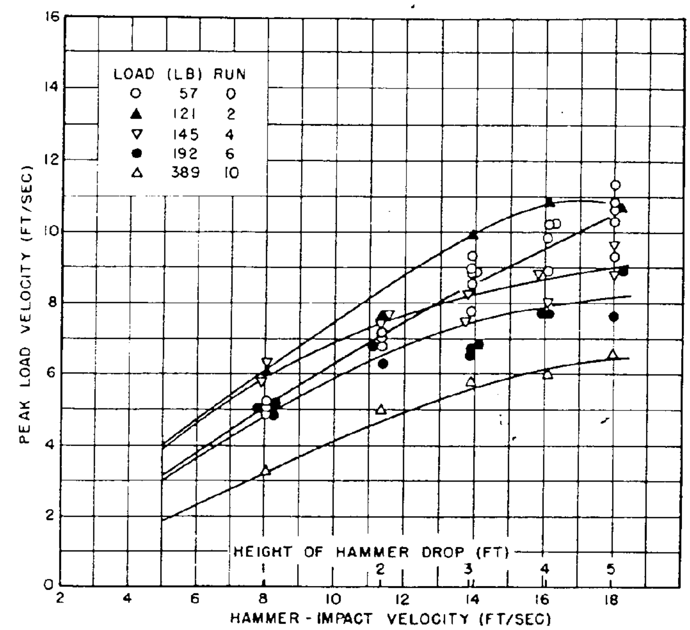


Fig. 18 - Maximum load velocities for top blows for loads on the 4A plate

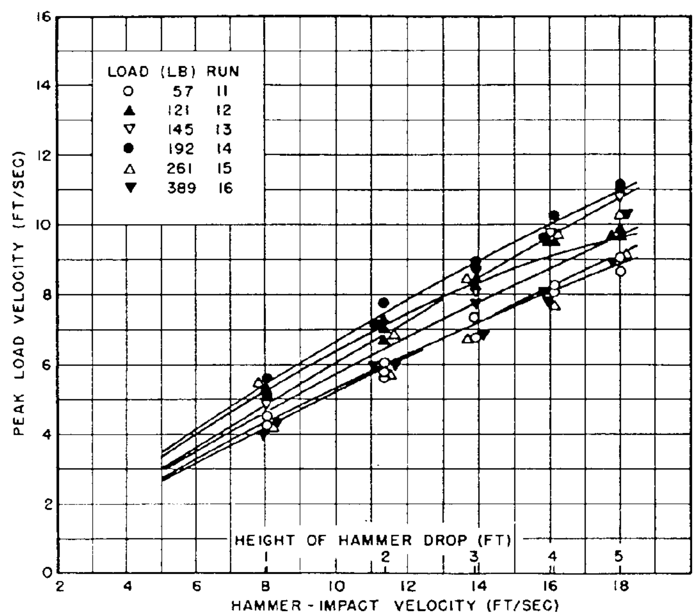


Fig. 19 - Maximum load velocities for back blows for loads on the shelf mounting plate

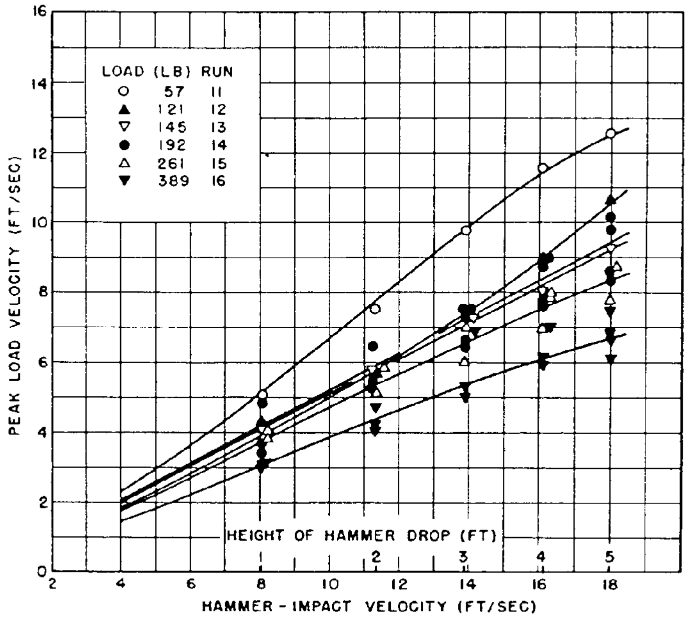


Fig. 20 - Maximum load velocities for edge blows for loads on the shelf mounting plate

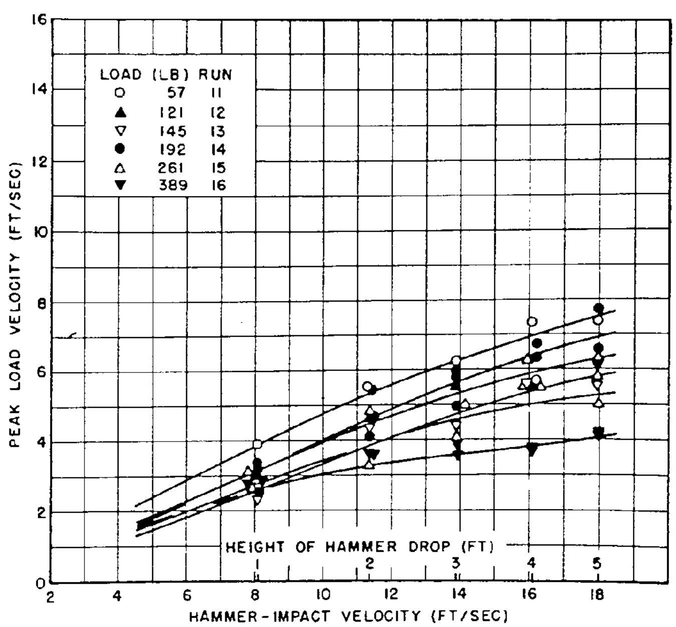


Fig. 21 - Maximum load velocities for top blows for loads on the shelf mounting plate

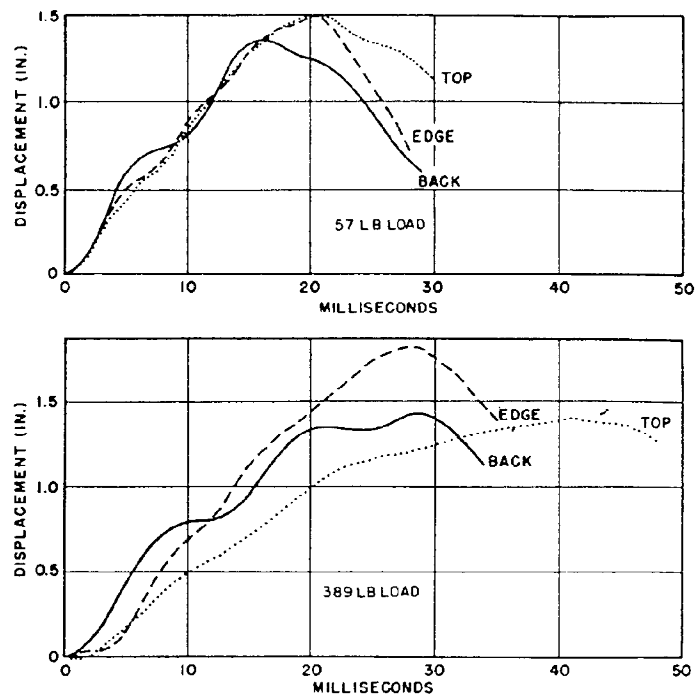


Fig. 22 - Displacement-time motions (from integrated velocity records) of load on the 4A plate for 5-ft back, top, and edge blows

Shock Spectra - Results relating to shock spectra are principally contained in reports by Dick (3) and Conrad (30). Shock spectra can be obtained directly from reed gages. When this method is employed, only a small number of frequency values can be obtained because of the difficulty of using many reeds simultaneously. A more complicated but preferable procedure is to record the appropriate acceleration or velocity signal on tape and to analyze the recording in terms of shock spectra.

Shock spectra, for motions of the load in combination with the 4A plate and the shelf plate are given in Figs. 23 through 25. Figure 26 indicates how the response varies with hammer drop-height, and Fig. 27 illustrates an equivalent velocity-shock or step velocity-change. This would be the average slope (see Fig. 23) of an acceleration shock-spectrum curve. Velocity-shock is useful for design calculations and provides a way of expressing shock intensity in terms of a single number.

Shock spectra for several different shock machines for several load conditions and hammer drop-heights have been plotted in Figs. 28 and 29 using the four-coordinate system. Considerable information is directly available from such a graph. For very low frequencies the displacement shock spectra is asymptotic to the maximum displacement involved in the actual shock motion. For very high frequencies the acceleration shock spectra is asymptotic to the maximum acceleration recorded for the shock. For intermediate frequencies, if peaks caused by resonances are neglected, the velocity-shock spectra can be taken as the best value of velocity-shock. Peaks in the shock spectra represent sustained vibrations in the shock motion.

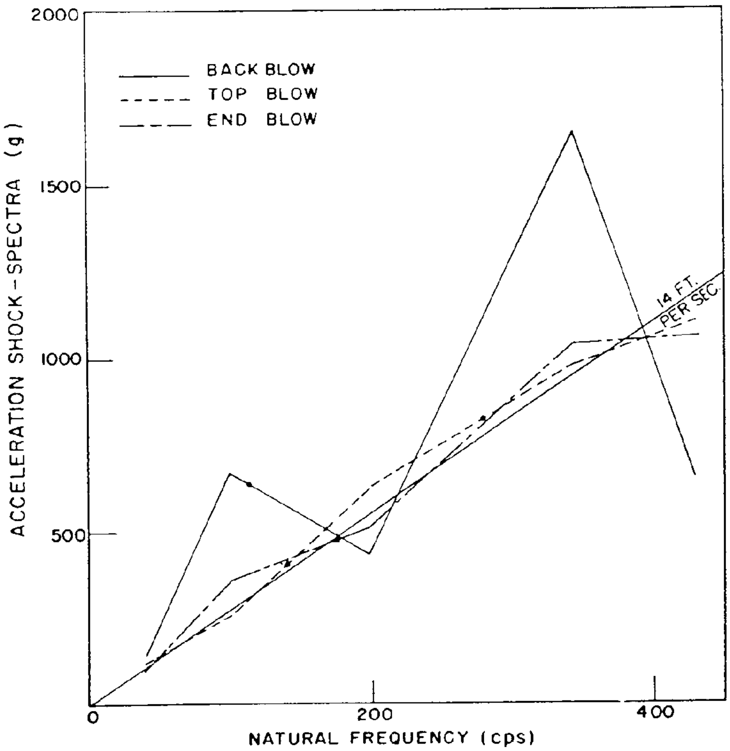


Fig. 23 - Shock spectra for motions of a rigid 57-lb load on the 4A plate for 5-ft back, top, and edge blows

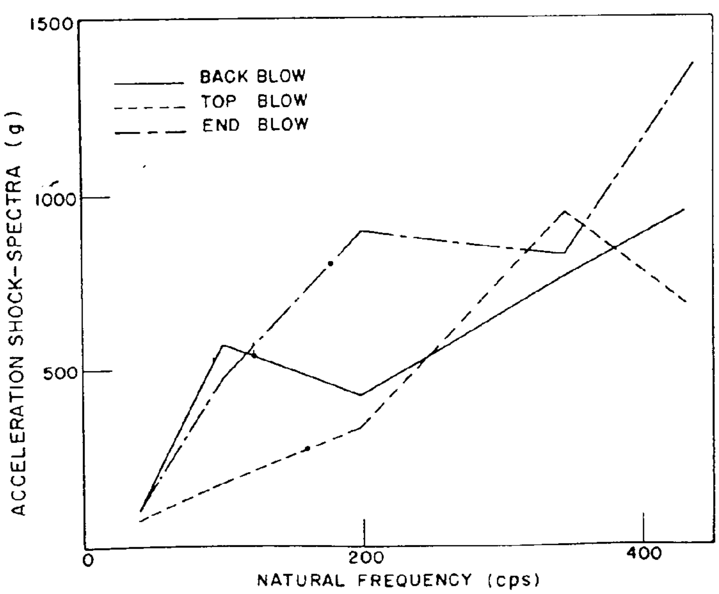


Fig. 24 - Shock spectra for motions of a rigid 121-lb load on the 4A plate for 5-ft back, top, and edge blows

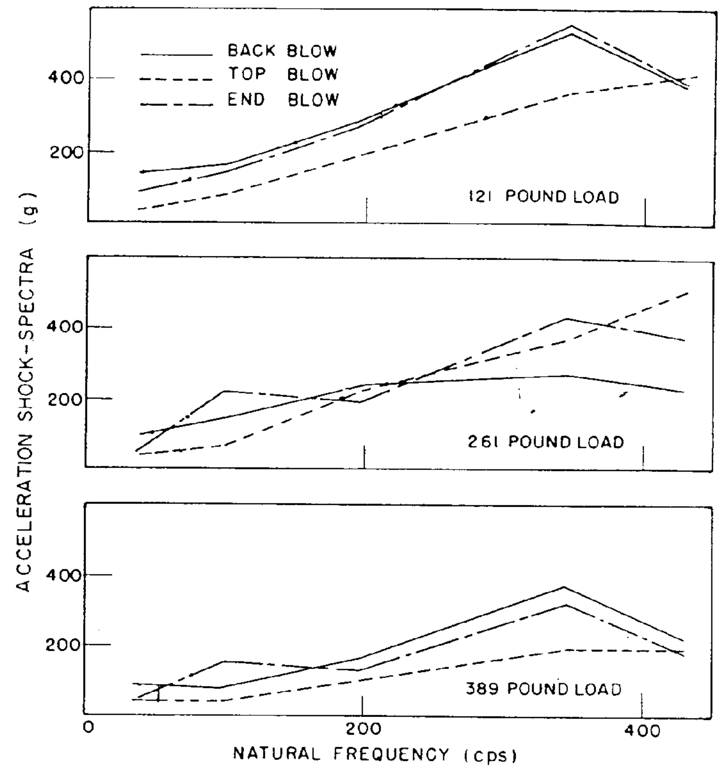


Fig. 25 - Shock spectra for motions of rigid 121-, 261-, and 389-lb loads on the shelf plate for 5-ft back, top, and edge blows

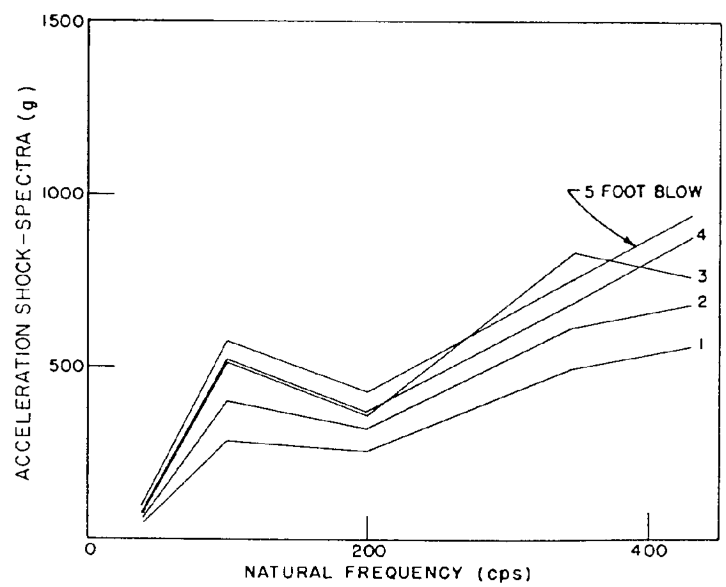


Fig. 26 - Shock spectra for rigid 121-lb loads on the 4A plate for various heights of hammer drop

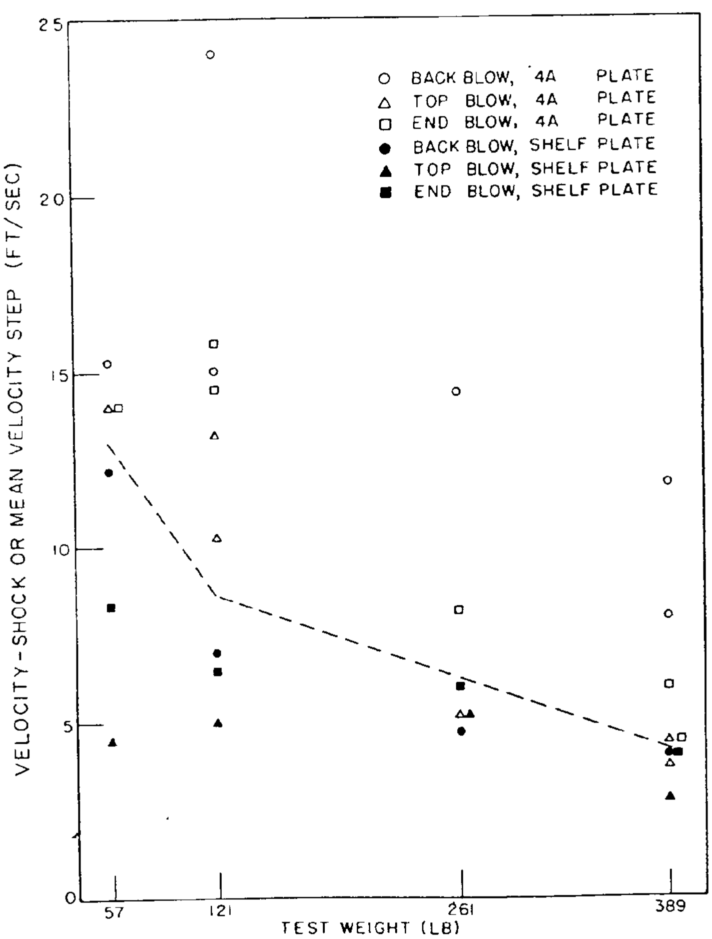


Fig. 27 - Values of velocity-shock for 5-ft blows as a function of load weight

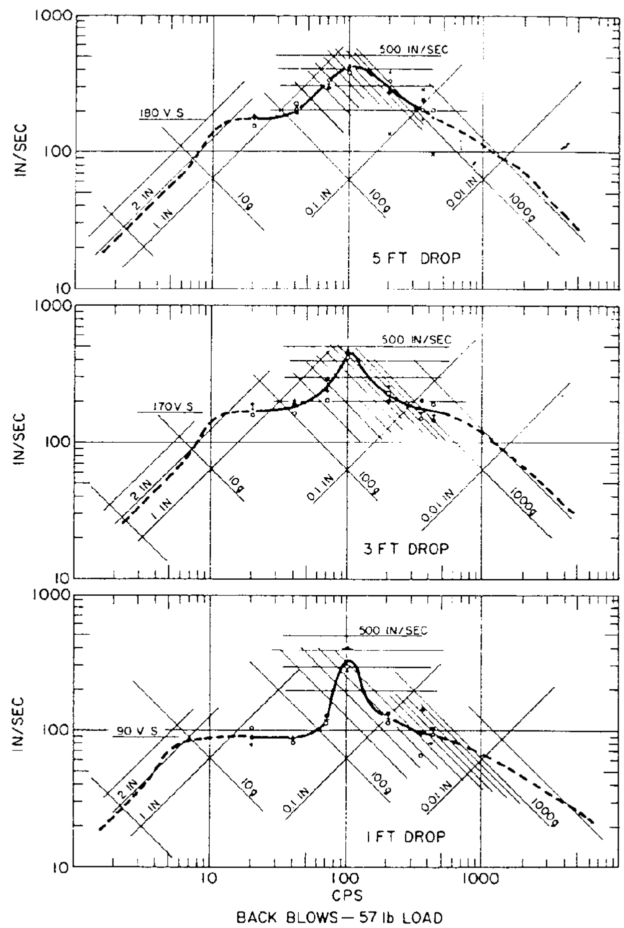


Fig. 28 - Shock spectra for motions of a rigid mass of 57 lb attached to the 4A plate of the HI Shock Machine for Lightweight Equipment. An equivalent velocity-shock (V.S.) for frequencies between 10 and 40 cps is indicated.

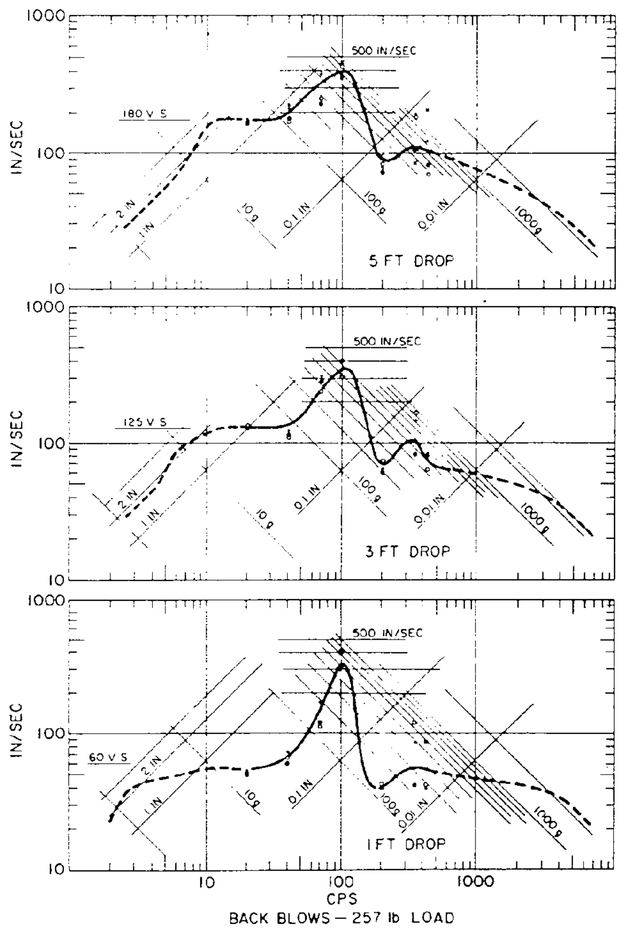


Fig. 29 - Shock spectra for motions of a rigid mass of 257 lb on the 4A plate of the HI Shock Machine for Lightweight Equipment

As a shock spectrum is by definition the maximum relative displacements, or the maximum accelerations, experienced by the masses of single-degree-of-freedom systems subjected to the shock motions, these values can in many cases be taken directly to represent relative displacements across flexible mounts and accelerations of items supported on flexible mounts.

The points plotted in Figs. 28 and 29 represent different machines. They show considerable spread in the high frequency end of the spectrum. These differences are in part caused by changes resulting from cold-working and deformation of the anvil. The shocks tend to become measurably more severe as the hammer-anvil contact area increases and the anvil work-hardens. This continues until cracks form in weld areas or the deformation becomes excessive. When the deformation exceeds a prescribed value, defined in Ref. 4, the anvil is repaired or replaced.

THE NAVY HI SHOCK MACHINE FOR MEDIUMWEIGHT EQUIPMENT

Description

The Hi Shock Machine for Mediumweight Equipment, Fig. 30, consists of a 3000-lb hammer which swings through an angle greater than 180 degrees and strikes an anvil. The anvil thereby suddenly acquires a velocity in the upward direction. The anvil, which weighs about 4000 lb, can be placed in either of two vertical positions. A maximum vertical travel of 3 in. is permitted by hold-down bolts from the lower position, and 1.5 in. from the upper position. Special arrangements may be used to permit other travel distances. The hold-down bolts cause a sudden reversal of the upward motion of the anvil. The machine is mounted on a heavy concrete block which is isolated from surrounding areas by coiled-spring supports. This prevents the shock from being transmitted to the surrounding area.

Equipment under test is not attached directly to the anvil table, but is attached to a set of channels that are separated from the table by spacers. This is illustrated in Fig. 31. The number of channels used is a function of the weight of the equipment and is given in specifications (5) for use of the machine. The anvil, channel system, and equipment, so assembled can approximately be represented by Fig. 32, where M_2 is the anvil, the spring and dashpot are the channels, and M_1 is the equipment. The hammer impact is applied at the bottom center of M_2.

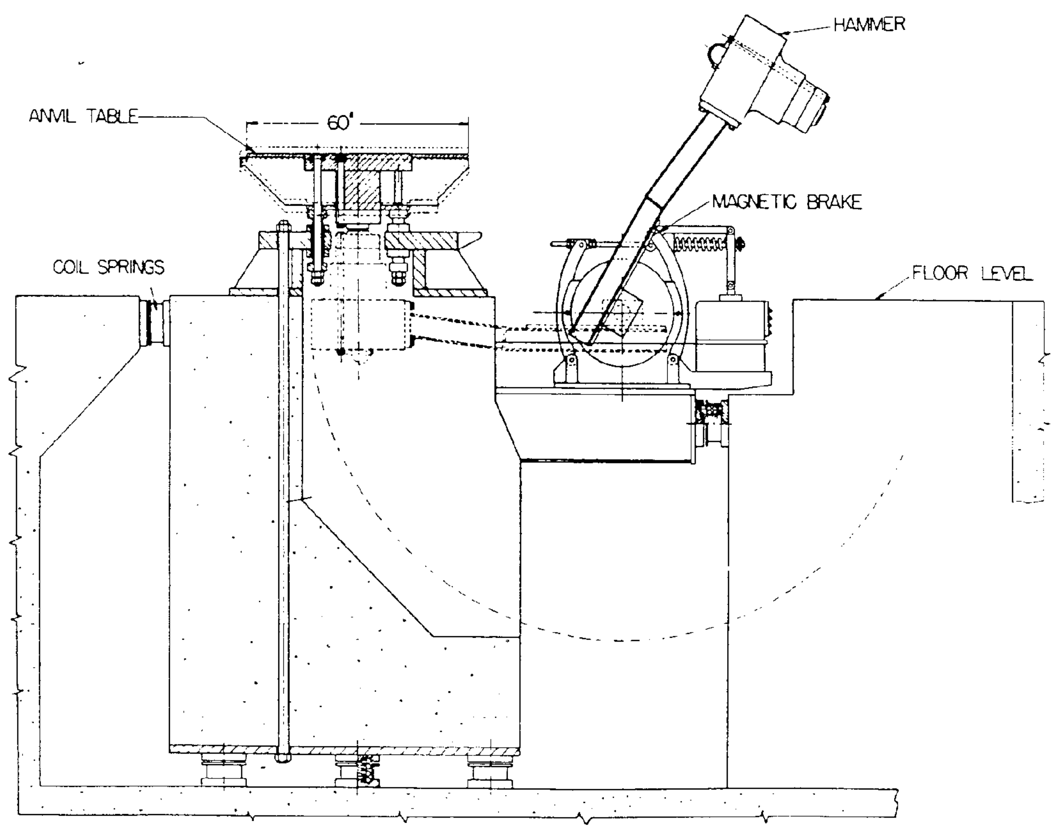


Fig. 30 - The HI Shock Machine for Mediumweight Equipment. The position of the hammer at the instant of impact is shown by the dotted lines.

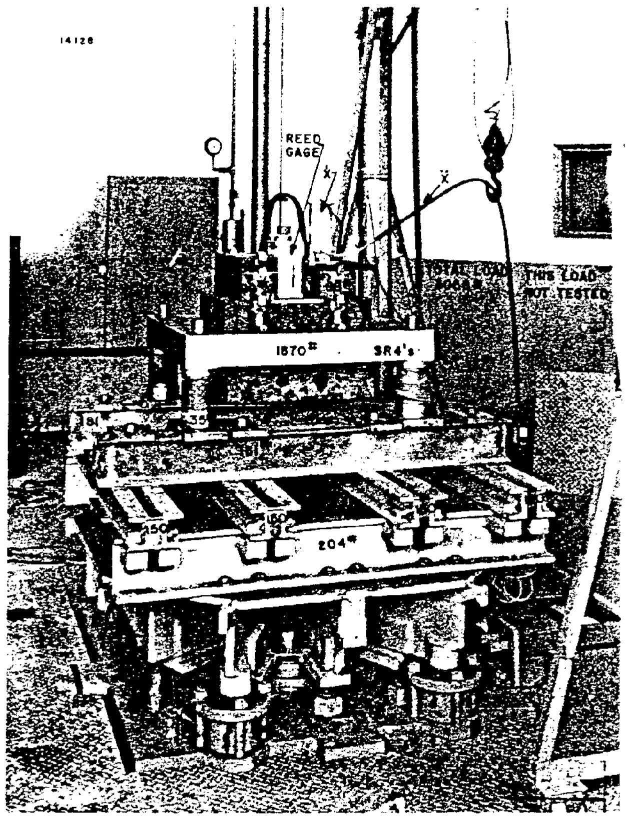
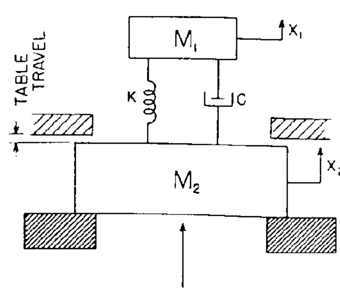


Fig. 31 - HI Shock Machine for Mediumweight Equipment with a 3058-lb load mounted on channels

Fig. 32 - Schematic representation of the HI Shock Machine for Mediumweight Equipment. M_1 represents the load, M_2 the anvil table, and K the channels.



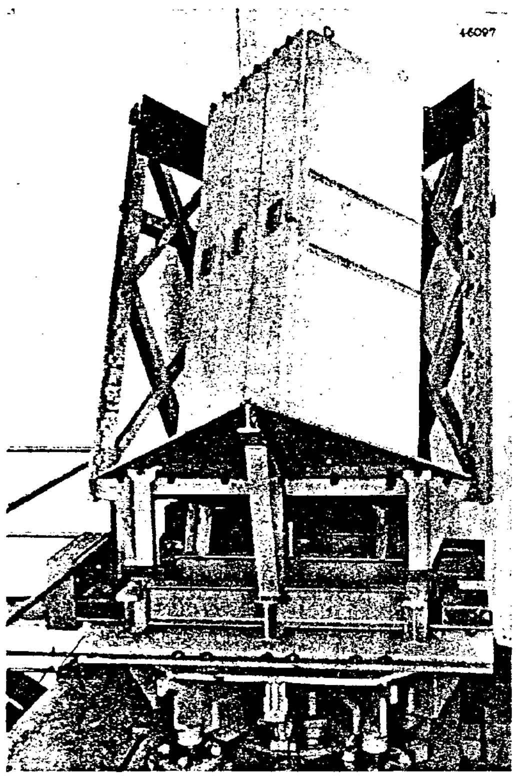


Fig. 33 - Thirty-degree inclined bulkhead fixture. An item of equipment is mounted in the fixture and is ready for test.

As the shock motion is only in a vertical direction, an alternative mounting arrangement is sometimes used. Two bulkheads and a deck section are assembled together, as shown in Fig. 33, to form a corner with three mutually perpendicular surfaces. The junction of the two bulkheads is made to incline 30 degrees from the vertical when the fixture is mounted on the shock machine. The fixture thus provides shock motions along all principal axes of an item of equipment under test and is convenient for items that require bulkhead supports.

Measurements

Reed gages and velocity and acceleration pickups were mounted on the loads and the anvil table to provide information for the determination of shock spectra and shock motions. The accelerometers signals were filtered by 300-, 1000-, or 5000-cps low-pass filters. In addition a set of bonded-wire resistance strain gages were cemented to one of the load-supporting feet. This provided information for determining the force exerted on the load by the channels.

A typical set of records for 4420-lb load and a 2.75-ft hammer drop is shown in Fig. 34. A 1000-cps low-pass filter was used for the acceleration records. It can be observed that the channels provide considerable isolation for the high-frequency components, but introduce a strong dominant frequency of about 70 cps. This corresponds to the natural frequency of the single-degree-of-freedom system shown in Fig. 32. The isolated acceleration pulse on the top curve is caused by the sudden arresting of the upward motion of the table, by hold-down bolts, after a table travel of 1.5 in. has been completed.

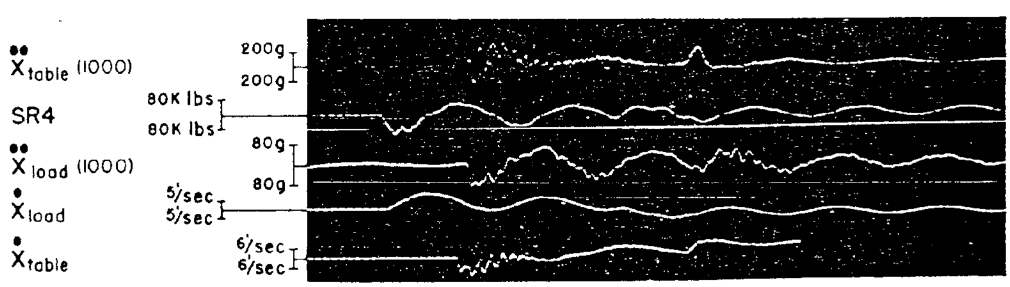


Fig. 34 - Recordings of typical shock motions. Blanking markers are spaced at 1 millisecond intervals. Top trace: acceleration of table, 1000-cps low-pass filter; second trace: force exerted on load; third trace: acceleration of load, 1000-cps low-pass filter; fourth and fifth traces: velocity of load and of table.

As a result of the reversal of velocity of the anvil table, when it reaches the limit of its upward motion, the load experiences an additional sudden shock. The damage potential of this velocity-reversal shock depends upon the relative positions and velocity of the load with respect to the anvil at the time the reversal occurs. If the anvil (see Fig. 32) is suddenly stopped when the spring is at its greatest extension, then the damage potential is the largest. That this is a matter of significance is realized when it is observed that, depending upon the load vibration phase angle at the time of impact, the velocity change caused by the stopping impact may vary from about 0.5 to 2.0 times the velocity change caused by the original impact. Various table travel distances and travel times involved in a test make it probable that under some condition an item of equipment will be exposed to a phase angle of most severe damage potential.

The relationship between hammer impact velocity (and height of hammer drop) and the maximum velocity and acceleration of the anvil are shown in Figs. 35 and 36. These are linear relations. This is to be expected when it is observed that the hammer-anvil impact time is a constant, about 0.001 sec, and is independent of the drop height and of the load. The initial maximum velocity attained by the anvil is little affected by the load, as it is attained suddenly compared with the period of the load on its channels. However, as shown by Fig. 37, the maximum velocity experienced by the load is affected, as one would expect, by the magnitude of the load.

The difference in magnitude of the acceleration values passed by the 300- and 1000-cps low-pass filters, as shown in Fig. 36, indicated a considerable amount of relatively high-frequency motion present in the anvil table. This is also shown in Figs. 38 through 41; however, the latter group also shows that the maximum accelerations of the load are about the same value whether or not they pass through the 1000- or the 300-cps low-pass filter. Actually, as can be seen by the record of the actual motion (Fig. 34), the maximum values of load acceleration and velocity are associated predominately with the 70-cps fundamental frequency. Generally the channels provide isolation from the high frequencies when the load is relatively rigid.

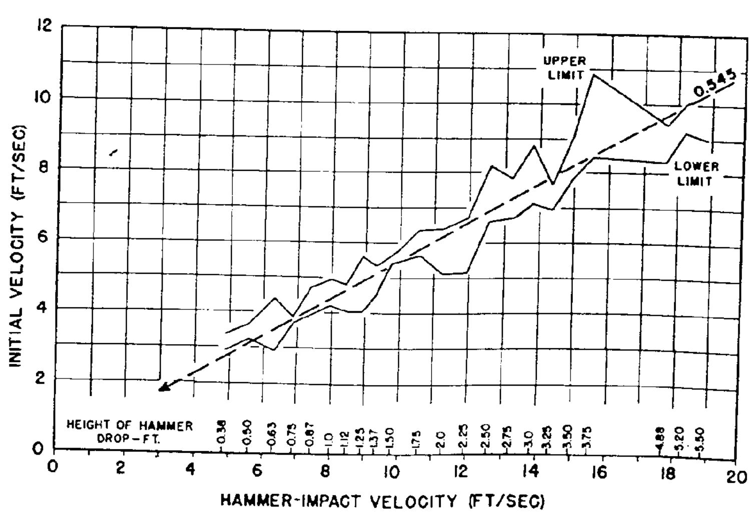


Fig. 35 - Impulsively attained velocity of the anvil table. The values are independent of the channel-mounted loads.

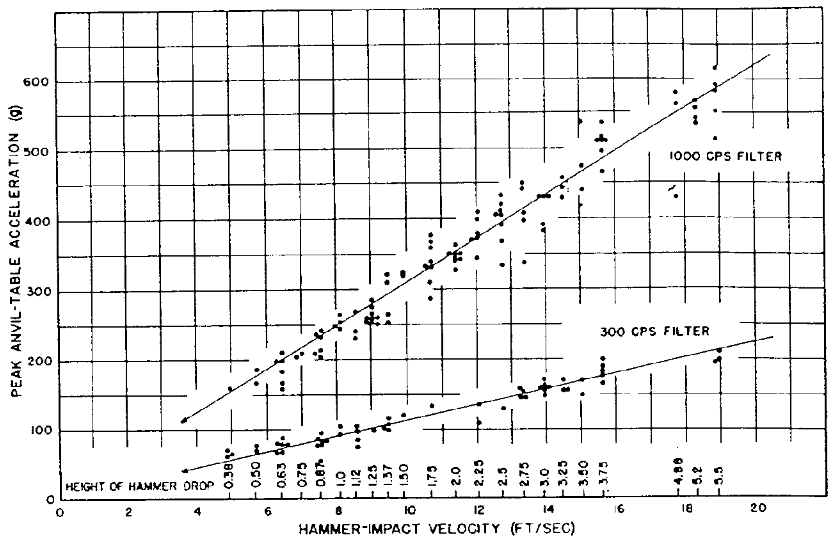


Fig. 36 - Maximum anvil-table accelerations for 1000- and 300-cps low-pass filtration. The values are independent of the channel-mounted loads.

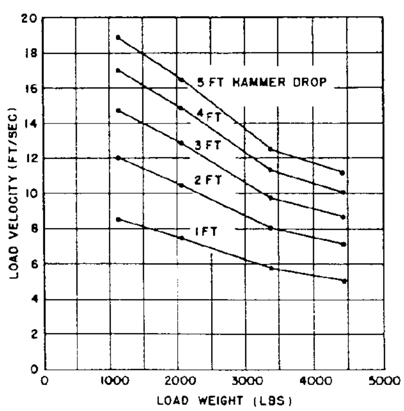


Fig. 37 - Maximum load velocity as a function of load weight for different hammer drops

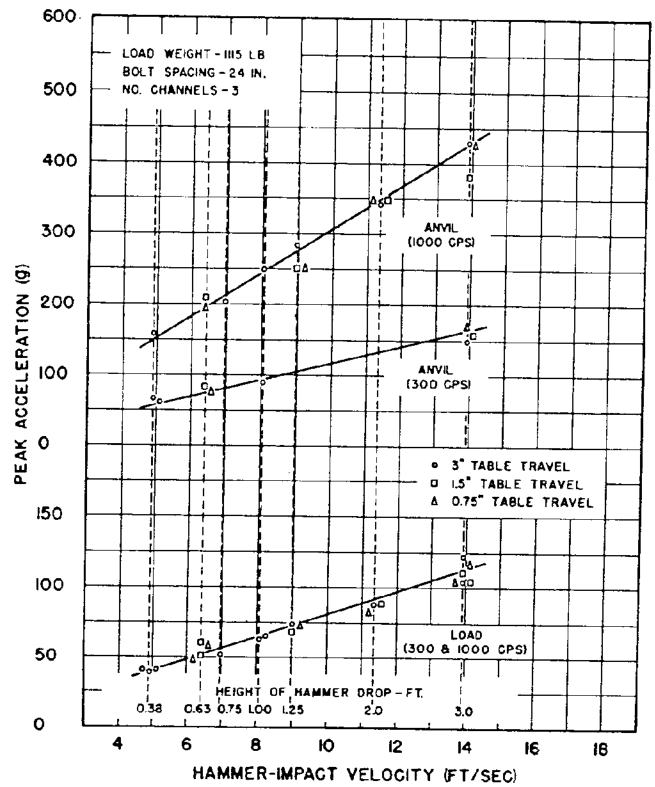


Fig. 38 - Maximum acceleration of "rigid" load and of the anvil table. Acceleration signals were filtered by 300- or 1000-cps low-pass filters as indicated.

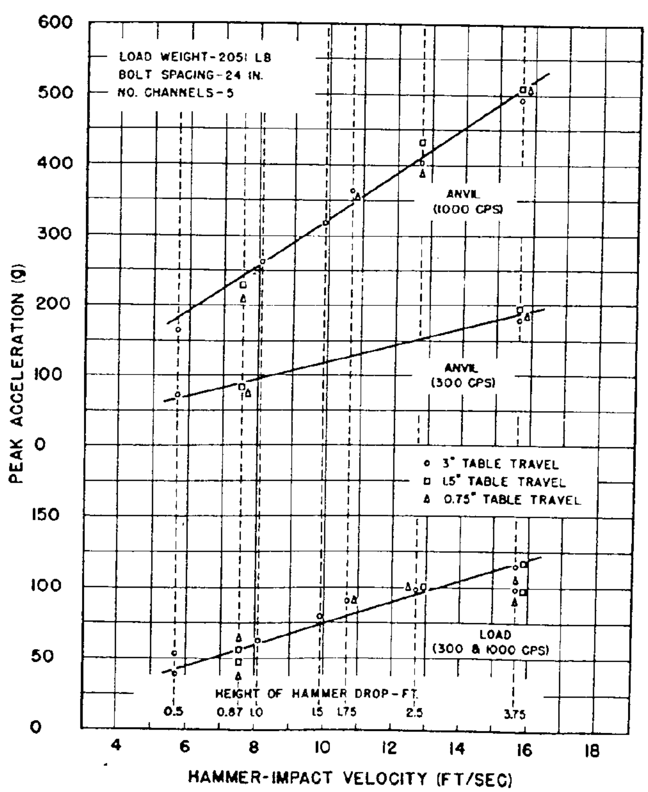


Fig. 39 - Maximum filtered acceleration values on a rigid load and on the anvil table

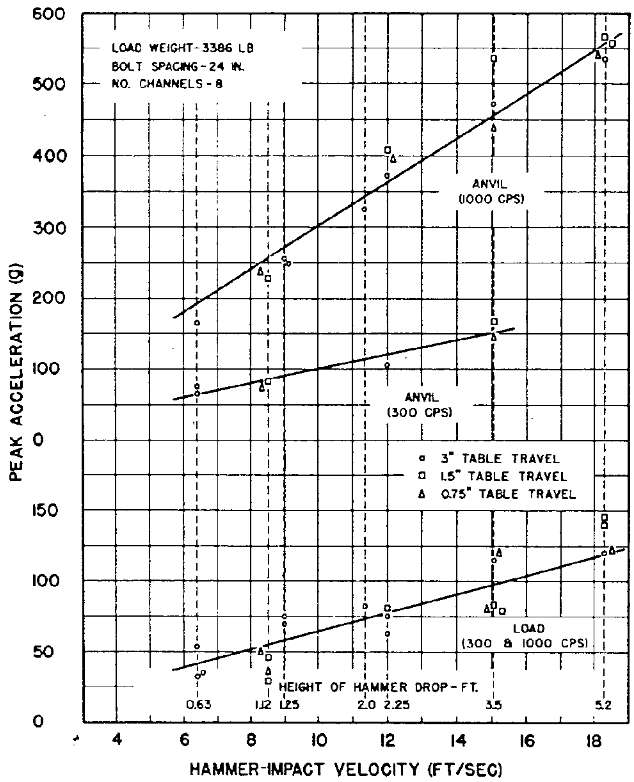


Fig. 40 - Maximum filtered acceleration values on a rigid load and on the anvil table

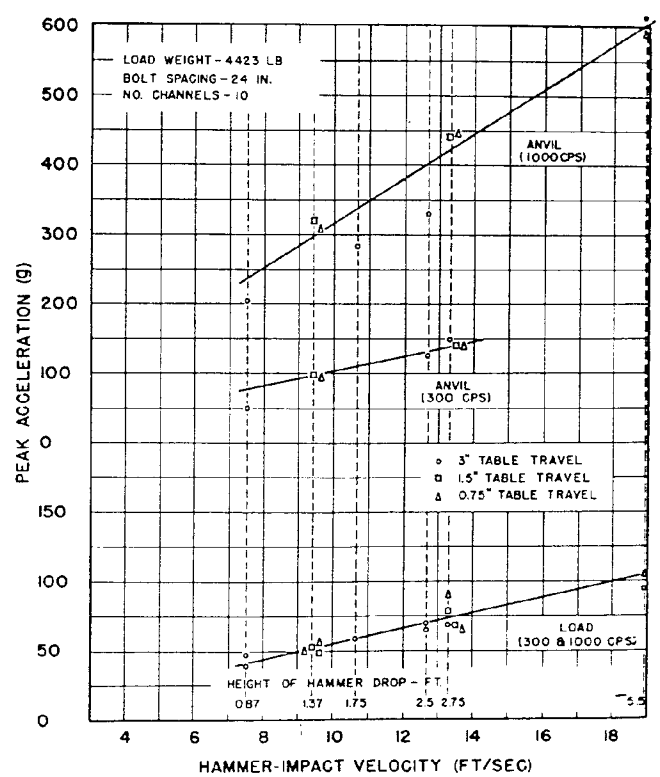


Fig. 41 - Maximum filtered acceleration values on a rigid load and on the anvil table

Typical values of shock spectra are shown in Figs. 42 and 43. These have been extrapolated by dotted lines to show probable values beyond the range of measurements. The values at low frequencies become asymptotic to the permissible table travel, which is usually 3 or 1.5 in., although 0.75 in. may additionally be used. The spectra of the motions of the load* provide an equivalent velocity-shock, which is applicable from a few cycles per second to about 40 cps. The spectra then rises to a maximum value due to the load-channel resonance at about 65 cps. Above 100 cps the load acceleration spectra remains about constant. If the load were a more flexible system higher values of acceleration would be experienced at the high frequencies.

The relatively simple spectrum, Fig. 43, is illustrative of that of the anvil table. The motion of the anvil table can be considered a velocity-shock for frequencies between 10 and 1000 cps. Below these frequencies the displacement spectra becomes asymptotic to the table-travel setting, and above these frequencies acceleration spectra becomes asymptotic to the true maximum value of acceleration, which is in the region between 5000 and 10,000 g.

CONCLUDING DISCUSSION

The Navy HI shock machines attempt to provide a simulation of types of shocks probable on board ships. The intensity of shock is of such a value as would occur when the ships structure is damaged but the ship is still seaworthy. Shock tests are specified in terms of the shock machine and procedures for its use, rather than in terms of shock motions or spectra.

Shock motions and spectra typical for various loading conditions of the shock machines are included in this report. A determination of these values under standard conditions describes their performance, and is sometimes called the calibration of the machine.

The shock motions expressed as a time function are not of themselves useful without further analyses. Perhaps the most useful, and meaningful, analysis method is in terms of shock spectra. Simpler, but less informative, statements of shock intensity are given in terms of velocity-shock, and in terms of maximum values of accelerations transmitted through filters of given bandwidth. All methods of analyses assume something of the nature of the item being subjected to the shock inasmuch as damage potentials of a shock motion are as much a function of the characteristics of the item being shocked as they are of the nature of the shock motions.

The use of four-coordinate log paper for the presentation of shock-spectral curves permits the natural extension of the concept of velocity-shock to displacement-shock and permits a determination of the maximum values of acceleration. The shock-spectral curves illustrate that below a given frequency the displacements approach a constant value. This value, together with the frequency range for which it is sufficiently accurate, is defined as the displacement-shock value and range. A middle frequency-range usually exists for which the velocity-shock spectrum is relatively constant. This velocity is defined as the equivalent velocity-shock for this range. And similarly an upper range of frequencies exist for which the acceleration-shock spectrum is relatively constant. This acceleration value is the maximum value present in the excitation. The maximum value of acceleration can be considered as a static value in this upper frequency range. Maximum values of changes of displacement and velocity and maximum values of acceleration used in this manner provide simple, significant, and meaningful descriptions of a shock motion but of course do not include all the information contained in the motion-time or shock-spectral curves.

*Since the load is rigid, its spectra and its motions are assumed to be the same as that of its mounting points on the channels of the shock machine.

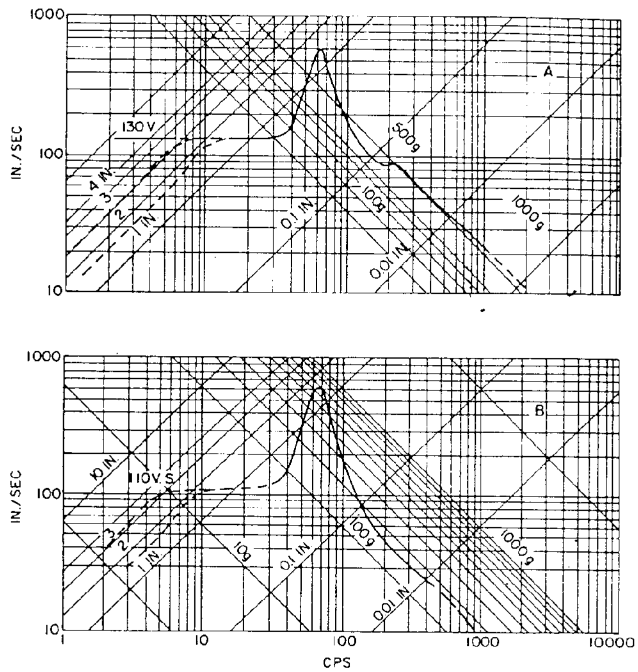


Fig. 42 - Shock spectra for motions of rigid load mounted according to prescribed specifications (5). A. 5.5-ft hammer drop, 3-in. table travel, 4423-lb load, 5390 lb on table, class A shock. B. 2-ft hammer drop, 3-in. table travel, 1115-lb load, 1783 lb on table, class A shock.

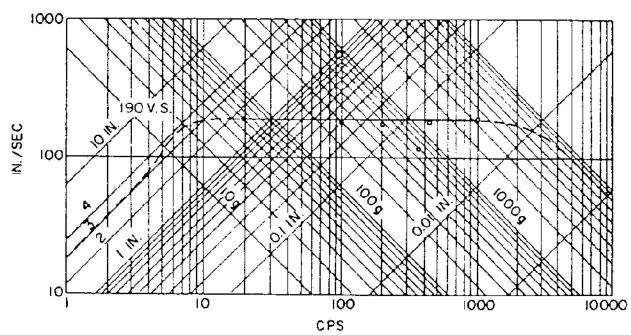


Fig. 43 - Shock spectrum for motion of anvil table of HI Shock Machine for Mediumweight Equipment; 3-ft hammer drop, 3-in. table travel, 1115-lb load, 1858 lb on table, drop-height 150 percent greater than specified (5) for class A shock

The HI shock machines provide good duplication of shock motions for successive impacts of a given test. However if the equipment is dismounted and then remounted in presumably an identical manner, differences in the nature of the high-frequency components are observable. These are caused by differences in damping and elasticity under the two conditions which is in turn caused by different seating of parts held together by bolts, and by differences in bolt tightness. In general, frequency components below several hundred cycles per second are not affected. The changes of physical properties of the anvil plate of the machine for lightweight equipment, as it deforms with use, are such as to gradually increase the severity of the shock motions.

It will occasionally happen that an item of equipment will consistently pass specification on one HI machine but fail an equivalent test on another similar machine. Studies of these instances have shown that there was little margin of safety in the first case, and that the critical value for failure was barely exceeded in the second. Obviously an item must pass the test on the machine used during acceptance tests.

The maxima in the shock-spectral curves which are caused by resonance vibrations of the mounting plates, or channels, of the shock machines are of some concern. It may be that the presence of narrow frequency regions, in which the damage potential is large, is not in accord with the idea of presenting a shock motion equivalent to a generalized field condition. It is probable that a shock motion without dominant frequencies would be preferable.

ACKNOWLEDGMENTS

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