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AUTHOR: H. A. Gaberson, Ph D and R. A. Eubanks, Ph D

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PORT HUENEME, CALIFORNIA 93043

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A simplified shock design method has been developed and is set forth. The method employs the shock spectrum as the fundamental measure of both the "severity" of a shock environment and the "hardness" of a particular piece of equipment (the "hardness" is the equipment's capacity to survive a shock environment). Both the severity and the hardness are given as shock spectrum plots on "four-coordinate paper" and are thus directly comparable. The method uses a tracing method of recording the fact that the hardness exceeds

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## FOREWORD

Dr. Gaberson, of the Naval Civil Engineering Laboratory, was the principal investigator for this study. Additional help and expertise were required both to review a preliminary draft of this work and to prepare a new section on the simplified application of shock isolators. The services of Dr. Robert A. Eubanks, Professor of Civil Engineering and of Theoretical and Applied Mechanics, University of Illinois, Urbana, were obtained to fulfill this requirement. He, along with B. R. Juskie (Ref 2), originally invented the method that was modernized for use here, and, thus, was singularly the most qualified expert to assist the Laboratory in this endeavor.

CONTENTS

## INTRODUCTION

The Naval shore establishment has many facilities that must be designed to not only survive explosive or earthquake loadings, but continue to perform their functions properly. This design effort is usually referred to as Shock Design - the subject of this report.

Several years ago the Naval Civil Engineering Laboratory (NCEL) undertook, for the Naval Facilities Engineering Command (NAVFAC), the study of routine equipment hardness; the study stemmed from a gaping lack of the so-called "fragility" data referenced in all nuclear design manuals. The idea assumed a simple measure of shock resistance, such as g's, and a simple design procedure once the measure was known. It was hoped that the results of all previous tests could be collected and made useful. After the work was initiated, no theoretical justification for "g's" as a measure of fragility could be found, nor did the data available support any concept. Other investigations came to similar conclusions (Ref 1-4).

A measurement or data analysis result that does have both theoretical and practical relation to the fragility of equipment is the shock spectrum. This is an old concept presentable in many forms; it can be thought of as a relationship between relative deformations in the equipment and the frequencies at which they occur. Specifically, the particular form selected is the maximum overall spectrum plotted on four-coordinate paper with pseudo-velocity as the ordinate (other forms of "shock spectra" exist in the literature). This form has a direct relationship to equipment hardness and is common in works on earthquake and blast-resistant engineering, some of which are cited in the references.

This manual presents a brief introduction of the concepts behind the shock spectrum and initiates a "Hardness Plan," which includes a simplified design procedure for equipment subjected to shock loadings, and a catalog of equipment hardnesses. The "Hardness Plan" involves cataloging common potential dynamic environments by using shock spectra, and similarly cataloging shock hardnesses of equipment, including the resulting hardnesses that can be presumed on the basis of prior history of shock or shock test survivability. The design procedure is used with the above to (a) prove the equipment can be installed directly, (b) prove the equipment could be installed if isolators were used, or (c) conclude that this simple design procedure is inadequate to assure a safe installation.

## BACKGROUND

A shock environment, in general, refers to that category of dynamic environment which is of high intensity and short duration such that the equipment is set into a vibratory motion and continues to respond after the disturbance has passed. It begins and ends, as opposed to a vibration environment that continues for some long period of time. The shock can come from a blow or short-duration force, such as an air blast, or it can come from the foundation of the equipment being set into a momentary violent motion. The latter method, the base-excited shock motion, is specifically the subject of this report.

Briefly, the base-excited shock problem includes motions resulting from chemical and nuclear explosions, earthquakes, drops and falls, vehicle collisions, wind gusts, and transportation environments (railroad car bumping, roadway irregularities on surface vehicles, etc.). Thus, the payoff for an improvement in designer capability is widespread.

The organization of this design procedure focuses attention on any prior shock testing to which planned equipment may have been subjected. It should make the designer lean toward equipment with a known history of surviving shock tests, since, in most cases, he can be relieved of the responsibility for the equipment survival by documenting his use of the design procedure. Once successful shock test history becomes an essential or attractive qualification of candidate equipment, manufacturers will begin to upgrade their products so as to qualify them for use in a broader range of potential dynamic environments. In essence, the practice of these design concepts will work to correct that which has heretofore been an area of virtual nondesign due to a lack of an economical design procedure.

This procedure, which focuses on equipment survivability, differs from that of References 5 and 6, the seismic design procedures, in that they consider "breakaway" forces for heavy equipment in earthquake environments. Most earthquake environments contain insufficient higher frequency energy to damage typical equipment internal members; thus, internal equipment damage is seldom a concern. Conversely, the explosive shock environment has a large portion of its energy in this higher frequency range. To account for the potential for internal equipment damage, this procedure introduces a more empirical approach, depending heavily on the shock spectrum as the measure of severe equipment motion. Thus, in no way does this procedure conflict with, or modify, References 5 and 6. It treats the environment in a more specific manner, and considers internal equipment dynamic phenomena as they pertain to continued equipment function during and after the shock environment.

Any meaningful design procedure evolves through mutual exchange of information. The Naval Civil Engineering Laboratory will serve as a clearinghouse for information on this design procedure. Designers are invited to communicate with the principal investigator:

Dr. H. A. Gaberson
Mechanical Systems Division
Naval Civil Engineering Laboratory
Port Hueneme, California 93043

Telephone: autovon - 360-5975
commercial - (805) 982-5975

## FUNDAMENTAL DESIGN ASSUMPTIONS

It is assumed that the equipment (and its associated shock isolator, if any) is attached to a rigid fixture such that the equipment attachment points do not move with respect to each other; it is assumed that this fixture undergoes the shock motion. Initially, it is assumed that the equipment to be installed is light with respect to the foundation and, thus, does not affect the foundation motion. In general, the motion of the rigid base can be described by simultaneous translations along three mutually perpendicular axes together with rotations about these same axes. Input motion of this generality is too complex to be considered here; it is fortunate that the lateral dimensions of most equipment are small enough and the magnitude of rotational motions experienced are small enough that the effect of pitch, roll, and yaw can be neglected. Also, experience has shown that, in most cases, the coupling of the response of the equipment to these three mutually perpendicular excitations is secondary. Good results can be obtained by assuming that each input translation is independent of the other two. In particular, the design procedure to be described is based on the fundamental assumption that the equipment will survive the complete input shock motion if it will survive each of the translations independently.

The acceleration time history of the rigid base for the equipment is a valid measure of the shock input in a particular direction. Unfortunately, as was mentioned in the Introduction, this acceleration input can be misleading as a design criterion. In particular, a design procedure that assumes the equipment has been subjected to a force equal to the product of its mass and the peak acceleration of the shock time history and then attempts to design the equipment to resist a force of this magnitude will usually be fruitless. Many studies have shown that a much more valid procedure is to look at the rate at which the various components of the equipment will be deformed. This concept is the basis for the shock spectrum approach, which will be introduced in the following section.

## UNDAMPED SHOCK SPECTRUM

The capacity of a shock motion to cause damage at any particular frequency, $f_1$, is related, monotonically, to the peak response that the shock motion is able to cause in a simple damped or undamped vibratory system with natural frequency, $f_1$. Extensive evidence supports this contention. Thus, an earthquake shock motion is unable to cause significant response in high frequency mass-spring systems, while it is devastatingly capable of imparting strong motions to mass-spring systems with lower frequencies in the 1/10 to 2 Hertz (cycles/sec) range. Conversely, a slight hammer tap to the edge of a 6-inch-square, 1/2-inch-thick aluminum plate with a china cup bonded to its center will fracture the cup due to the high frequencies so generated. Thus, the frequency content of a shock motion can be spoken of in terms of its capacity to cause response in simple mass-spring systems of various frequencies (e.g., a motion, with great content at 100 Hertz, can cause the mass to attain a velocity of 500 in./sec in a 5% damped 100-Hertz oscillator).

The undamped shock spectrum is a function that presents frequency content of a shock motion. It is a plot of the maximum positive or negative response of an undamped simple vibratory system versus the frequency of the system. It is computed from an acceleration-time history. Reference 7 gives a thorough account of these calculations and a computer code for carrying them out.

It is convenient to visualize the equipment to be composed of a number of lumped masses, each of which is attached to the rigid base by a spring (see Figure 1). This visualization has a firm theoretical foundation which is discussed briefly in Appendix A. The modifications that this concept undergoes when damping of the system is considered shall be discussed later.

Consider a single mass-spring element with mass, m, and denote the position of the mass by x, while the position of the rigid base is y, and the connecting spring has stiffness, k. If dots over the variables indicate differentiation with respect to time, the motion of this element will be governed by the differential equation

$$m \ddot{x} + k(x - y) = 0 \qquad (1)$$

Spring deformation is important and will be denoted by z; thus,

$$z = x - y \qquad (2)$$

Hence,

$$m \ddot{x} = -k z \qquad (3)$$

which is a simple statement that the Newtonian acceleration force is resisted by the force generated in the spring.

If the rigid base did not move, but the mass spring element was excited by some other means, the mass would still move and would be governed by the differential equation

$$m \ddot{x} = -k x \qquad (4)$$

A solution of this differential equation, which assumes that the spring was not compressed at time zero, is

$$x = x_o \sin \omega t \qquad (5)$$

where $x$ is the maximum displacement attained over time, and the parameter $\omega$ is given by the relationship

$$\omega^2 = \frac{k}{m} \qquad (6)$$

The sine function will begin to repeat itself after a time

$$T = 2 \pi / \omega \qquad (7)$$

where T is the period of the oscillatory motion, or the time required to complete one cycle. The reciprocal of T is f, where

$$f = \frac{1}{T} = \frac{\omega}{2\,\pi} \tag{8}$$

Hence, f is the number of cycles completed in one second, or the number of Hertz for this frequency. The quantity

$$\omega = 2\,\pi\,f \tag{8a}$$

is commonly called the circular frequency for the element.

Returning to the mass-spring element, which is excited by a base motion, the governing equation can be rewritten in the form

$$\ddot{x} = -\omega^2\,z \tag{9}$$

Hence, the basic characteristic of this simple mechanical oscillator is its frequency, and this frequency directly connects the deformation of the spring with the acceleration of the mass. This is a simple, but very important, relationship that will later permit accelerations and deformations to be recorded on the same graph.

Equation 9 also illustrates another point that was referred to earlier: a mass-spring element can experience extremely small deformations but still show very high accelerations because the frequency of the element is high. Since these deformations strongly affect the failure or the survival of the equipment, they represent a good criterion for equipment design. If Equation 2 is substituted into Equation 9, the spring deformation will satisfy the differential equation

$$\dot{z} + \omega^2\,z = -\ddot{y} \tag{10}$$

This shows that the relative deformation is a vibratory response that is forced by the time-dependent acceleration of the base. A particular mass-spring combination will thus experience a relative deformation that depends upon the base motion and the natural frequency of the element. The maximum absolute value of the relative deformation could be taken as a measure of the potential damage for that particular equipment element at its natural frequency. It is actually convenient and conventional to modify this approach by considering the quantity V, where

$$V = \omega |z|_{max} \tag{11}$$

is the primary measurement quantity. This quantity is called the pseudo-velocity as the result of an analogy with the free oscillation solution of Equation 5, where the velocity is

$$\dot{x} = \omega x_o \cos \omega t \tag{12}$$

so that $x_o$ times $\omega$ is the peak velocity of the motion. This definition, in turn, permits the undamped shock spectrum of a shock motion to be defined as the maximum positive or negative response in terms of pseudo-velocity of a range of mass-spring elements of varying natural frequencies. As discussed below, for this design procedure, the shock spectrum will be presented on four-coordinate paper.

If it were actually possible to separate a piece of equipment into all of its mass-spring elements and determine the relative deformation at which each "spring" would fail, the pseudo-velocity could be plotted as a function of the natural frequency of the various equipment elements to obtain a graph that represents a curve which shows when the equipment would fail as a function of the frequency it is subjected to. This plot of velocity with frequency as an independent variable is called the "hardness spectrum" of the equipment. The term hardness is suggested in lieu of the older term fragility for positiveness, but in the context of this report the two are used synonomously.

A spectrum of this type is neither easy to obtain or simple to use; an alternate approach is called for. There are many shock machines and other shock environments, such as earthquakes, for which a pseudo-velocity spectrum has been obtained through electromechanical transducers or a bank of reed gages. If a piece of equipment is placed in this known shock environment and it survives, the shock spectrum calculated from the test environment represents a "survival spectrum" or a "hardness spectrum" for that particular piece of equipment. It is clear that a single piece of equipment could have several hardness spectra associated with it because it may have been shown to survive several different test environments. Regardless of this, the hardness spectrum has a direct relationship to the theoretical equipment hardness spectrum which was described earlier. It can be shown that a simple harmonic system subjected to a harmonic input at its natural frequency will have a vibrational displacement with amplitudes which increase with time and are ultimately limited by the time duration of the input, or the internal damping of the system (of which more will be said later), or both. The limiting value of the displacement is proportional to the amplitude of the input. Hence, it is possible to obtain, through experiment, a measure of the hardness of a particular piece of equipment. One needs only to be able to compare this hardness with the severity of an environmental input in a meaningful way. This comparison is facilitated by a standardized plotted procedure.

## FOUR-COORDINATE PAPER

Four-coordinate paper permits the interrelationship between the maximum value of the absolute acceleration, the maximum value of the relative displacement, the pseudo-velocity, and the frequency to be displayed in a single graphical plot. To see this reconsider Equation 9 and observe that if only maximum absolute values are considered

$$\left|\ddot{x}\right|_{max} = \omega^2 \left|z\right|_{max} \tag{13}$$

Also recall from Equation 11 that the pseudo-velocity can be written in the form

$$V = \omega |z|_{max} \tag{11}$$

These two equations are most revealing if written in logarithmic form. If the logarithms of the two equations are taken and recalling that $\omega = 2 \pi f$,

$$\log |\ddot{x}|_{max} = 2 \log 2 \pi f + \log |z|_{max} \tag{14a}$$

$$\log V = \log 2 \pi f + \log |z|_{max} \tag{14b}$$

If Equation 14b is subtracted from Equation 14a and the results are rearranged, Equation 14c appears in the form

$$\log |\ddot{x}|_{max} = \log f + \log V + \log 2 \pi \tag{14c}$$

Thus one can see that, in a log-log plot of the pseudo-velocity against an abscissa of frequency f, the curves which represent a constant acceleration are straight lines with a slope of -1 (i.e., the constant acceleration line is at 45 degrees with the horizontal). Equation 14b also shows that for the same plot the curves which represent constant displacement are straight lines with a slope of +1. These relationships are quite important; in many cases, both specification and testing procedures are such that constant acceleration, constant displacement amplitude, or constant pseudo-velocity would be maintained over a significant frequency range. Thus, it is reasonable to plot shock spectra on log-log paper with pseudo-velocity V as the ordinate and the frequency f as the abscissa. In addition, if a second set of orthogonal lines are introduced on the graph that are 45 degrees to the ordinate and abscissa, the four-coordinate graph paper of Figure 2 is produced, in which the lines of constant frequency, constant maximum absolute acceleration, and constant pseudo-velocity appear explicitly. Given any two of the four basic parameters for a shock spectrum, the other two associated parameters can be found immediately by reference to the four-coordinate plot.

The designation of the pseudo-velocity as the primary ordinate is not only a matter of graphical convenience. The pseudo-velocity is a measure of the stored peak energy in the system at a particular frequency and, thus, has a direct relationship to the survival or failure of this system.

Now return to the model of equipment as an assemblage of mass-spring elements. Looking at a single element, one can see that the energy stored in the deflected spring is U, where

$$U = \frac{1}{2} k z^2 \tag{15}$$

If both sides of Equation 15 are divided by the mass of the oscillator and notice is taken that the spring constant divided by the mass is the square of the circular frequency, it is found that

$$\frac{U}{m} = (\omega z)^2/2 \tag{16}$$

Since the pseudo-velocity has been defined as the product of the natural circular frequency and the maximum relative displacement, therefore can see that

$$V = \sqrt{\frac{2 U_{max}}{m}} \tag{17}$$

In other words, the pseudo-velocity can be interpreted as the square root of twice the peak energy per unit mass that is stored in the oscillator during the shock. This interpretation emphasizes the importance of pseudo-velocity as a meaningful measure of system response.

The discussion thus far has ignored the fact that there is some dissipation of energy in all mechanical systems. The effect of this damping on shock spectra will be considered in the following section. It is pertinent here, however, to make some observations on spectra and their four-coordinate plots that hold true whether or not damping is considered.

High frequency oscillators have very stiff springs. For an extremely stiff one, the mass could be expected to merely follow the acceleration of the foundation; thus, this oscillator would record a peak absolute acceleration equal to the peak acceleration of the pulse.

Low frequency oscillators are very flexible. If their foundations are given a very quick or short duration wiggle, they barely move until the foundation motion is over. If the shock being analyzed is one that begins and ends with zero velocity, the peak relative deflection will approach the peak absolute deflection; the peak deflection of all of the sufficiently low frequency oscillators will approach that level.

Thus, one can expect to see two asymptotic values: at the high frequency end of the shock spectrum the curve should approach the peak pulse acceleration, and at the low frequency end of the spectrum it should approach the peak shock deflection. For intermediate values of the frequency, the peak pseudo-velocity is often almost constant. Thus, an idealized shock spectrum is often drawn in the form shown in Figure 3; such idealized spectra are sometimes used in preliminary design analyses. Unfortunately, this rigid perception of the necessary character of shock spectra is not always observed, since the analysis is often not carried out sufficiently far to include the low frequency and high frequency asymptotes.

For example, imagine a 4-foot drop test onto some sort of a pad. The table drops, impacts, and the brakes bring the table to rest. The peak deflection was clearly 4 feet. But an analysis would have to include the constant 1-g fall, the impact, and the braking acceleration.

Usually only the impact portion of the record is analyzed, and one cannot hope to see a low frequency asymptote of 4 feet. Thus, here is a case, and a very typical one, in which the low frequency constant deflection asymptote would not appear because the shock spectrum analysis performed was not sufficiently complete.

## EFFECTS OF NONLINEARITIES AND DAMPING

Normally, the equipment is fabricated from solid elastic materials that deflect in direct proportion to an applied force and then spring back to their original position. These are linear systems, and their theoretical behavior is very well defined. Occasionally, however, equipment will be encountered that has construction which permits it to act nonlinearly. Perhaps the most common of these situations is one in which various elements of the equipment will bump into each other on the supporting cabinet, limiting their motion. Such insufficient rattle space is often the result of poor design and would, of itself, imply equipment rejection. On the other hand there are times when a good design requires limit stops or bumpers and nonlinearity cannot be avoided. Similarly, equipment that contains members whose support is such that the member does not deflect proportionately to an applied force, or members which can deform plastically, are also nonlinear in nature. It must be emphasized that the procedures discussed here are not applicable to such equipment. Often only environmental testing can be relied upon to qualify nonlinear components. Linear equipment response is basic to the proposed design procedure. This restriction should have little significance with regard to most mechanical and electromechanical equipment.

All physical motion involves some dissipation of energy. In many cases this energy dissipation is so small that it can be neglected entirely, as was done in the previous discussion. On the other hand, dissipation of mechanical energy should be discussed because it often involves nonlinear behavior that can be adequately approximated. An example of a common nonlinear energy dissipation mechanism is dry friction.

Practice has shown that the effects of nonlinear energy dissipation can usually be well approximated by linear dissipative behavior, particularly when the dissipation is small, but not negligible. One linear dissipative device is the theoretical "dashpot" which opposes motion with a force that is proportional to velocity. In the case of a mass-spring element, this damping can be assumed to be parallel with the restoring spring as shown in Figure 4. Since the velocity across it is the relative velocity between the mass and the rigid base, the governing differential equation becomes

$$m \ddot{x} = -k x - c \dot{x} \tag{18}$$

where the constant c is the damping coefficient. A nonlinear energy dissipation mechanism is often represented by a dashpot with a constant c which is so determined that the energy dissipation per cycle is the same for the true mechanism and for its linear substitution.

The value of c can become so large that, instead of oscillating, a disturbed mass will slowly return to its original position. Although such a large value of damping is seldom observed in the equipment, the value of c that corresponds to a transition between oscillatory and nonoscillatory behavior is important because it is a basis for measuring the amount of damping which is actually present in the structure. This transitionary "critical damping" coefficient is shown to have the value

$$c_c = 2\sqrt{k\,m} \tag{19}$$

Normally describe the amount of damping in a structure is described by specifying the fraction of critical damping which is present. Hence, one speaks of the damping ratio, $\zeta$, which satisfies the relationship

$$\zeta = c/c_c \tag{20}$$

Simple algebraic manipulation will show that Equation 18 can be written in the equivalent form (z is used to include the base motion, y.)

$$\ddot{x} = -\omega^2 z - 2 \zeta \omega \dot{z} \tag{21}$$

In the case of no damping this equation corresponds to the previous Equation 9.

Clearly, unless the damping ratio is quite small, it is no longer true to say the square of the frequency times the relative displacement is equal to the absolute acceleration. It is, however, standard practice to continue to use a four-coordinate plot to show damped spectra. A primary justification for this is the fact that damping does not affect the physical interpretation of the pseudo-velocity as a measure of the peak energy in the spring. On the other hand, the observed discrepancy reinforces the fact that only dampings below 10% of critical be considered when using the shock spectrum approach.

In general, damping reduces the values of the shock spectrum. All peak values are reduced. A 5% damped shock spectrum lies under an undamped shock spectrum, and a 10% damped shock spectrum lies under a 5% damped shock spectrum. High narrow peaks occur at the prominent frequencies and are usually clearly visible on the spectral record. The high narrow peaks are affected most by damping; the reasonably flat regions are affected the least. Since any real structure has some damping, a damped shock spectrum is often a more realistic representation of the behavior of the equipment. On the other hand, undamped shock spectra have the advantage of conservatism for expressing the severity of an environment, particularly in cases where the damping is small or unknown. In all but the most unusual cases, one can be sure that a piece of equipment which is predicted to survive an undamped severity spectrum will be overdesigned. However, undamped shock spectra equally will overstate the hardness of a piece of equipment with significant internal damping.

Undamped shock spectra represent the absolute peak response that could occur to the oscillator if it were truly undamped. High damping on the order of 8% is a reasonable assumption when many bolts and friction connections that can dissipate energy are present. Low or undamped shock spectra should be used for a single member, or for welded structures for which the smallest amount of yielding would constitute failure.

## DESIGN PROCEDURES

The shock hardness of a piece of equipment is the highest amplitude shock spectrum of any shock loading the equipment is known to have survived. The shock severity of an environment is the highest amplitude shock spectrum of every potential shock that may be delivered to the equipment. An example hardness curve and several example severity curves are given in Figures 5 through 9. Overall shock spectra are compared, rather than those which are experienced only while the shock is applied, or those which are generated only after the shock has terminated.

The damping of the hardness and severity spectra should be equal to, and in no case greater than, that which is reasonable for the equipment in question. Damping reduces all spectrum levels. Therefore, low damping inflates a hardness estimate, and high damping clouds a severity; to err in the opposite direction is conservative.

Clearly, any shock machine that can provide a progressively severe environment can be used to estimate the hardness of the equipment. The level must be increased until a failure is obtained. The time history taken just prior to that causing failure is used to compute a shock spectrum which is the estimate of the hardness. A complete determination requires measurement in three mutually perpendicular directions. Different shock tests provide different shock spectra and cause different failures. If one had a choice, one would select a test which imposes a motion similar to that of the potential environment.

The design procedure presented here has been implied in the previous discussion:

If a test spectrum which the equipment has survived ("the hardness curve") is greater than the environmental spectrum ("the severity curve") for all frequencies above the lowest natural frequency of the equipment, the equipment will survive.

## Hardness Exceeds Severity Everywhere

When the hardness curve everywhere exceeds the severity plot the equipment can be hard-mounted. The fasteners must be at least as strong as those used during the test which determined the hardness spectrum. The hardness catalog will indicate if any additional measures are required to attain the hardness displayed. The required rattle space must be estimated. Often this can be done by assuming that the lowest vibration mode corresponds to the equipment moving on its supports as a rigid body. When this is the case, the displacement shown by the severity curve for this lowest frequency is the required rattle space. In some cases a substructure, such as a cantilevered arm, is quite flexible, but has a natural frequency which is higher than that of the total package. This may represent the determining factor for clearances. In general, the required rattle space will be established by the greatest displacement shown by the severity curve at frequencies at or above the lowest for the equipment.

As an example of this type of situation, consider a case where one is called upon to design the installation of a blower, with hardness given in Figure 5, on a floor that is expected to have a potential environment as given in the severity curve of Figure 7. Tracing paper will be used to compare hardnesses and severities. To do this draw a vertical axis through the center of the tracing paper and a horizontal axis slightly above the middle of the page. Place the paper on the hardness plot in Figure 5 with the horizontal line over the 100-ips line on the four-coordinate paper and the vertical axis on the 100-Hertz line, and trace the hardness curve onto the sheet. Now position the tracing with the "100-100" axes properly placed on the severity curve of Figure 7. Note that the hardness everywhere exceeds the severity. For record purposes, trace the severity onto tracing paper, label the construction, and include it with the design analysis. A sample tracing is shown as Figure 10.

## Severity Somewhat Exceeds Hardness Only Below Low Mode Natural Frequency

The low mode frequency, $f_1$, must be estimated when it is not given in the Hardness Catalog. If the severity only exceeds the hardness below this frequency, no problem exists, and the equipment may be hard-mounted as above. The equipment has no modes which can build up a dynamic response in the region where S > H. The fact is that S > H in the low frequency region cannot cause a problem; all of the high frequency oscillators that were used in the severity shock spectrum calculation had to feel the low frequency capability of the excitation and this is reflected in the severity curve at the high frequencies.

The hardness and severity comparison is made as before by tracing the hardness onto tracing paper and marking the curve with an arrow at the lowest natural frequency. Since the mechanics are the same as previously described, a separate example for this situation is not given.

## Low Mode Frequency

In all cases where the severity exceeds the hardness, the lowest natural frequency of the equipment will have to be estimated for motion in the direction of interest.

One can test for it. A resonant search with the equipment placed on a vibration test shaker will usually make the first mode response and frequency visible to the eye; if not, an accelerometer successively placed at suspected points of motion will reveal the mode shape. Additionally, one could twang or bump the equipment with an accelerometer placed at suspected motion points and look at the trace on a storage oscilloscope; the resulting motion almost always contains the low mode frequency predominately, and the frequency can be counted on the stored "scope" trace.

One can also usually make a good estimate by calculating. The natural circular frequency of a simple oscillator in radians per second is given by

$$\omega^2 = k/m \tag{22}$$

In practice, one should always use the weight and never the mass, and so this should really be expressed as

$$\omega^2 = k\ g/w \tag{23}$$

The weight divided by the spring constant, w/k, is (if the mass is vertically over the spring) the static deflection, and so it can conveniently be written

$$\omega^2 = g/d \tag{24}$$

where d here represents the static deflection in inches and g is the gravitational constant, 386 in./sec$^2$. Frequency in the more normal cycles per second is the value of radians per second divided by 2 $\pi$. Using this value and the numerical value of g, an expression for natural frequency in cycles per second as a function of static deflection in inches is

$$f = 3.13/\sqrt{d} \tag{25}$$

Thus, any mass supported on a deflecting member with a static deflection of 1 inch has a natural frequency of 3.13 Hertz. These values can be read off the four-coordinante paper by looking at the intersection of the 1-g line and the static deflection of interest. Note that 1-inch deflection intersects the 1-g line at a frequency of 3.13 Hertz on Figure 2.

Thus, calculating the natural frequency can often be as simple as calculating a static deflection. One examines the equipment to identify the largest mass most flexibly supported, calculates its deflection due to gravity, and looks up the frequency on four-coordinate paper. Usually the 1-g line is drawn in red to make it stand out on the four-coordinate paper.

Finally, if at all possible, the low mode frequency should be reported with the hardness information.

## Severity Exceeds Hardness at Frequencies Above the Lowest Natural Frequency*

When no test spectra are available which guarantee that the hardness will exceed the severity above the lowest natural frequency, one must modify either the equipment or the environment which it observes. The design procedure is necessarily conservative. It must be assumed that the equipment may have natural vibration modes at any frequency above its lowest. Therefore, it must be assumed that any peak in the severity curve which rises above the hardness curve could excite motions that were not experienced during the hardness test. As a consequence, the only safe procedure that is usually available to the designer is to assume that the worst case can exist and to proceed accordingly.

*This section of the design procedure was prepared solely by Dr. Eubanks, so that a procedure could be outlined for at least some limiting cases. There are numerous approaches to isolator selection; many believe that each case should be treated separately, employing time history or modal analysis to simplified theoretical models of the equipment on the isolators. Others contend that only a test can suffice. Nonetheless, a designer approach can be outlined for use where the indicated conditions exist.

Normally a change in the hardness curve requires mechanical modification of the equipment and subsequent testing which is beyond the scope of the shock design engineer. Hence, the usual procedure followed is to reduce the severity of the environment felt by the equipment; this is accomplished by mounting the equipment on a shock isolator. A wide variety of such isolators is available, and many techniques exist for selecting and using them. It should be recognized that blind or haphazard selection can introduce new problems, such as the possibility of generating additional shocks through bottoming, or a requirement for additional rattle space. The subject is well beyond the scope of this manual; readers who desire additional information are urged to review the excellent report by Walsh and Blake (Reference 1).

There are two idealized cases of shock isolator introduction that can be incorporated into this design procedure with little difficulty: (1) the case where the weight of the isolator is small compared with that of the equipment and the isolator is considered to be an additional soft spring that has been introduced between the equipment and its mounting structure; and (2) the case where the isolator is so massive that the motion of the equipment has little effect on the isolator itself.

Negligible Isolator Mass. When the mass of the isolator is negligible, it can be thought of as representing an additional spring that acts in series with each of the masses of the lumped mass-spring elements of the model of the equipment. This situation is depicted graphically for the undamped equipment model in Figure 11. If the spring constant for the shock isolator is K, one can thus expect each modal mass to now be supported by a new spring with spring constant K', where

$$\frac{1}{k'} = \frac{1}{k} + \frac{1}{K} \qquad (26)$$

If Equation 26 is multiplied by the mass, m, each oscillator will now move with a circular frequency, $\omega'$, where

$$\frac{1}{(\omega')^2} = \frac{m}{k'} = \frac{m}{k} + \frac{m}{K} \tag{27a}$$

$$(\omega')^2 = \frac{\omega^2}{1 + \frac{k}{K}} \tag{27b}$$

Clearly, this new frequency is smaller than the original frequency even though the isolator spring constant may be rather large. Recalling that the frequency, f, is related to the circular frequency by the equation $f = \omega/2\pi$, one sees that

$$f' = \frac{f}{\sqrt{1 + \frac{k}{K}}} \tag{28}$$

and, if the logarithm of both sides of Equation 28 is taken, one has

$$\log f' = \log f - \log \sqrt{1 + \frac{k}{K}} \tag{29}$$

which shows that the horizontal abscissa of the severity spectrum is changed by a simple horizontal shift. If a plot of the severity spectrum is overlaid on a plot of the hardness spectrum, the relationship of Equation 29 simply requires that the entire severity spectrum be moved to the left the required amount. This procedure is advantageous because most spectra tend to have a decreasing pseudo-velocity at the higher frequencies. The approach can be effective in cases where the hardness spectrum begins to drop off more at lower frequencies than does the severity spectrum. This procedure is indicated by Figures 12 and 13. There can be low frequencies for which the severity curve is higher than the hardness curve. This is of no concern so long as these frequencies are below the lowest natural frequency of the equipment.

Application of this approach may require some estimates by the designer, since the equivalent spring stiffnesses for the equipment are often unknown. The weight of the equipment, W, and the lowest natural frequency, $f_o$, are known and can be used to estimate a value of k. Since

$$\omega^2 = (2\pi f)^2 = \frac{k}{m} = \frac{kg}{W}$$

it follows that

$$k = \frac{4 W \pi^2 f^2}{g}$$

For the blower of Figure 5a, its weight is 91 pounds and its lowest natural frequency is 40 Hertz. With g = 386 in./sec$^2$, an effective value of the spring constant would be k = 14,900 lb/in. The maximum horizontal deviation of the S and H curves of Figure 12 (at V = 30 ips) is the difference between 150 Hertz and 215 Hertz. This determines the frequency shift which must be generated. By Equation 28 one sees that

$$\frac{f}{f'} = \sqrt{1 + \frac{k}{K}} = \frac{215}{150} = 1.433$$

Thus, an isolator with constant K = 14,900 lb/in. will produce the needed frequency shift.

Figure 13 compares the shifted severity spectrum with the hardness spectrum. Note that a displacement of about 0.36 inch is indicated by the severity curve at the equipment natural frequency of 40 Hertz. This value is a guide to the rattle space as well as the displacement the isolator must be able to experience before the springs "bottom out."

Since stability would probably require that the equipment be supported on four springs, each of which has a spring constant of 3,725 lb/in. or less, a displacement of 0.36 inch would present no problem.

The immediately preceding analysis assumed that the lowest mode for the equipment was one in which the equipment moved on its supports as a rigid body. If the lowest mode is one in which a substructure experiences the greatest motion, then the weight of that substructure should be used to estimate k.

The proposed procedure is inherently conservative. At higher equipment frequencies the effective value of k will almost always be higher than that which is used to compute the frequency shift. Hence, in practice, the severity will not only be shifted in frequency, but its frequency range will also be compressed. This means that a true version of Figure 13 would show a greater gap between the S and H curves. The conservatism of this approach is increased by the lack of knowledge of the ever-present damping.

Only vertical motion is considered in this example. Similar analyses must also be carried out in the other two perpendicular directions of translation.

Significant Isolation Mass. The preceding analysis assumed that the effective mass of the isolator could be neglected. In many cases this cannot be done, and at each equipment frequency considered one must analyze a two-degree-of-freedom system similar to that of Figure 14, where $f_2$ is the frequency of the isolator and $\zeta_2$ is its relative damping, while $f_1$ is the equipment modal frequency and $\zeta_1$ is its relative damping.

Shock design procedures become extremely complicated when the mass of the isolator is a significant fraction of the mass of the equipment. All simple procedures available can lead to extreme overdesign. Accurate analyses cannot be conducted on the basis of the simple knowledge of spring stiffness ratios and frequency ratios. Instead, a rigorous analysis would require that the value of every modal mass and every associated spring constant be known and, indeed, that the designer have available the true time history of the environmental shock. Approximate analyses of limited accuracy presently used require the construction of an artificial time history and an extensive computer analysis to obtain results of questionable validity. It is clear that a procedure presented in a manual such as the present one must carry with it the danger of extremely conservative overdesign.

Many of the complications that arise in the analysis of structural systems with many degrees of freedom come about because of the interaction, or "coupling," of the possible motions. To state this problem simply in the present case, it must be noted that the motion of the shock isolator will be affected by the motion of the supporting element as well as by the forces of the environmental shock. Hence, it appears one must know the motion of the equipment before one can find it. This assumption of complete decoupling is equivalent to a situation where the weight of the equipment is negligible compared with that of the shock isolator. In a typical "heavy isolator" case, where the isolator weight may be only 20-25% of the equipment weight, this assumption can, obviously, be much in error at lower frequencies. However, if complete decoupling is assumed, a safe procedure can be generated for the identification of an isolator that will permit the equipment to survive the environment.

Heavy shock isolators will not only shift the frequency at which a response occurs, but they will also change the magnitude of that response. Some difficulties arise because this change can actually involve an increase in the response amplitude of the equipment. This design procedure, which is constructed in an attempt to ensure that the combination of response amplification and frequency shift will result in an input spectrum which can be survived by the equipment, is based on the use of amplification curves. These curves utilize the input shock and time-dependent uncoupled isolator response that were generated by Mindlin (Ref 8) for the development of a related set of amplification curves. In the present application, however, pseudo-velocity amplification is considered.* These amplification curves are presented in Figures 15

*Accurate numerical computation is much easier in 1978 than it was 35 years ago. Some of the "fine structure" missing from Mindlin's has been included. Also, a few errors have been corrected.

through 19. Each figure is drawn for a different isolator damping ratio and contains curves for equipment damping ratios of 0.005, 0.01, 0.05, and 0.10. Since isolator damping can be introduced artificially, a graph for isolators with 50% damping is included.

This design procedure is best explained by reference to Figure 12. Here one can see that S exceeds H in a frequency range $80 \leq f \leq 320$; the entire range lies above the lowest natural equipment frequency of 40 Hertz. For this specific example there are three comparison points. These are the lowest natural frequency for the equipment at 40 Hertz, the crossover point at 80 Hertz, and the frequency at which the pseudo-velocity for S minus the pseudo-velocity for H is the largest. Here, this is approximately 150 Hertz, where the respective magnitudes are 55 in./sec and 30 in./sec.

An isolator will be selected on the basis that the amplification, A, must be less than the ratio of the hardness pseudo-velocity to the severity pseudo-velocity at each comparison point. It is convenient to construct a table:

| $f_1$ (Hz) | S (ips) | H (ips) | Required |
|---|---|---|---|
| 150 | 55 | 30 | $A \leq 0.55$ |
| 80 | 85 | 85 | $A \leq 1$ |
| 40 | 100 | 160 | $A \leq 1.6$ |

In the absence of more specific information, the equipment is assumed to have 1/2% damping ($\zeta_1 = 0.005$) so that one works with the top curve of each figure. Referring to Figure 15, A = 0.55 corresponds to a frequency ratio $f_1/f_2 = 2.8$. The frequency of an isolator with 1/2% damping is $f_2 = f_1/2.8 = 150/2.8 = 54$ Hertz. This is higher than $f_o$, the lowest equipment natural frequency, and since the amplification curve reaches a peak of 36 for $f_1/f_2 = 1$, it is unacceptable. The same conclusion is reached for all other isolator damping ratios except that represented in Figure 19, where $\zeta_2 = 0.5$ (50% damping). In this case, A = 0.55 corresponds to a frequency ratio $f_1/f_2 = 3.4$, which yields an isolator frequency $f_2 = f_1/3.4 = 150/3.4 = 44$ Hertz. Although $f_2 > f_o$, the peak amplification at $f_1/f_2 = 1$ is about 1.5, and the crossover point is at $f_1/f_2 = 80/44 = 1.8$, where $A = 0.82$. Thus, all criteria are satisfied, and an isolator with 50% damping and a frequency of 44 Hertz or less will insure equipment survival.

Such highly damped isolators may not be available. If, for example, the available isolators were essentially undamped, one would use the curves for $\zeta_2 = 0.005$. For $\zeta_1 = 0.005$, $A = 1.6$ corresponds to $f_1/f_2 = 1.7$, so that at $f_1 = 40$ Hertz, $f_2 = 40/1.7 = 23.5$ Hertz. A check for $f_1 = 80$ yields $f_1/f_2 = 80/23.5 = 3.4$ with corresponding $A = 0.4$, while a check for $f_1 = 150$ yields $f_1/f_2 = 150/23.5 = 6.4$ with $A < 0.2$. The isolator with very low damping and a frequency of 23.5 Hertz or less is acceptable.

Figures 15 through 19 show that shock isolator selection often reduces to a trade-off between low isolator frequency and high isolator damping. The designer is, however, cautioned that highly damped oscillators do not work well at high equipment frequencies. This behavior is often mitigated by a rapid decrease in spectral pseudo-velocity with frequency (constant acceleration response).

A very low isolator frequency is also to be avoided, since it may lead to a very large required rattle space or an unstable equipment platform. This behavior may be mitigated by the constant displacement characteristic of many spectra at low frequencies.

## SEVERITY AND HARDNESS DATA

There are two types of severity requirements to which the procedures of this manual can be applied. It is possible that a particular test specification will be prescribed for a piece of equipment known to be capable of surviving a different test. If the spectra for both test environments are known, the design procedures can be used to ensure and certify compliance. The very complete survey by Clements (Ref 9) is valuable for this application. Clements collected spectra for Navy

Hi-Impact Shock Simulation Devices available in 1972. His survey includes complete discussions of such variables as load weight and measurement location that must be taken into account in a careful analysis.

A second way of imposing a severity requirement is to describe the environment (an earthquake, for example) by presenting a time-dependent wave form (time domain) or a shock spectrum (frequency domain). If the time domain description is used, standard procedures will convert this into a shock spectrum. The procedure is described in Reference 7, which also includes a computer code. Once the severity spectrum has been given to the designer, he need only compare this to the hardness spectrum which the equipment is known to be able to survive.

It is clear that the major designer requirement is a hardness catalog which implicitly or explicitly presents hardness spectra for different equipment items. One form of a catalog of this nature was presented in Reference 2, where the results of specified equipment tests were given and cross-indexed to the spectrum for the shock simulator used. Since documentation was required for all such tests, voluminous information of this type was available throughout the military establishment. The major Navy project which was established to collect and index this material (Project STARSHINE) is now dormant; no other major depository is known. It is fortunate that some military testing agencies have on file hardness information on many of the equipment items with which they are concerned.

The hardness catalog of this report was obtained from data collected during a large testing program. The Safeguard Facilities Project was carried out by the Huntsville Division of the Corps of Engineers, who tested a huge amount of technical support equipment to establish facility hardness (Ref 10). Those data have been made available and summarized; they are included in Appendix B as a hardness catalog.

Hardness data increase in value as they become more extensive and more available. The Civil Engineering Laboratory will assist in information collection and dissemination on shock design procedures and hardness data to the extent permitted by its primary mission.

## FINDINGS AND CONCLUSIONS

1. A simplified design procedure for equipment installation in shock environments has been presented. Workable definitions of equipment hardness and environmental severity have been provided.

2. The overall shock spectrum for the three orthogonal directions for the equipment attachment points is the measure of choice to be recorded in equipment hardness catalogs. Four-coordinate paper with pseudo-velocity as the ordinate is the plotting method of choice.

3. The shock spectrum of the anticipated most severe shock environment is to be plotted on four-coordinate paper and used for direct comparison with the equipment hardness. It is termed the shock severity.

4. The procedure is definite when the equipment hardness can be shown to exceed the anticipated shock severity. The procedure is imprecise where the severity exceeds the hardness above the low mode frequency. Only in limiting cases as presented by Dr. Eubanks does any convincing procedure exist. Many authors, therefore, recommend the modeling of the isolators with a simplified theoretical model for such situations. A detailed procedure for that approach is beyond what could be included here.

5. Equipment hardness can be cataloged effectively with damped or undamped shock spectra, and an example of such a catalog is given in Appendix B.

## RECOMMENDATIONS

1. Additional effort at accumulating hardness data for technical support equipment in the form of shock spectra is to be encouraged. This type of effort can be efficiently carried on at a low key, monitoring the

efforts of other DOD designs as was done in the case of this work unit with the Safeguard data. Panic last-minute crash programs are totally unreliable; one is forced to accept what the budget permits in the time frame allowable, regardless of the integrity. Shock design is one area where steady accumulation and history taking is far more cost-effective than heroic last-minute efforts.

2. Continued participation with Corps of Engineers at Huntsville in hardening studies is to be encouraged. The Laboratory can continue to play a clearinghouse role for accumulation and distribution of these data.

3. As has been alluded to, computer simulation of generic equipment models should reveal some more definitive isolation procedures, at worst multi degree-of-freedom calculation routines for the hand-held programmable calculators can provide estimates of isolator effectiveness. In this manner a defined procedure can be provided without the limiting restrictions of Dr. Eubanks' procedures.

4. The preliminary version of this document, Reference 11, predicted a more straightforward approach. A significant contribution of Dr. Eubanks was to point out that the theoretical justification of the procedure was inadequate; that no matter how straightforward, the approach could not be recommended until it could be thoroughly substantiated to finest detail. A directed study using perhaps four and five degree-of-freedom equipment models should be conducted to define shock isolation application in these situations and provide proof of type of filtering and protection available from commercially available shock isolators.

5. The careful reader will note ambiguity or at least imprecision in the suggestions for selecting a damping factor for use with the shock spectra in this procedure. It is possible that additional study could clarify this situation; for example, during the course of this research, a proof was developed that no damping higher than 8% could be realized from either dry friction or structural yielding. This proof has not

been refuted. It may be that in fact a damping maximum does exist and could be incorporated in the design procedure so that one does not mistakenly assume excessive damping levels from yielding structure as has often been done in the past.

6. As was mentioned, a preliminary version of this design procedure (Ref 11) was issued as an informal draft maual for comment. One excellent suggestion that was offered was to include copious example problems; time did not permit that effort. It has been noted that Reference 6 is effective because examples are precisely what it does present. Perhaps shock design is an area where generalization has been rampant, and anything the least bit specific has been avoided. At any rate, it is suggested that example problems be accumulated and compiled for any future revisions of shock design publications.

## ACKNOWLEDGMENT

The Laboratory is indebted to the U.S. Army Corps of Engineers, Huntsville Division, and especially Mr. Richard J. Bradshaw, Jr., for making available the extensive hardness test data taken in designing the facilities for the Safeguard Project. Their scientifically organized approach made possible the ready accumulation of their results into catalog form (Appendix B).

## REFERENCES

1. Naval Research Laboratory. Report 3596: The determination of shock isolator performance, by J. P. Walsh and R. E. Blake. Jan 1950.

2. Naval Research Laboratory. "Shock hardening of equipment," by R. A. Eubanks and B. R. Juskie, Shock and Vibration Bulletin 32, part III, p 23ff, 1963.

3. Department of the Army, Office of the Chief of Engineers. Study of shock isolation for hardened structures. Washington, D.C., Jun 1966. (AD 639303)

4. Air Force Weapons Laboratory. Technical Report No. 74-102: The Air Force manual for design and analysis of hardened structures, by R. E. Crawford, C. J. Higgins, and E. H. Bultmann. Oct 1974. (AD B004252)

5. Naval Facilities Engineering Command. NAVFAC P-355: Seismic design for buildings. Alexandria, Va., Apr 1973.

6. __________. NAVFAC P-355.1: Seismic evaluation of supports for existing electrical-mechanical equipment and utilities. Alexandria, Va., Mar 1975.

7. Civil Engineering Laboratory. Technical Note N-1590: Shock spectrum calculation from acceleration time histories, by H. A. Gaberson. Port Hueneme, Calif., Sep 1980.

8. Mindlin, R. D. "Dynamics of package cushioning," Bell System Technical Journal, vol 24, p 343ff, Jul-Oct 1945.

9. Naval Research Laboratory. Report 7396: Shipboard shock and Navy devices for its simulation, by E. W. Clements. Jul 1972.

10. Boyd, M. A., and C. C. Huang. "Experiences on shock isolation of equipment in the SAFEGUARD system," Shock and Vibration Bulletin No. 47, part 1, pp 151-161, Sep 1977.

11. __________. Letter Report No. YF53.534.006.01.017: A design method for shock resisting equipment installation, by H. A. Gaberson. Port Hueneme, Calif., Apr 1975.

## BIBLIOGRAPHY


Air Force Intelligence Center. Vulnerability handbook for hardened installations, vol I, chapter 12 (revised). Washington, D.C., Aug 1965. (AD 365316)

Air Force Special Weapons Center. Technical Documentary Report AFSWC-TDR-62-64: A guide for the design of shock isolation systems for underground protective structures. Kirtland Air Force Base, N.M., 1962. (AD No. 298578 Section 4)

Air Force Systems Command, Air Force Weapons Laboratory. Technical Report No. RTD-TDR-63-3096: Design procedures for shock isolation systems of underground protective structures; Vol I, Structure interior motions due to air blast induced ground shock; Vol II, Structure interior motions due to directly transmitted ground shock; Vol III. Response spectra of single-degree-of-freedom elastic and inelastic systems; Vol IV, Response spectra of two-degree-of-freedom elastic and inelastic systems; Vol V, Response spectra of multi-degree-of-freedom elastic systems. Kirtland Air Force Base, N.M., 1963 to 1965.

Clough, R. W., and J. Penzien. Dynamics of Structures. McGraw-Hill Book Co., 1975.

Jacobsen, L. S. and R. S. Ayre. Engineering Vibrations with Applications to Structures and Machinery. McGraw-Hill Book Co., Inc., 1958.

Jet Propulsion Laboratory. NASA Technical Memorandum 33-484: Design of subsystems in large structures, by T. K. Caughey. Pasadena, Calif., 1971.

Newmark, N. M., and E. Rosenblueth. Fundamentals of Earthquake Engineering. Prentice-Hall, Inc., 1971.

University of Illinois. Civil Engineering Studies, Structural Research Series No. 396: Approximate dynamic response of light secondary systems, by T. Nakhata, N. M. Newmark, and W. J. Hall. Urbana, Ill., 1973.

U.S. Army Construction Engineering Research Laboratory. Technical Report S-23: A guide for the design of shock isolation systems for ballistic missile defense facilities, by R. F. Crawford, C. J. Higgins, and E. H. Bultmann. Champaign, Ill., Aug 1971.

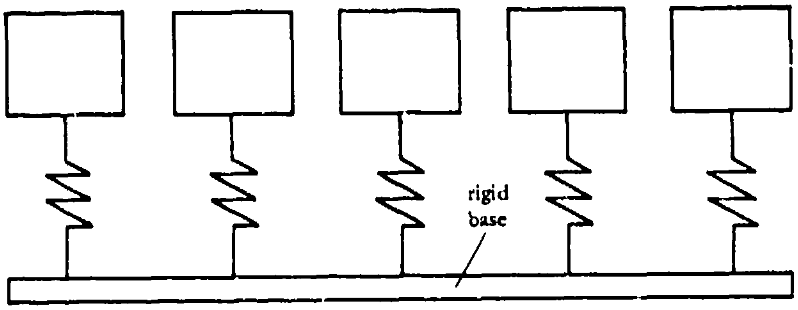


Figure 1. Idealized model of equipment.

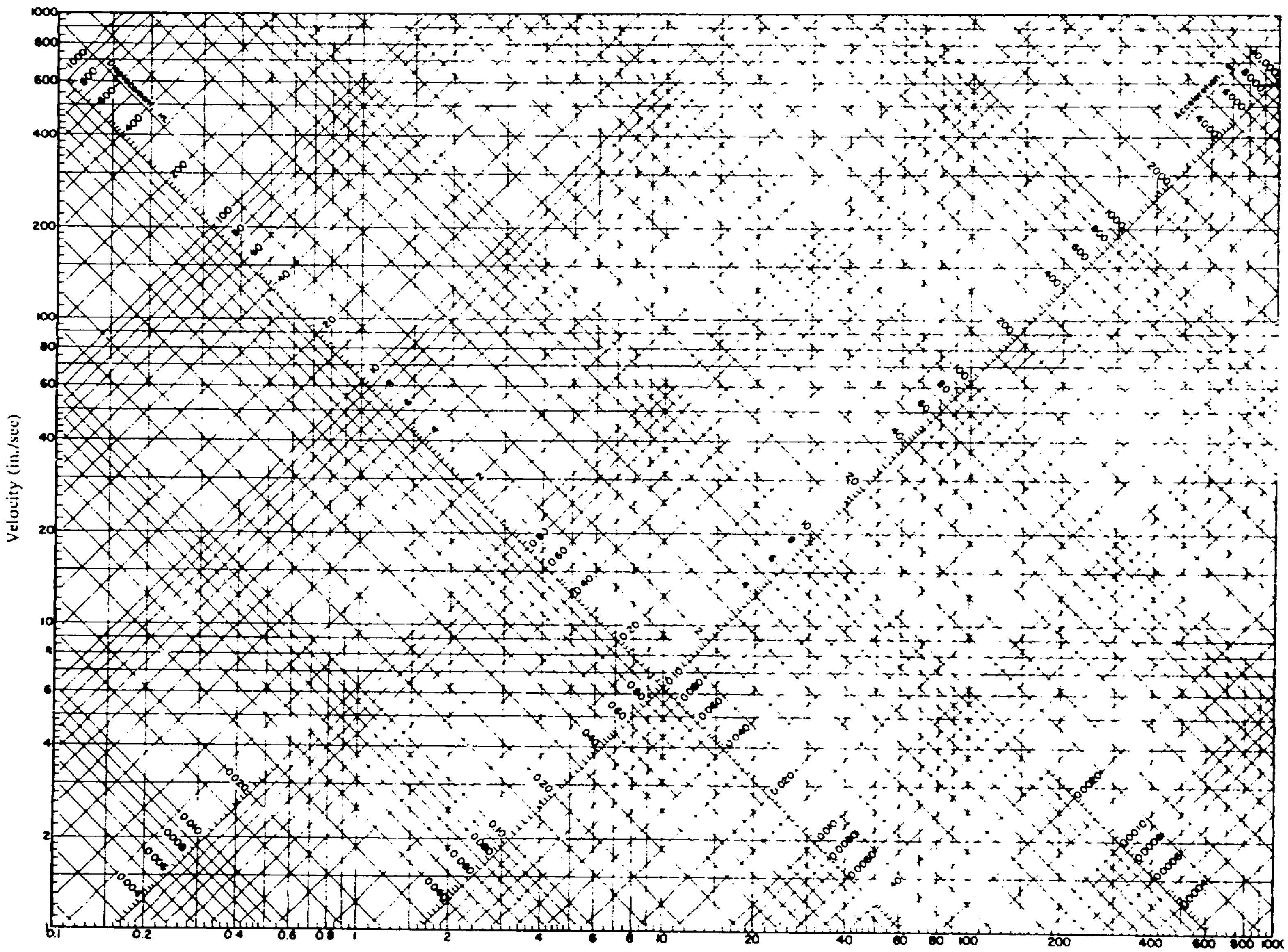

Figure 2. Four-coordinate paper.

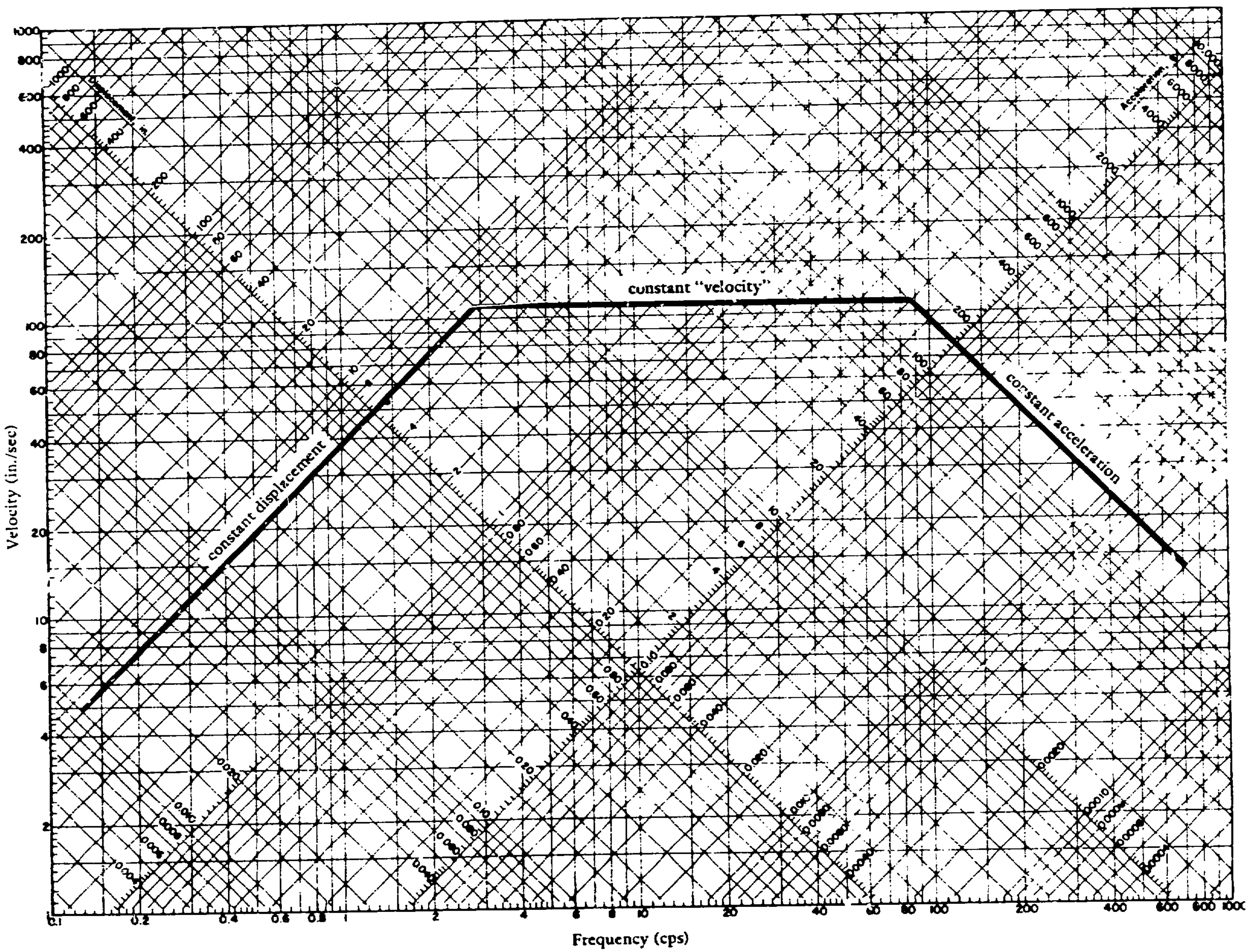


Figure 3. Idealized shock spectrum.

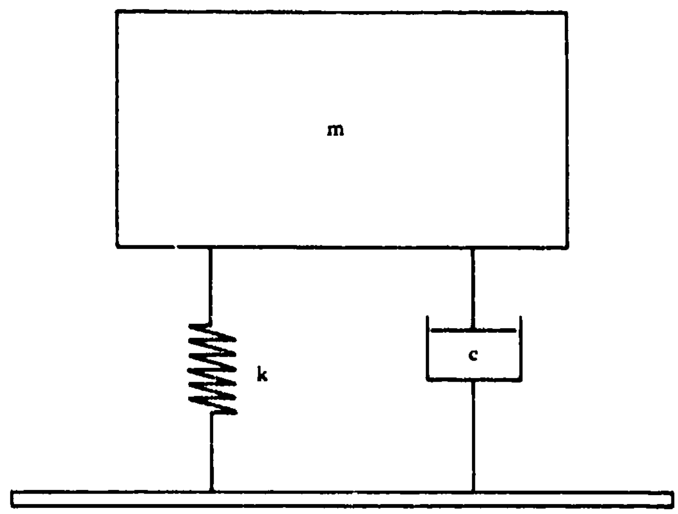


Figure 4. Linearly damped lumped element.

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SHOCK DATA ANALYSIS

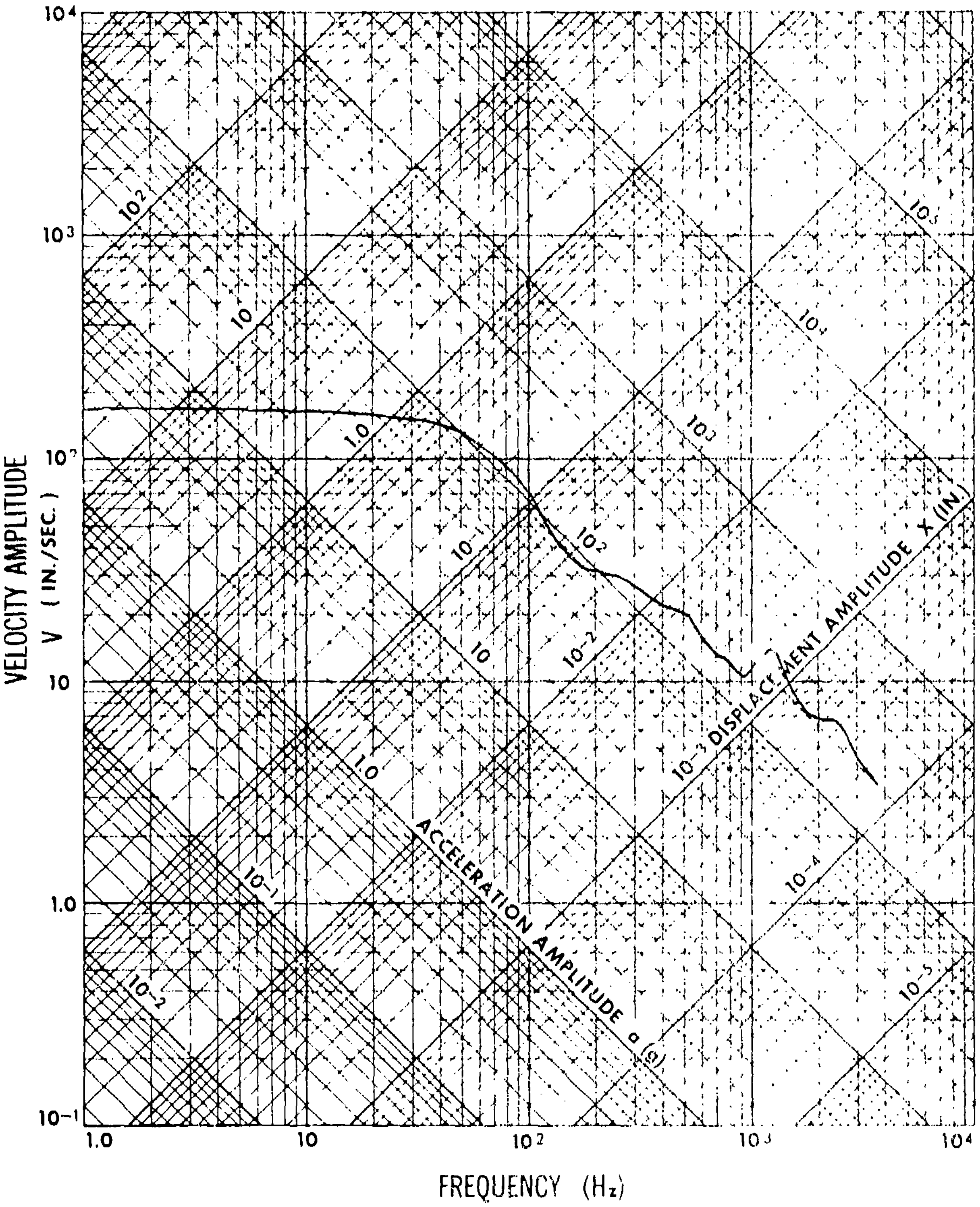


Figure 5a. Illustrative hardness (not certified for design use).

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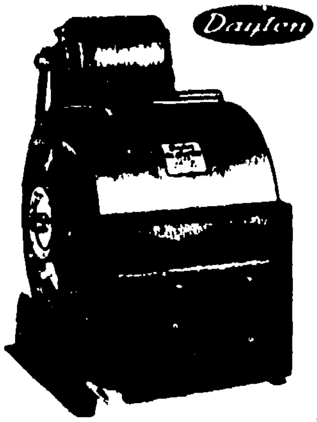
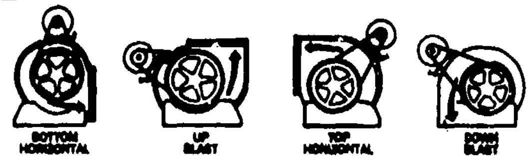



| Wheel Dia. | Wheel W | Blower Dimensions: Shaft Dia. | Outlet H | Outlet W | Overall H | Overall W | Overall D | Less Motor & Drive: Stock No. | Each | Shpg. Wt. |
|---|---|---|---|---|---|---|---|---|---|---|
| 7½" | 7" | ½" | 7½" | 9¼" | 14" | 14" | 12" | 2C970 | $28.63 | 14 |
| 9½ | 9½ | ½ | 10½ | 11¾ | 17 | 15 | 15 | 2C972 | 29.84 | 21 |
| 10⅝ | 10⅝ | ½ | 11½ | 13½ | 19 | 17 | 17 | 2C974 | 32.98 | 25 |
| 12⅝ | 12⅝ | 1 | 13½ | 15¾ | 22 | 20 | 19 | 2C976 | 44.84 | 37 |
| 15 | 15 | 1 | 15½ | 18½ | 26 | 23 | 23 | 2C978 | 68.11 | 66 |
| 18⅛ | 18 | 1 | 18½ | 21⅞ | 31 | 29 | 27 | 2C980 | 131.58 | 108 |

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|---|---|---|---|---|---|---|---|---|---|---|---|
| 7½" | 1075 | 940 | 765 | 570 | 1030 | 1/4 | 115 | Split | 7C672 | $61.26 | 35 |
| 9½ | 1580 | 1350 | 1000 | — | 635 | 1/4 | 115 | Split | 7C655 | 63.64 | 42 |
| 10⅝ | 1750 | 1400 | — | — | 492 | 1/4 | 115 | Split | 7C657 | 65.43 | 46 |
| | 2050 | 1790 | 1490 | — | 560 | 1/3 | 115 | Split | 7C659 | 70.48 | 49 |
| 12⅝ | 2680 | 2280 | 1700 | — | 425 | 1/3 | 115 | Split | 7C661 | 84.77 | 64 |
| | 3140 | 2830 | 2420 | 1800 | 500 | 1/2 | 115 | Split | 7C663 | 98.26 | 65 |
| 15 | 3950 | 3450 | 2800 | — | 405 | 1/2 | 115 | Split | 7C665 | 122.54 | 91 |
| | 4550 | 4150 | 3600 | 2850 | 455 | 3/4 | 115/230 | Cap. | 7C667 | 146.53 | 95 |
| 18⅛ | 6300 | 5700 | 4500 | — | 350 | 1 | 115/230 | Cap. | 7C670 | 223.12 | 149 |
| | 6300 | 5700 | 4500 | — | 350 | 1 | 230/460 | 3-Ph. | 7C604‡ | 207.88 | 149 |

## PERFORMANCE FOR HEATING AND AIR CONDITIONING SYSTEMS

| CFM Air Delivery at RPM Shown: Wheel Dia. | Free Air | 1/2" SP | 5/8" SP | 3/4" SP | Blower RPM | Blower with 1725 RPM Motor and Drive: HP | Motor Data Volts 60 Hz | Type | With Automatic Thermal Protection Stock No. | Each | Shpg. Wt. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 7½" | * | 780 | 450 | — | 1190 | 1/2 | 115 | Split | 7C070 | $87.93 | 39 |
| 9½ | * | 1840 | 1580 | 1000 | 959 | 1/2 | 115 | Split | 7C071 | 90.13 | 46 |
| 10⅝ | * | 2080 | 1660 | — | 837 | 1/2 | 115 | Split | 7C072 | 89.99 | 50 |
| | * | 2500 | 2200 | 1800 | 910 | 3/4 | 115/230 | Cap. | 7C073 | 116.34 | 54 |
| 12⅝ | * | 3100 | 2680 | — | 692 | 3/4 | 115/230 | Cap. | 7C074 | 134.05 | 69 |
| | * | 3600 | 3280 | 2840 | 750 | 1 | 115/230 | Cap. | 7C075 | 141.39 | 74 |
| | * | 3600 | 3280 | 2840 | 750 | 1 | 230/460 | 3-Ph. | 7C062‡ | 124.40 | 74 |

(*) These blowers should not be installed in a system having below ½" static pressure; see table above for air delivery between free air and ⅜" SP. (‡) 230/460V, 60 Hz, 40°C, 3-phase motors—NOT thermally protected.

**SEE WARRANTY INFORMATION ON PAGE BEFORE INDEX** 721

Figure 5b. Reproduced catalog page, identifying blower and giving additional details.

NAVFAC / NCEL
SHOCK DATA ANALYSIS

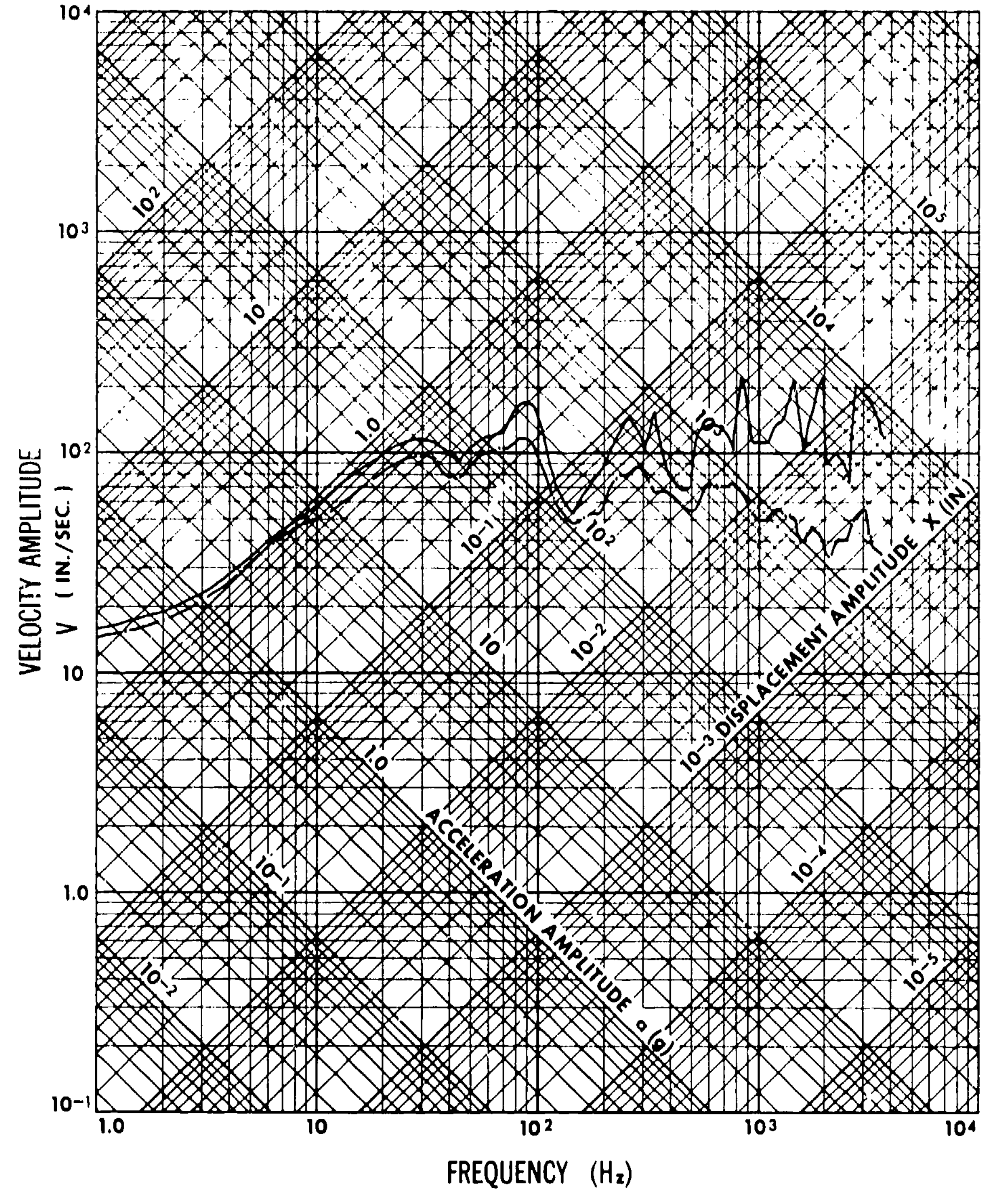



Figure 6. Illustrative severity, MIL-S-901C.

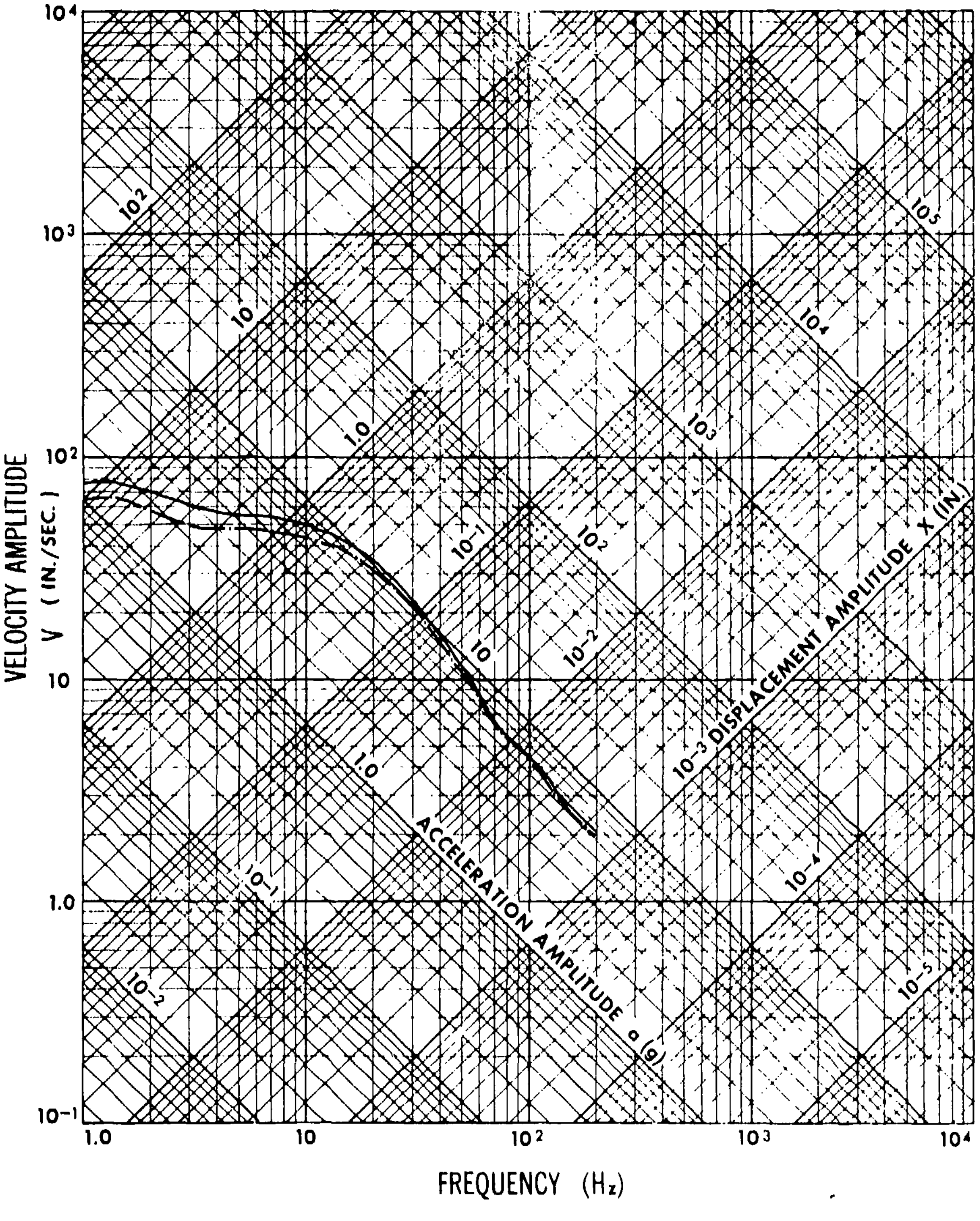



Figure 7. Illustrative severity, direct induced ground motion due to a nuclear blast.

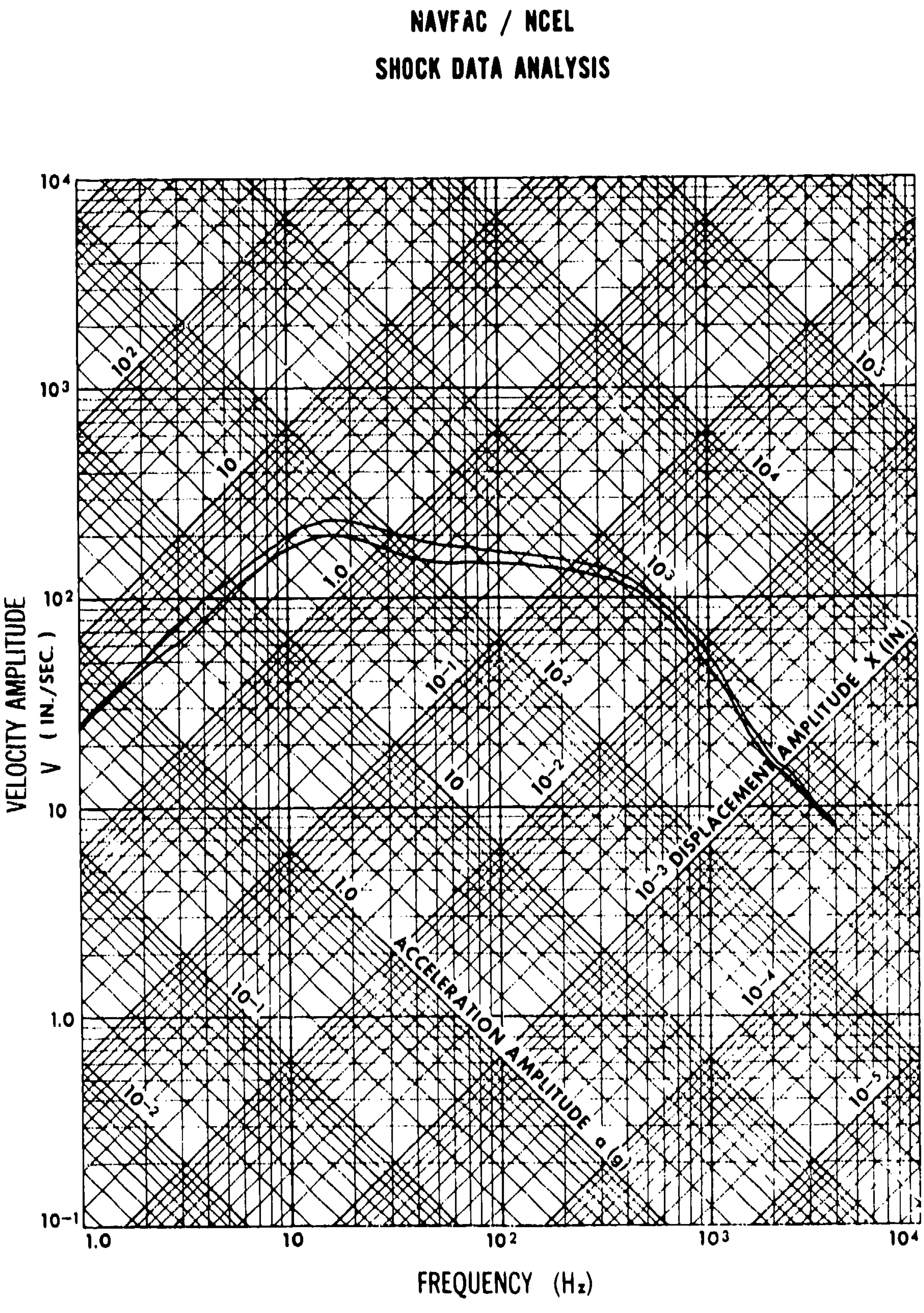



Figure 8. Illustrative severity, air induced ground motion from a nuclear blast.

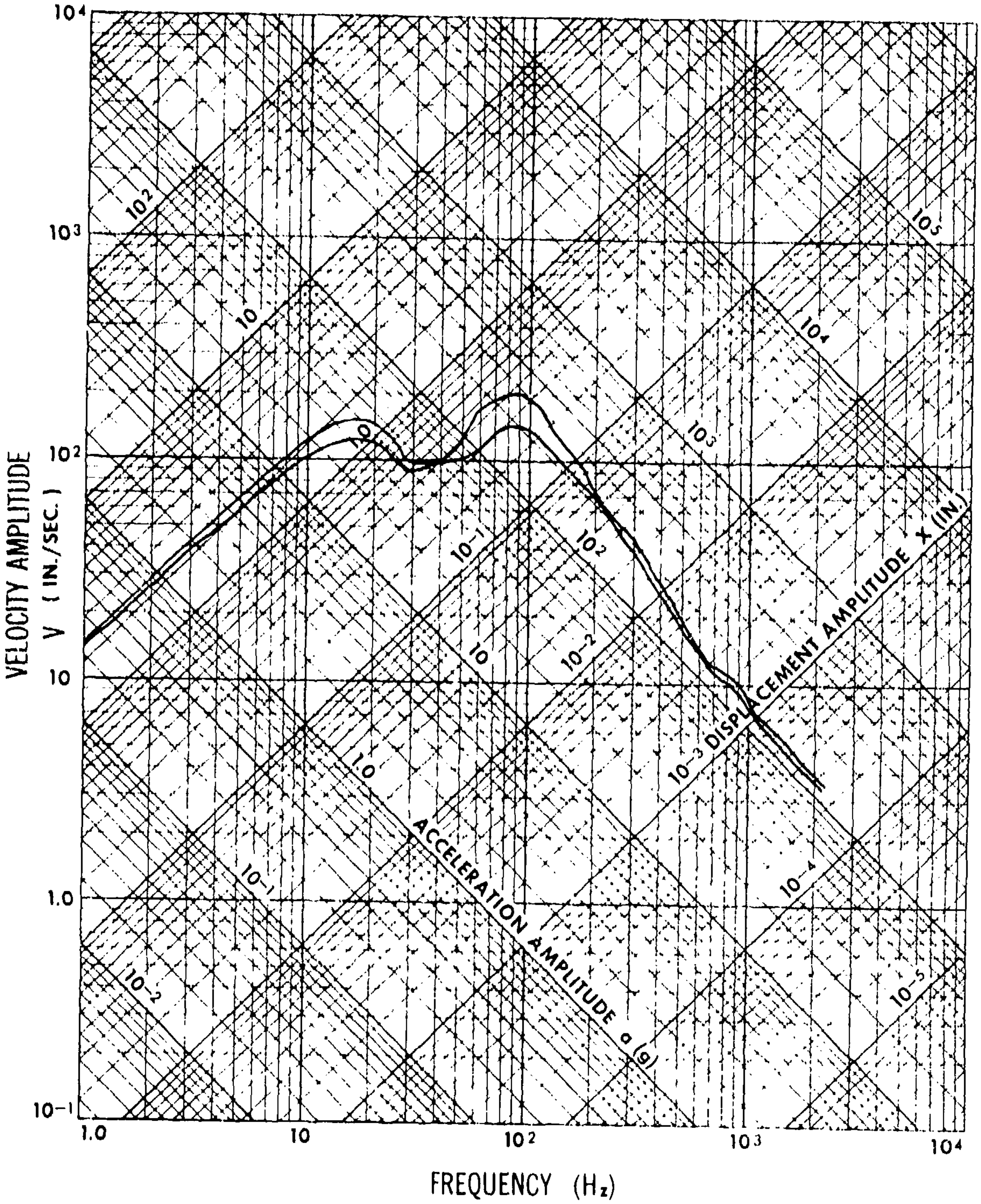



Figure 9. Illustrative severity, underwater explosion.

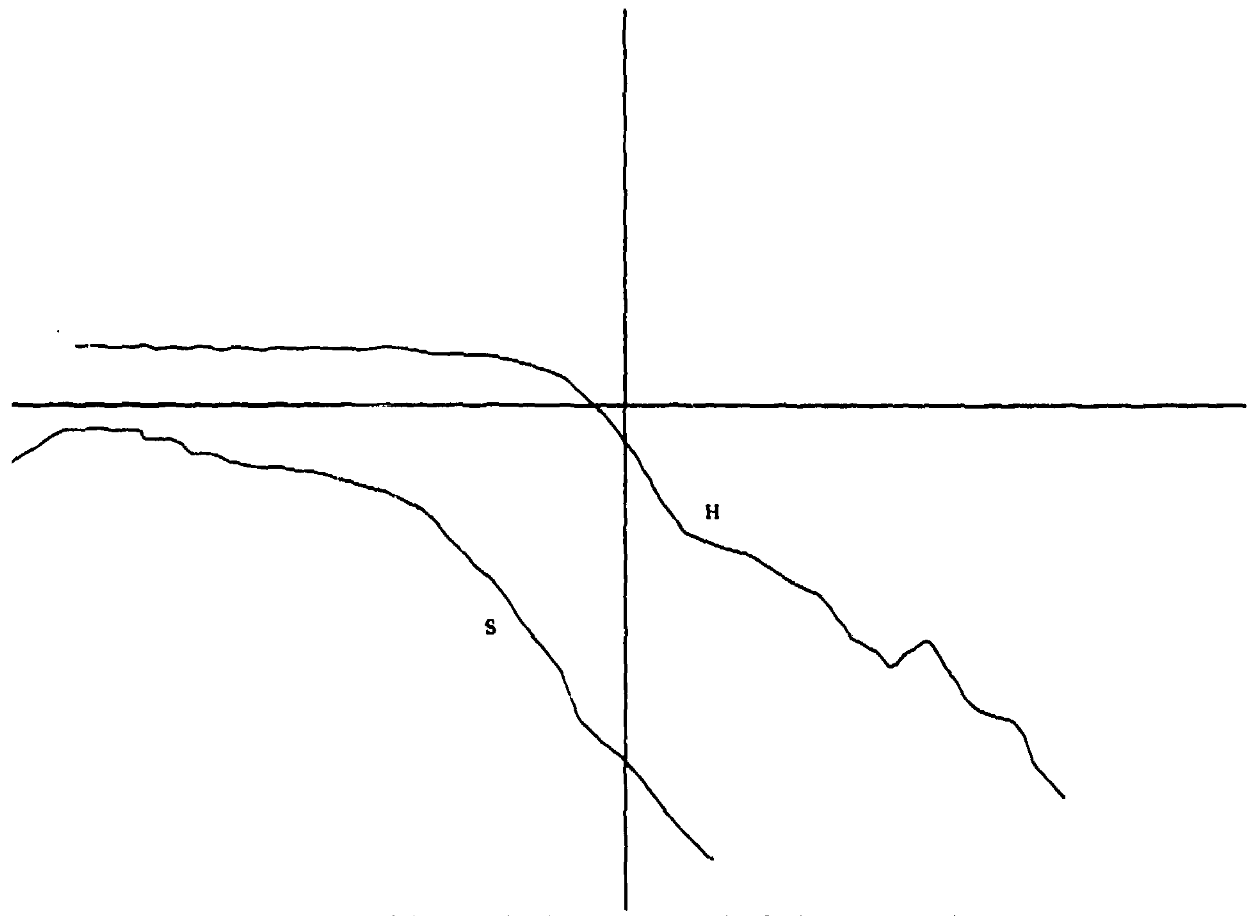



Figure 10. Vertical direction hardness versus severity for blower shown in Figure 5.

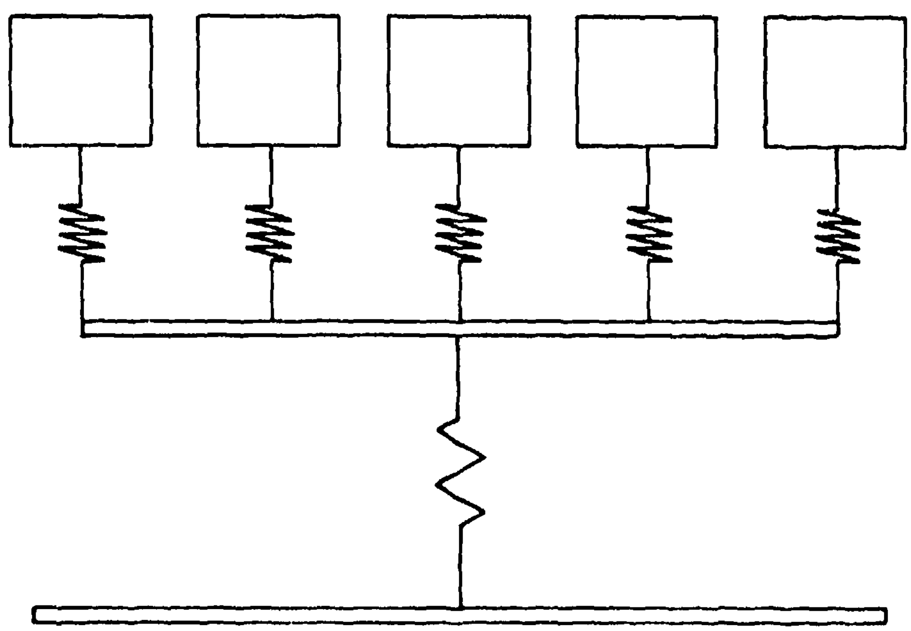


Figure 11. Idealized model of equipment on light isolator.

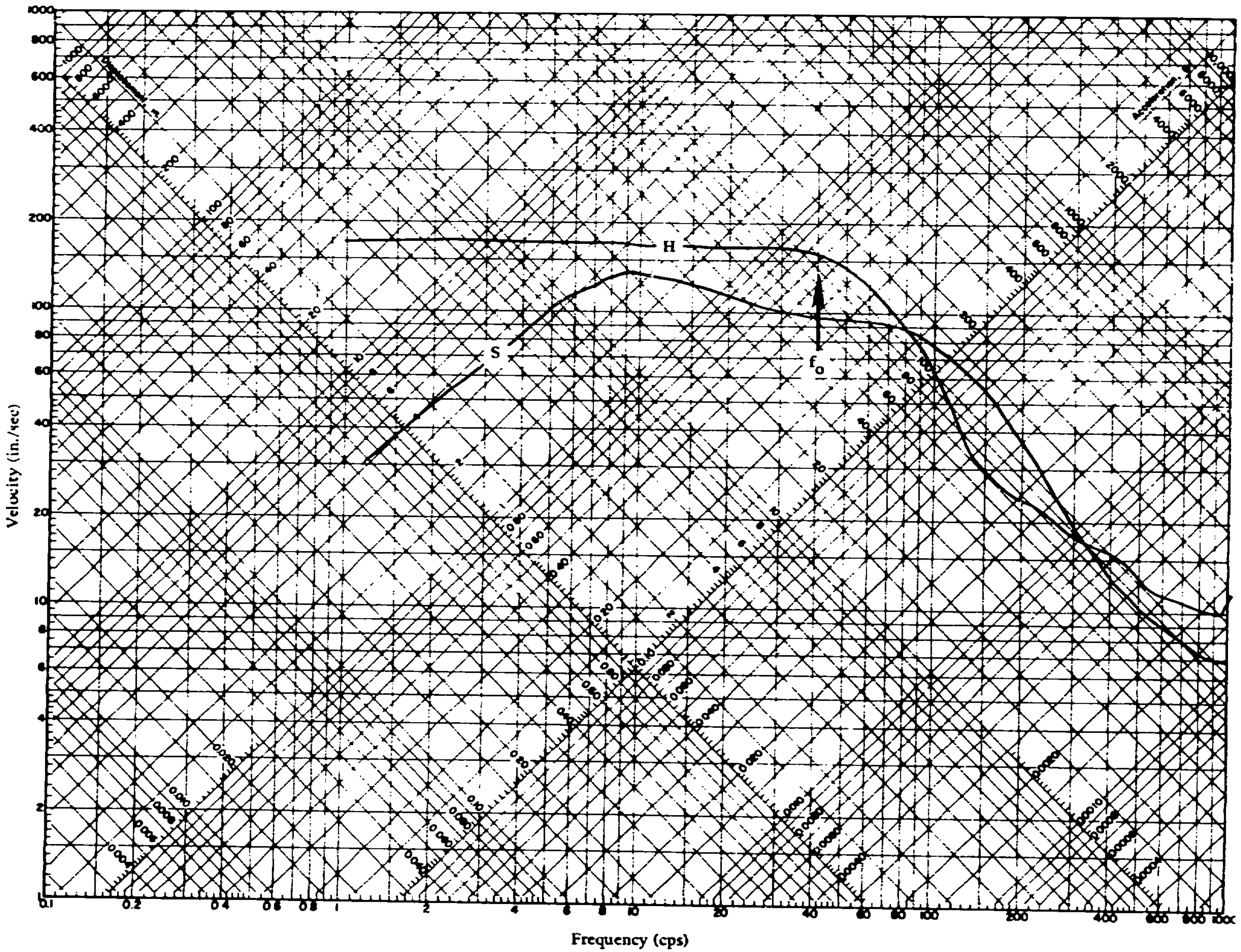



Figure 12. Example of severity exceeding hardness between 80 Hertz and 320 Hertz.

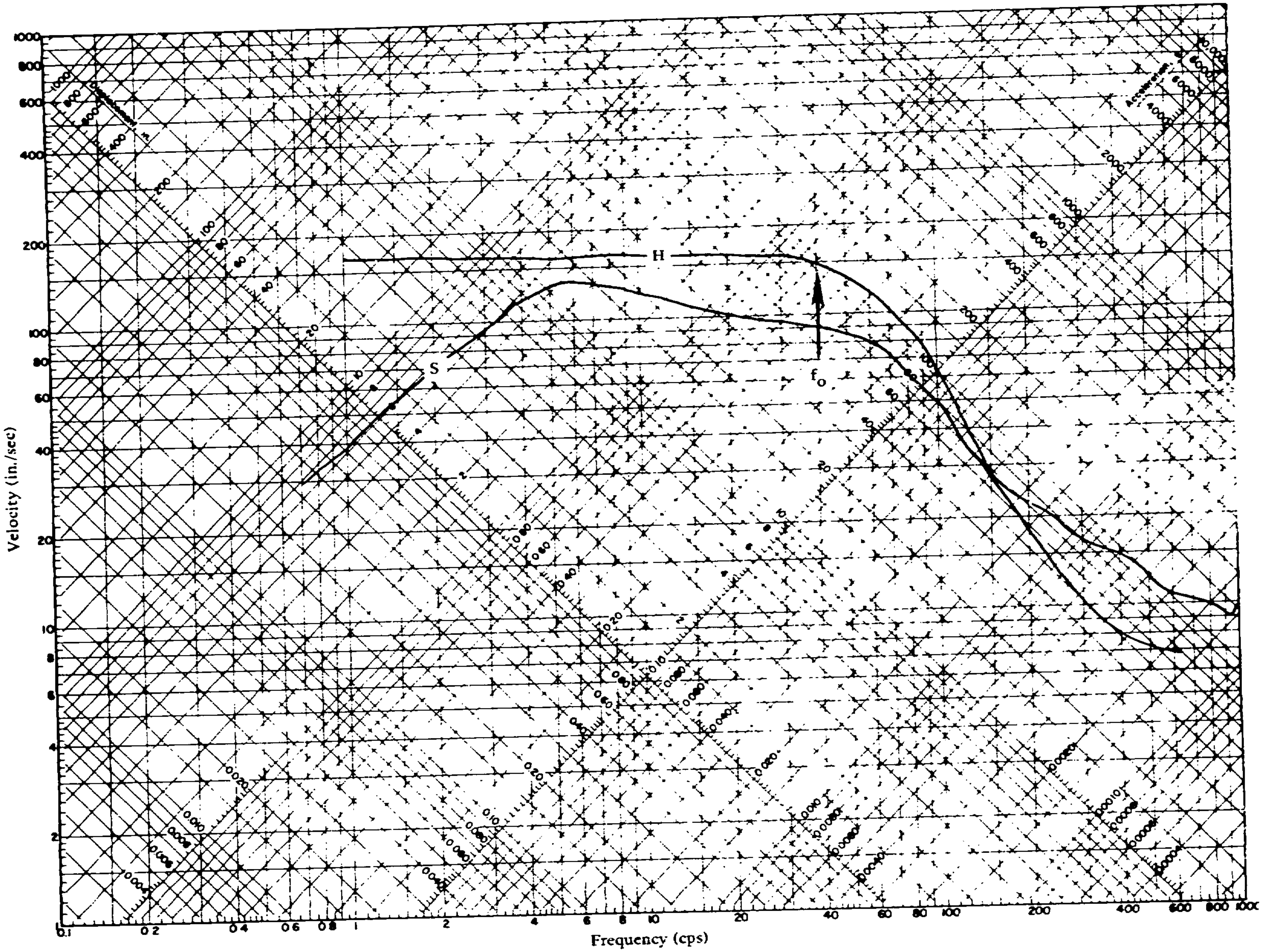



Figure 13. Shift of severity curve by light oscillator.

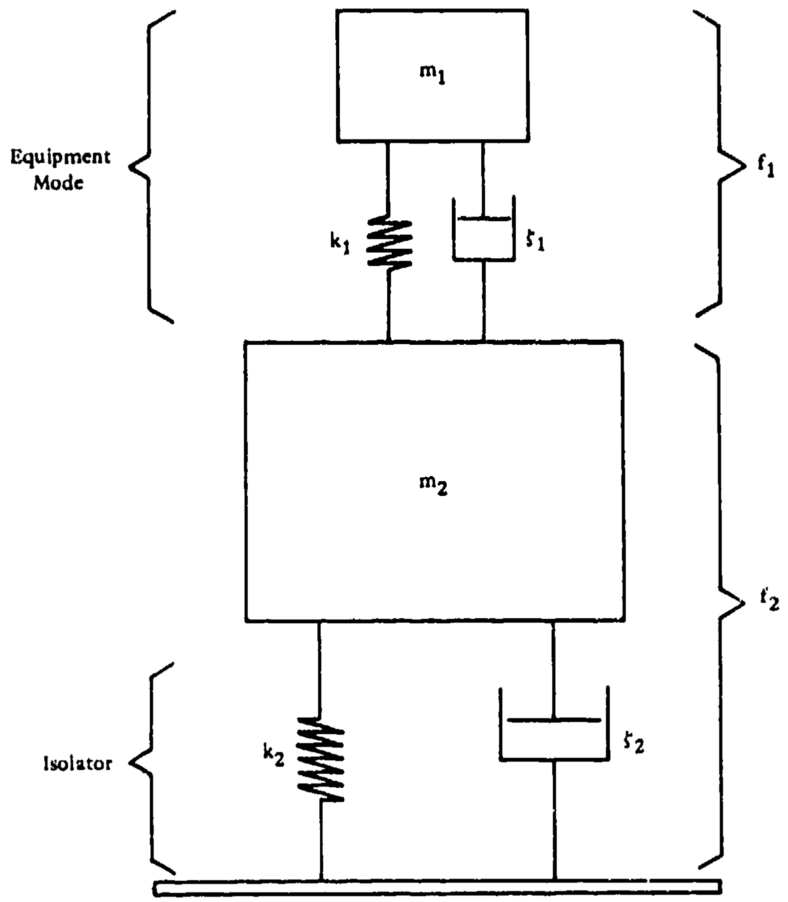



Figure 14. Heavy isolator of frequency $f_2$ at equipment frequency $f_1$.

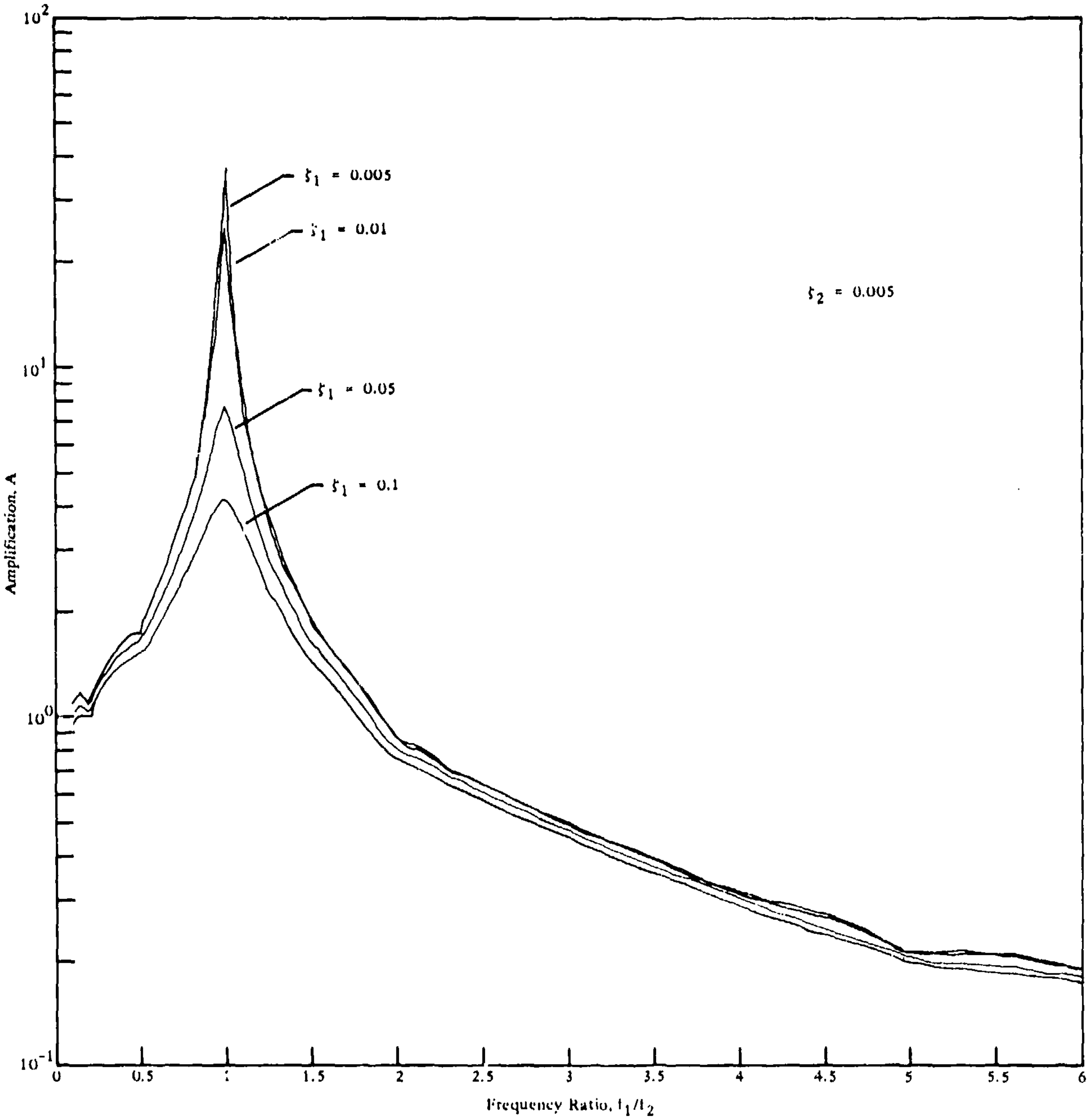



Figure 15. Amplification factors for isolator with 1/2% damping.

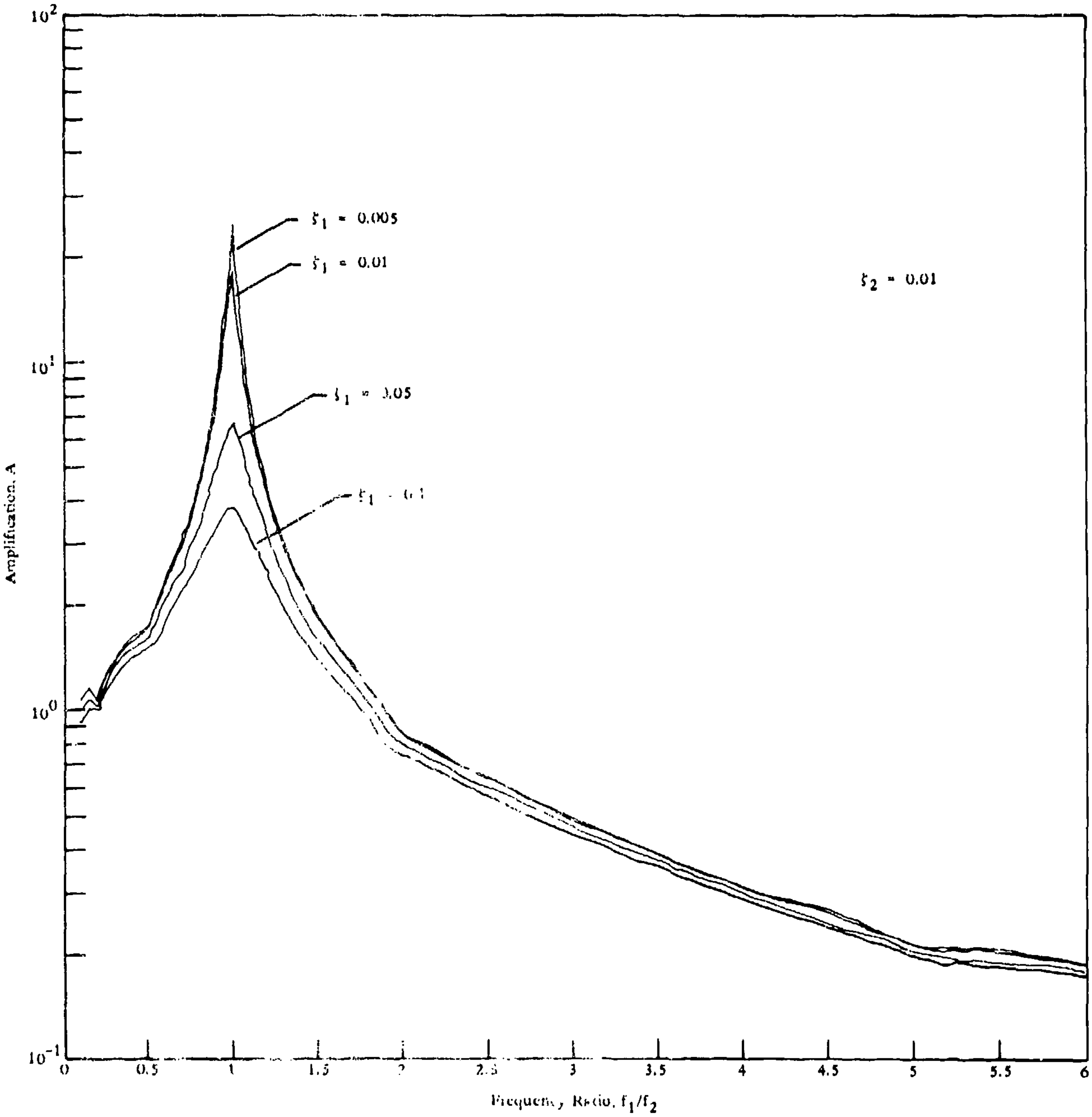



Figure 16. Amplification factors for isolator with 1% damping.

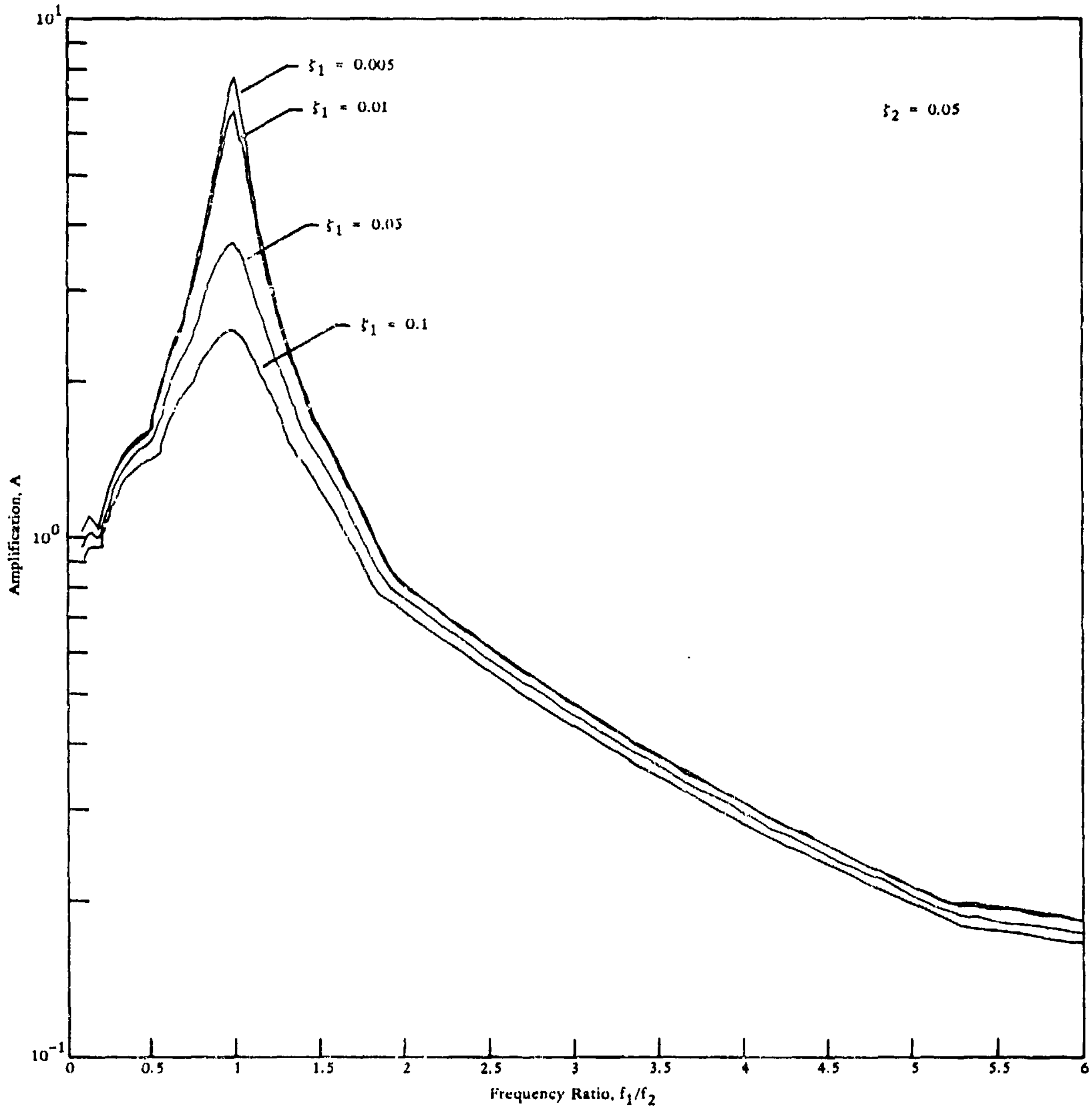



Figure 17. Amplification factors for isolator with 5% damping.

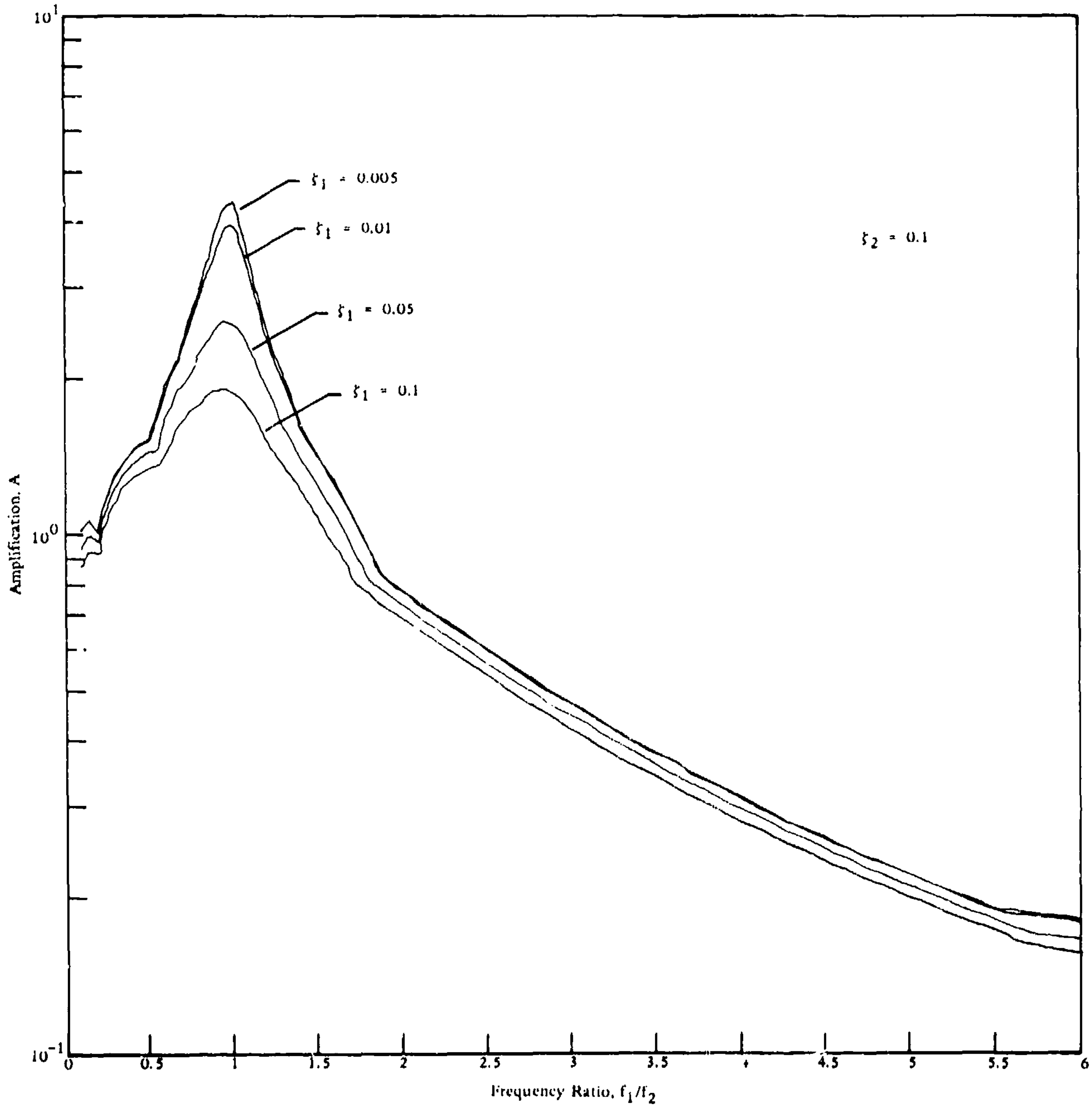



Figure 18. Amplification factors for isolator with 10% damping.

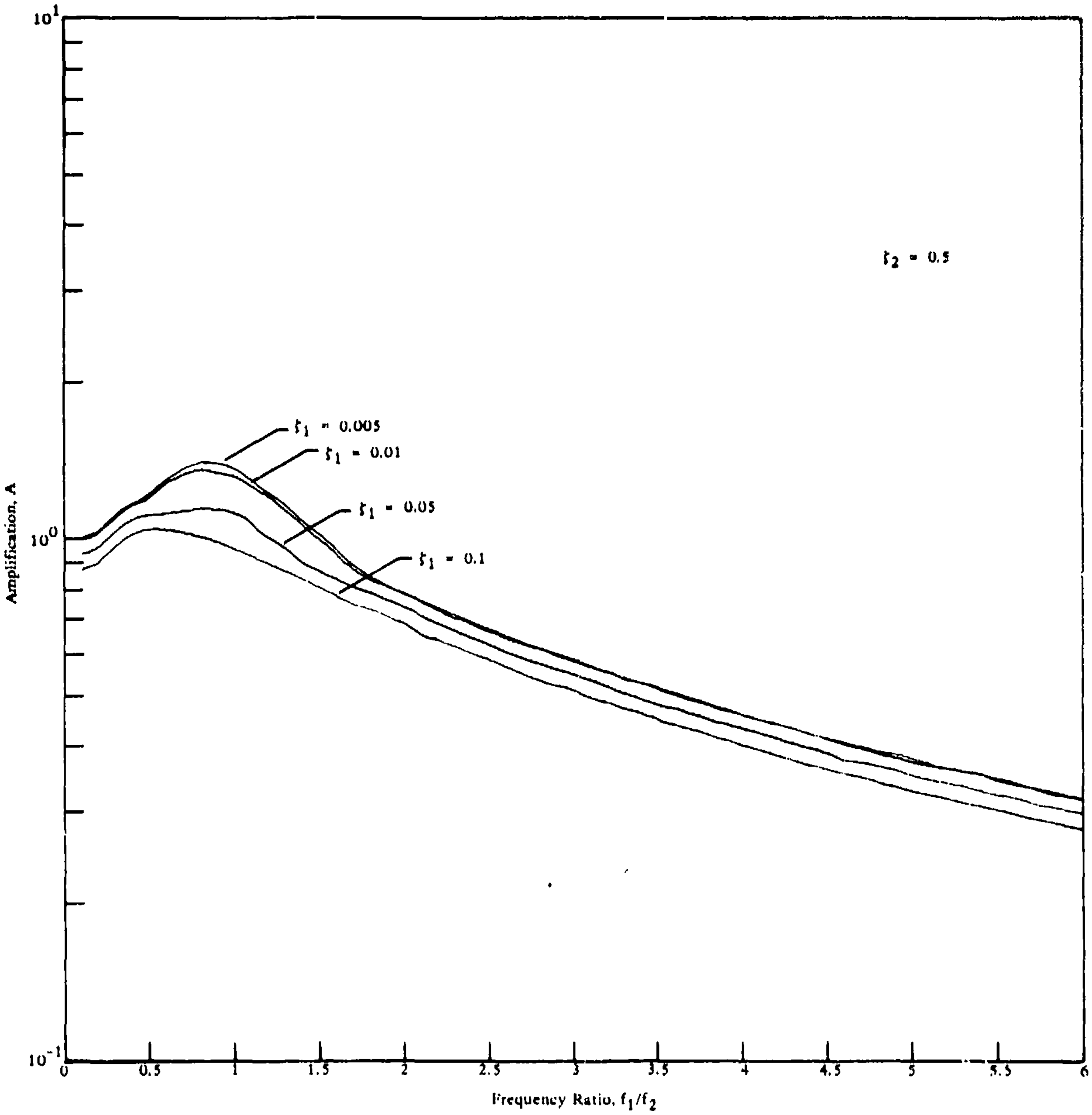



Figure 19. Amplification factors for isolator with 50% damping.

## Appendix A

## NATURAL FREQUENCIES AND MODES OF VIBRATION

In the body of the text the equipment was modeled to consist of a number of mass-spring elements of different frequencies that were mounted on a rigid foundation plate. This model, which might seem to be oversimplified, is actually a quite accurate representation of the frequency-dependent behavior of any piece of equipment for which damping is negligible. A piece of equipment will have some discrete components, such as relay reeds, that can be modeled exactly by a simple mass-spring element. In addition, all of the remaining structures that have distributed mass and distributed elasticity will vibrate at an infinite number of discrete frequencies. At very high frequencies, the displacements will be quite small, and the energy content of the vibrations will be negligible, so that it is quite valid to represent any linear reacting material with small energy dissipation by means of a finite number of discrete frequencies.

In general, these discrete frequencies are not the consequence of an easily observable single mass and single spring. They result instead from the fact that at various frequencies a continuous component will find that the least amount of energy is required if some of its parts move together or move in opposition of each other. Hence, at a particular frequency, a beam-like component may vibrate so that the middle will move up at the sane time the ends move down (and vice versa). Such configurations are easily frozen at the lower frequencies by a strobe light and are called "mode shapes" for the structure. Each frequency of natural vibration of the structure has associated with it a load shape, and these shapes are such that any physically attainable static or dynamic shape of the deformed structure can be represented by adding together the various natural mode shapes with particular amplitudes. A brief jolt or jar, which is transmitted to the equipment, may excite some or all of the modes. The fact that the displacement at a particular point is the sum of the displacements at that point of the various individual vibration modes would be quite disturbing were it not for two important facts. First, the modal displacements may well be of opposite signs, and superposition will actually tend to reduce the total deflection; perhaps more important is the observation that the inertia forces associated with a particular mode do not perform work when displaced through the configuration of a natural mode of different frequency. A concomitant observation is that all equipment has a lowest natural frequency and that excitations at frequencies below that contribute little damaging energy to the structure.

Structures with significant damping do not, in general, have natural modes of vibration in the real domain. Important exceptions occur in assemblages of lumped mass-spring elements when (1) there is a damping element in parallel with every spring element and the dashpot constants are proportional to the corresponding spring stiffnesses or (2) there is a damping element connecting every mass to the base and dashpot constants are proportional to the corresponding masses. When damping is slight, one is usually justified in approximating the system behavior by assuming that one or the other of these conditions is satisfied. The errors involved in these assumptions are small and have negligible effects on the validity of the design procedure.

# Appendix B

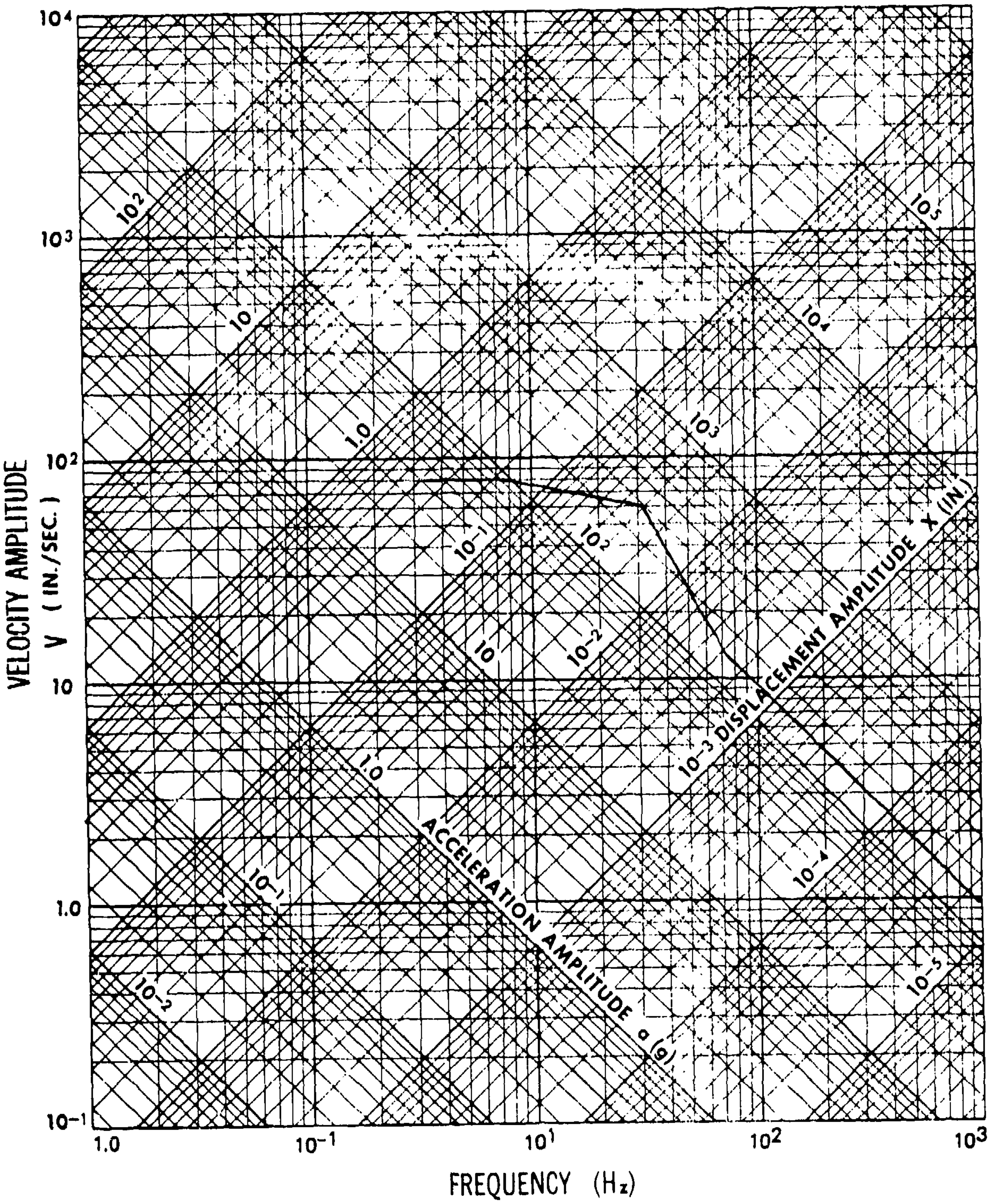
# EQUIPMENT HARDNESS CATALOG

## INDEX OF CATALOGED HARDNESSES

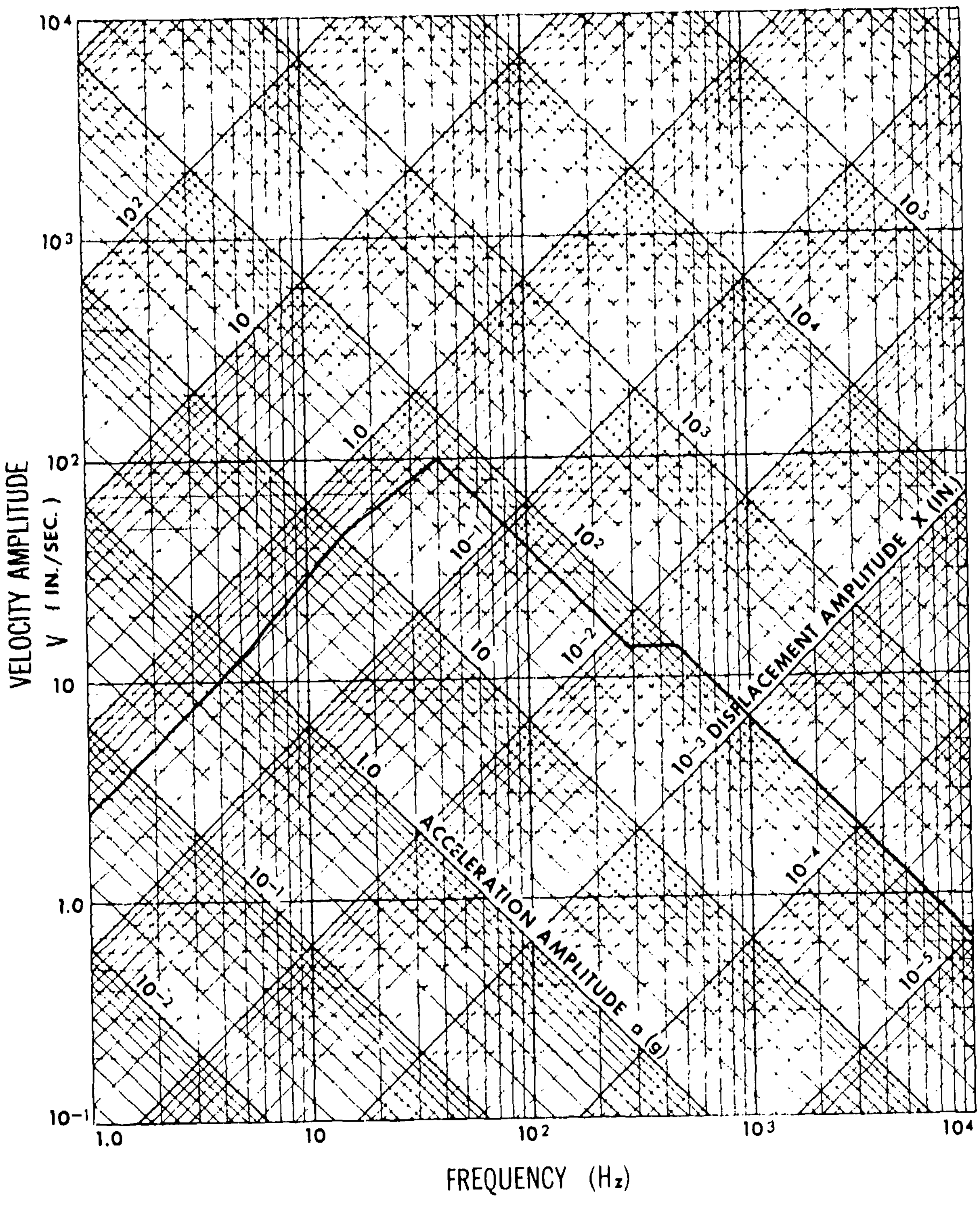
ITEM: Compressor, Reciprocating, Power Driven, Air
REFERENCE: B-1
DESCRIPTION: 250 psi, 300 cfm, two-stage, water cooled, V-belt motor driven, both cylinders double acting. Intercooler is shell and tube type, water cooled on tube side, air cooled on shell side.
MANUFACTURER: Chicago Pneumatic Tool Co.
STATISTICS: Weight: 14,055 lb; Size: 14 ft 6 in. long, 56 in. wide, 7 ft 8 in. high.
NOTE: A dynamic analysis was performed on this unit to verify that it could be hard-mounted to the building and survive the local in-structure shock environment. The attached spectra are the envelopes used in the analysis.

NAVFAC / NCEL

SHOCK DATA ANALYSIS

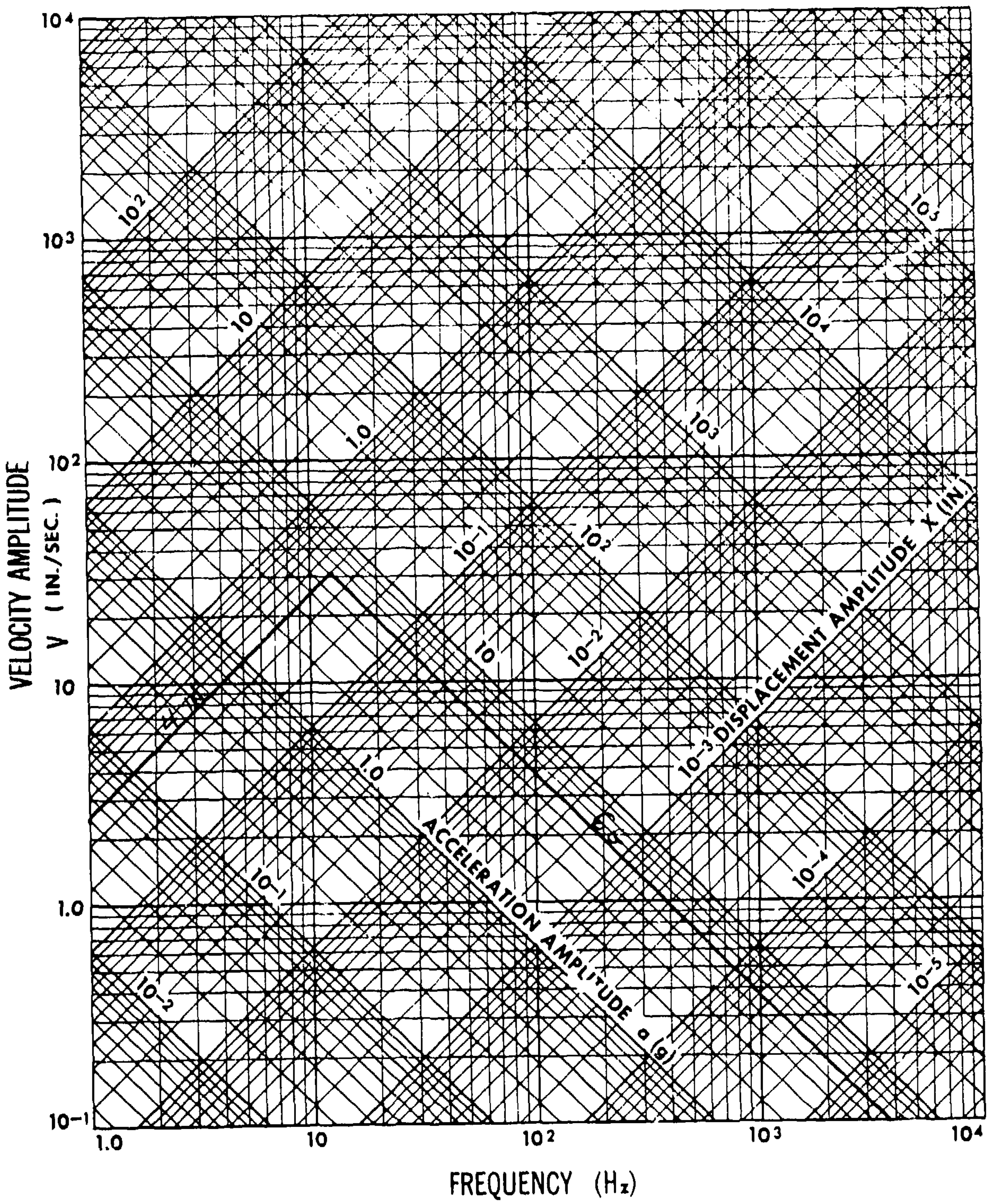



POICR
Vertical Envelope for Dynamic Analysis




P01CR
Vertical Envelope for Dynamic Analysis

ITEM: Generator, Diesel Engine
REFERENCE: B-2
DESCRIPTION: 4-35 hp ,360 rpm, 2,880 kVA, 13,800 volts, 16 cyl
MANUFACTURER: Cooper-Bessemer
NOTE: This unit was too large for actual testing and a dynamic analysis was performed which indicated that the unit could survive the attached spectra.

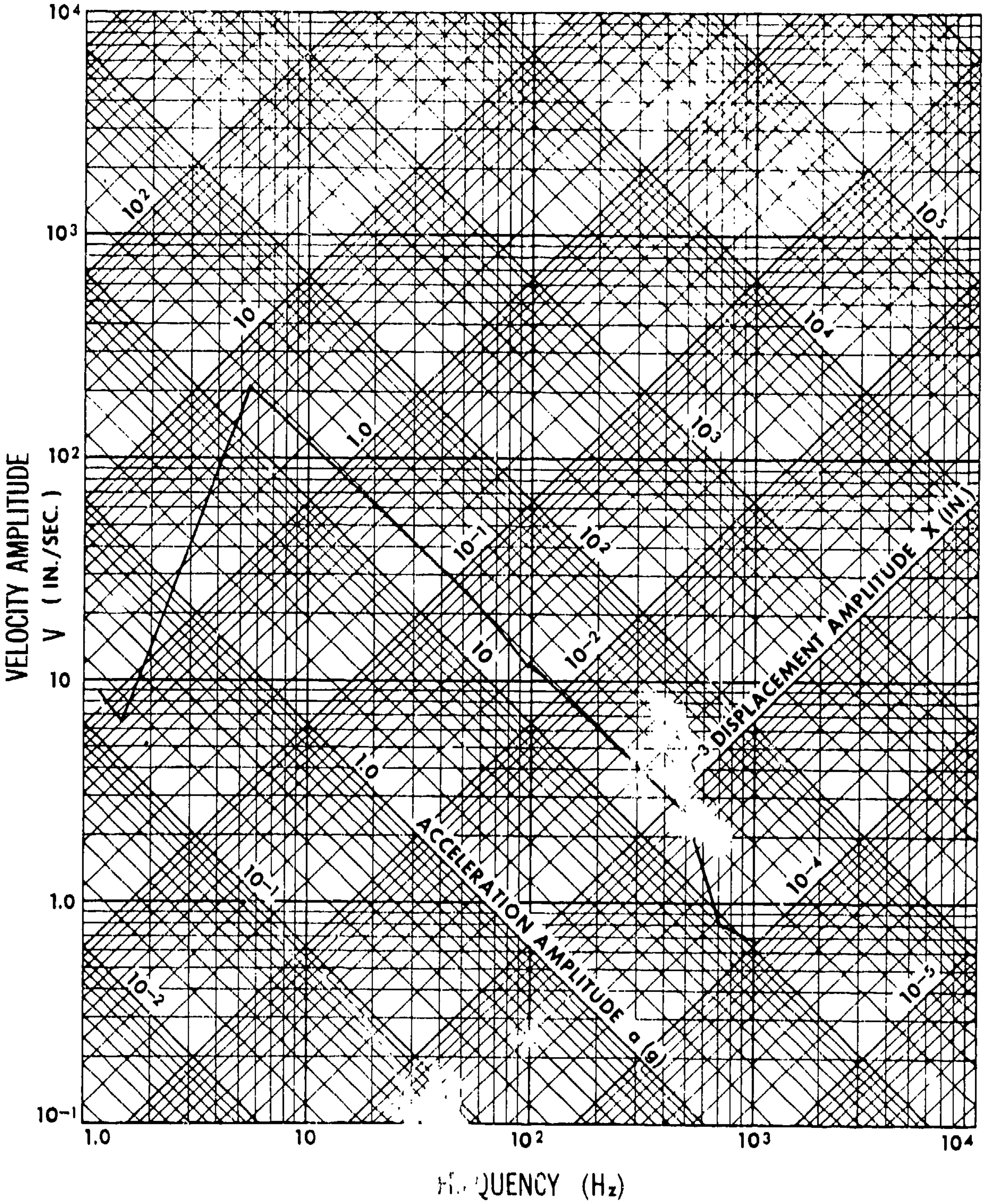



E01GD
Input Vertical and Horizontal Response Used for Dynamic Analysis

ITEM: Heat Sensing Device, Syphon Type
REFERENCE: B-3
DESCRIPTION: Used with model 93 electric controller
MANUFACTURER: Automatic Sprinkler Corp.
STATISTICS: Weight: 1 lb; Size: 6 x 3 x 3 in.
AXIS IDENTIFICATION: x - longitudinally; y - vertical (longest dimension); z - transverse

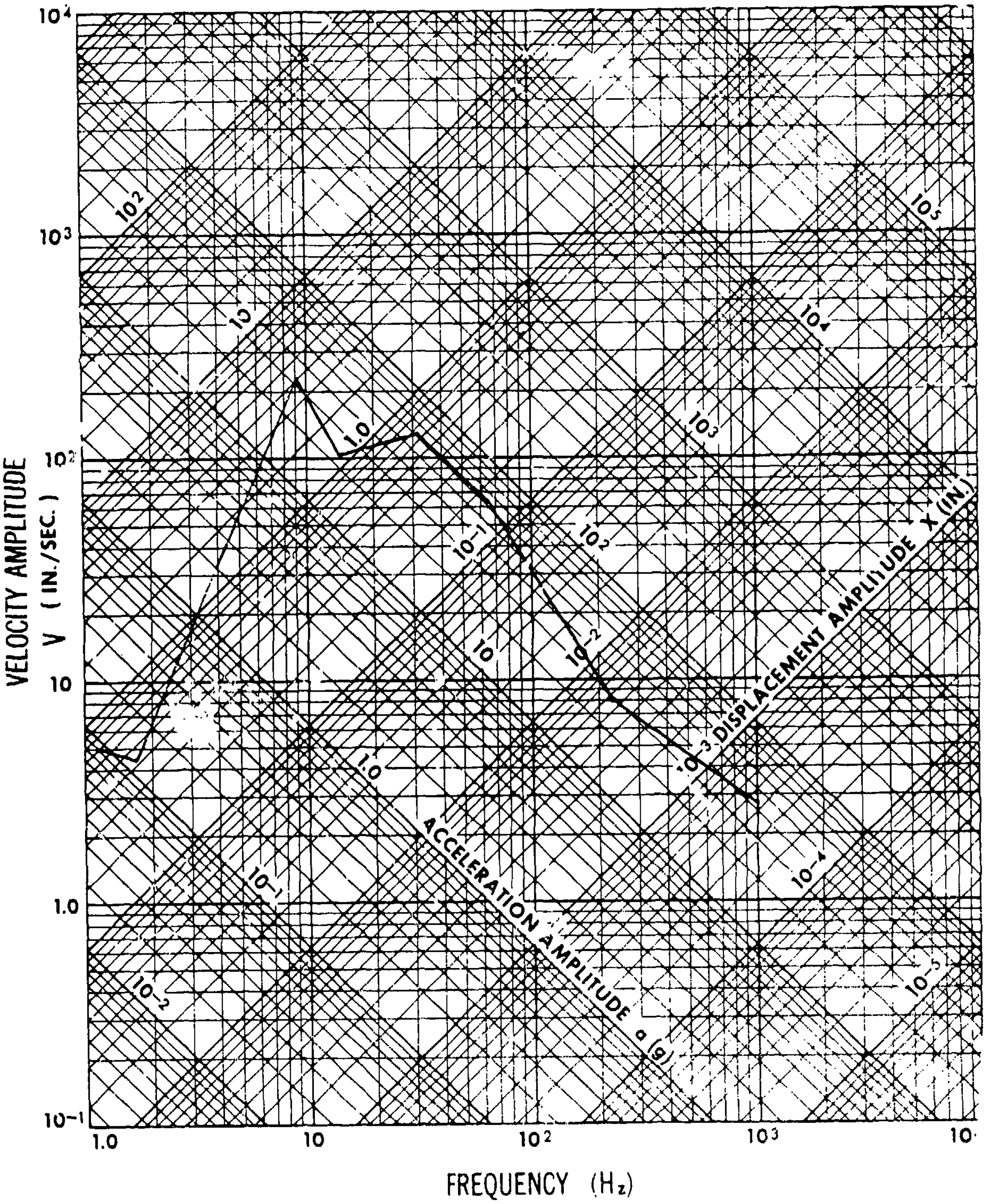
ITEM: Electric Controller
REFERENCE: B-3
DESCRIPTION: Used with syphon type heat sensing device
MANUFACTURER: Automatic Sprinkler Corp.
STATISTICS: Weight: 16 lb; Size: 6 x 6 x 18.5 in.
AXIS IDENTIFICATION: x - longitudinally; y - vertically (longest dimension); z - transverse

## NAVFAC / NCEL
## SHOCK DATA ANALYSIS



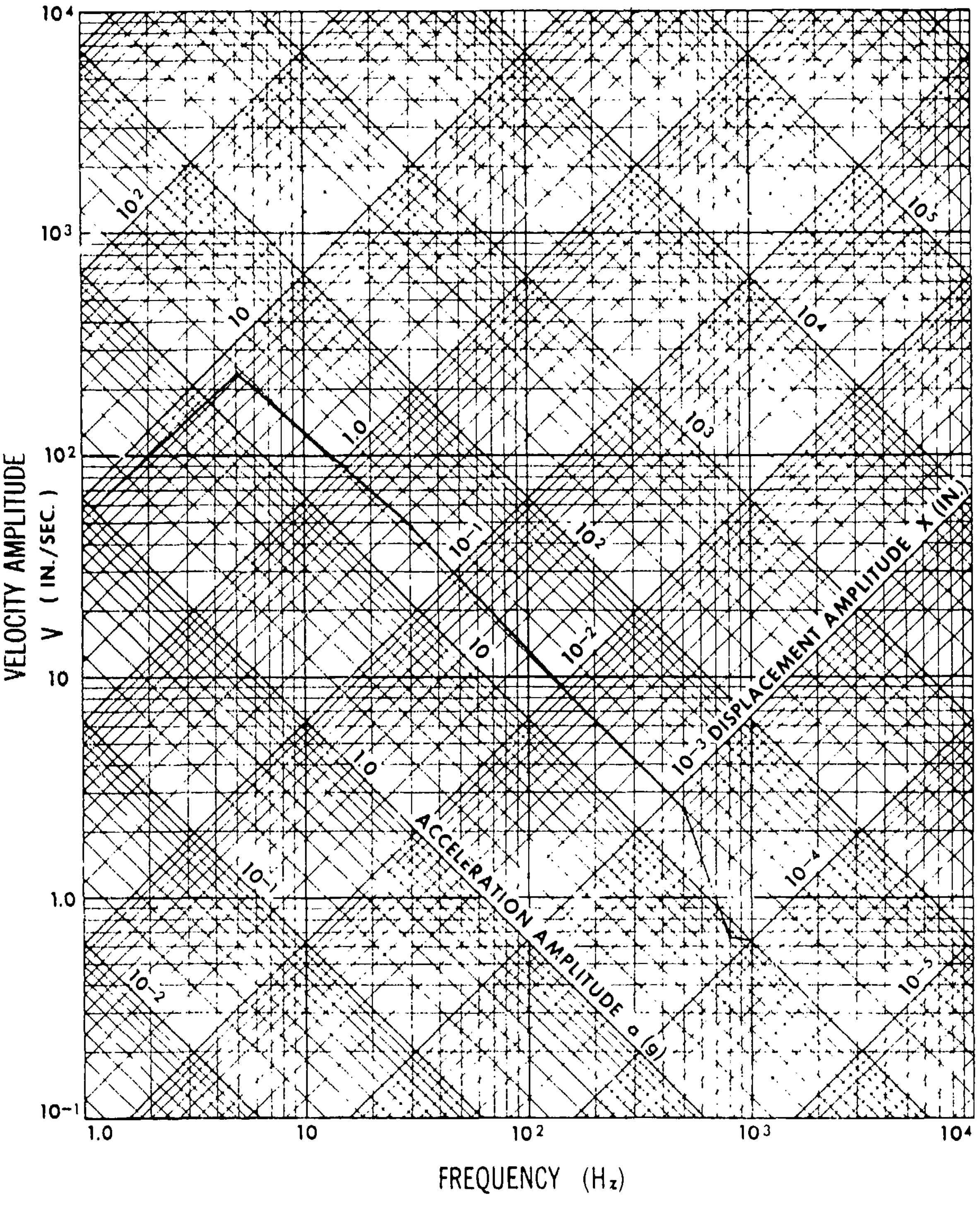

Heat Sensing Device with Controller
X-Axis

NAVFAC / NCEL
SHOCK DATA ANALYSIS




Heat Sensing Device with Controller
Y-Axis

# NAVFAC / NCEL
## SHOCK DATA ANALYSIS



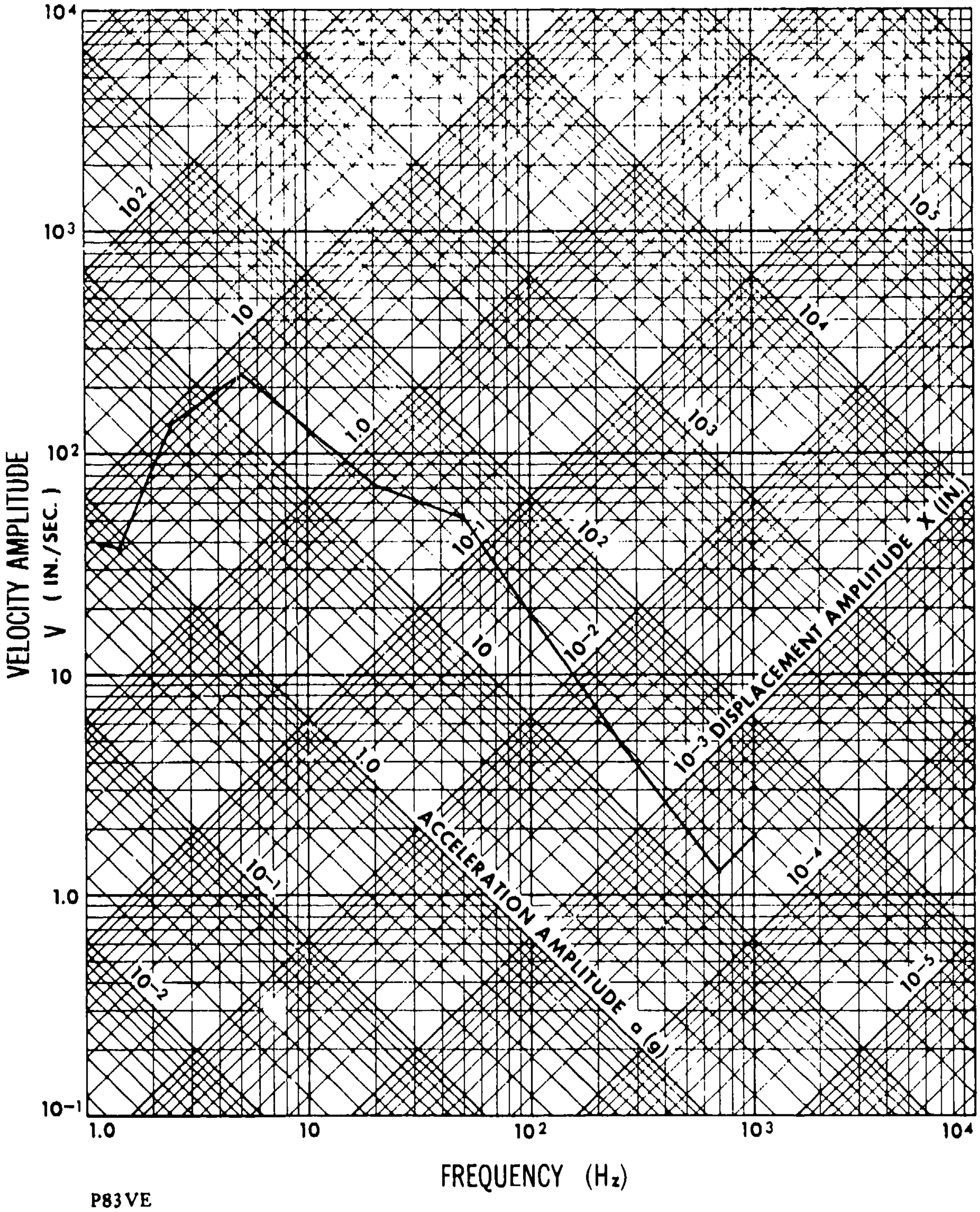

Heat Sensing Device with Controller
Z-Axis

ITEM: Valve, Pressure Control
REFERENCE: B-4
DESCRIPTION: 75-100 psi
MANUFACTURER: Jordon Valve Co.
P/N: Serial No. J01921
STATISTICS: Weight: 300 lb; Size: 18.5 x 12.5 x 24 in.
AXIS IDENTIFICATION: x - longitudinal (along longest dimension);
y - vertical; z - transverse

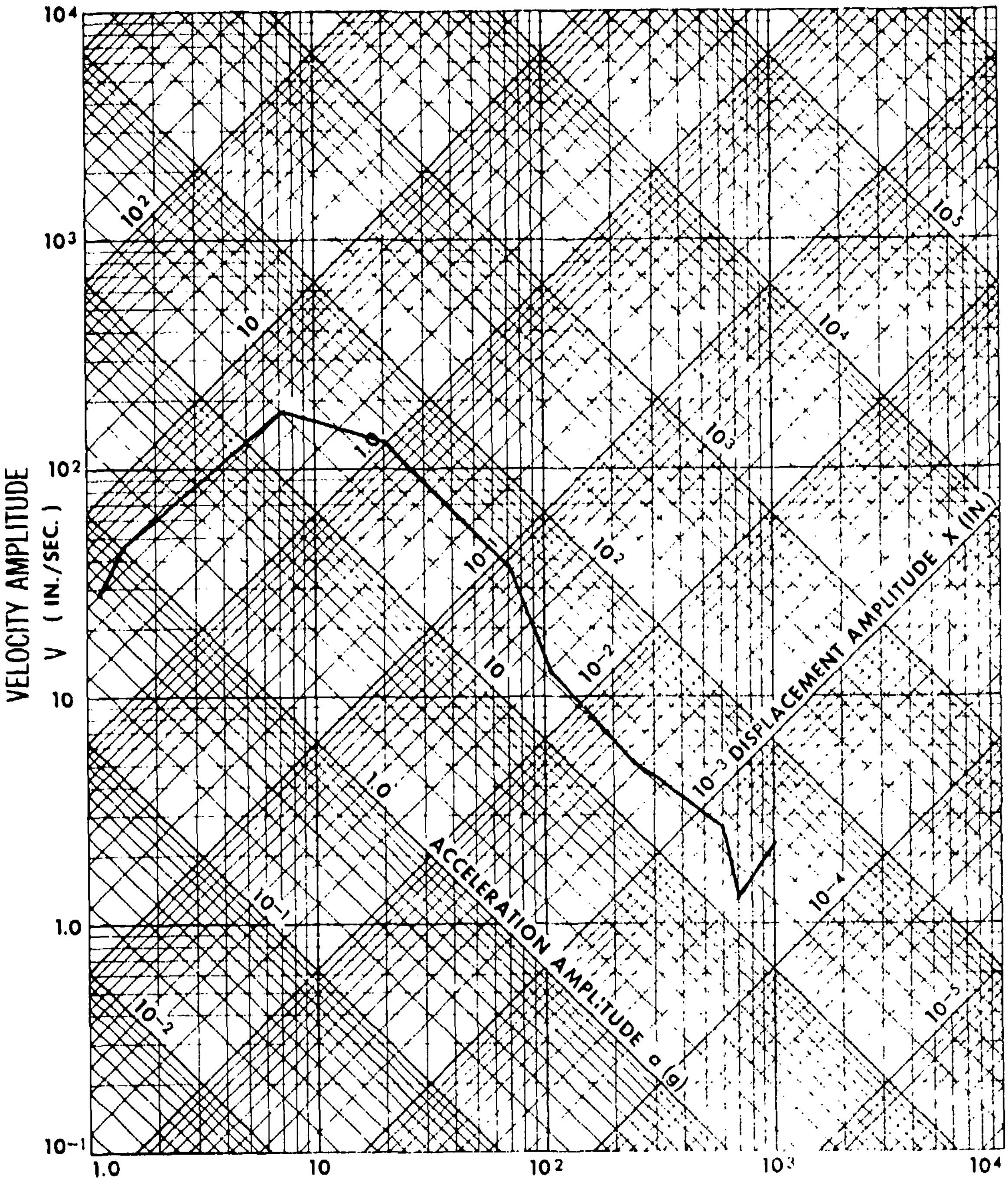
NOTE: No structural degradation during or after testing. Some deviations from the regulated pressure occurred, but not enough to affect functional performance.

NAVFAC / NCEL

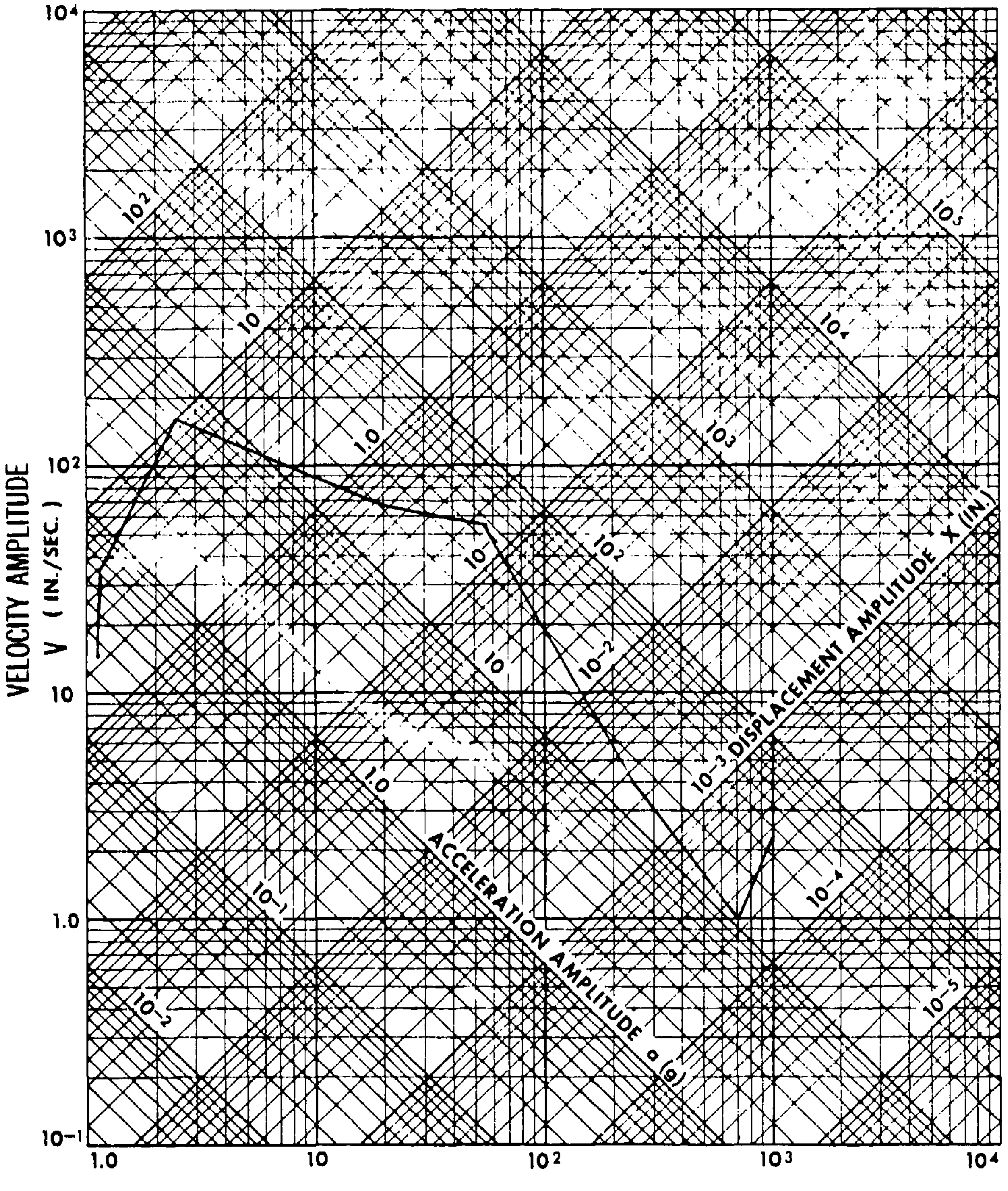
SHOCK DATA ANALYSIS




P83VE
X-Axis

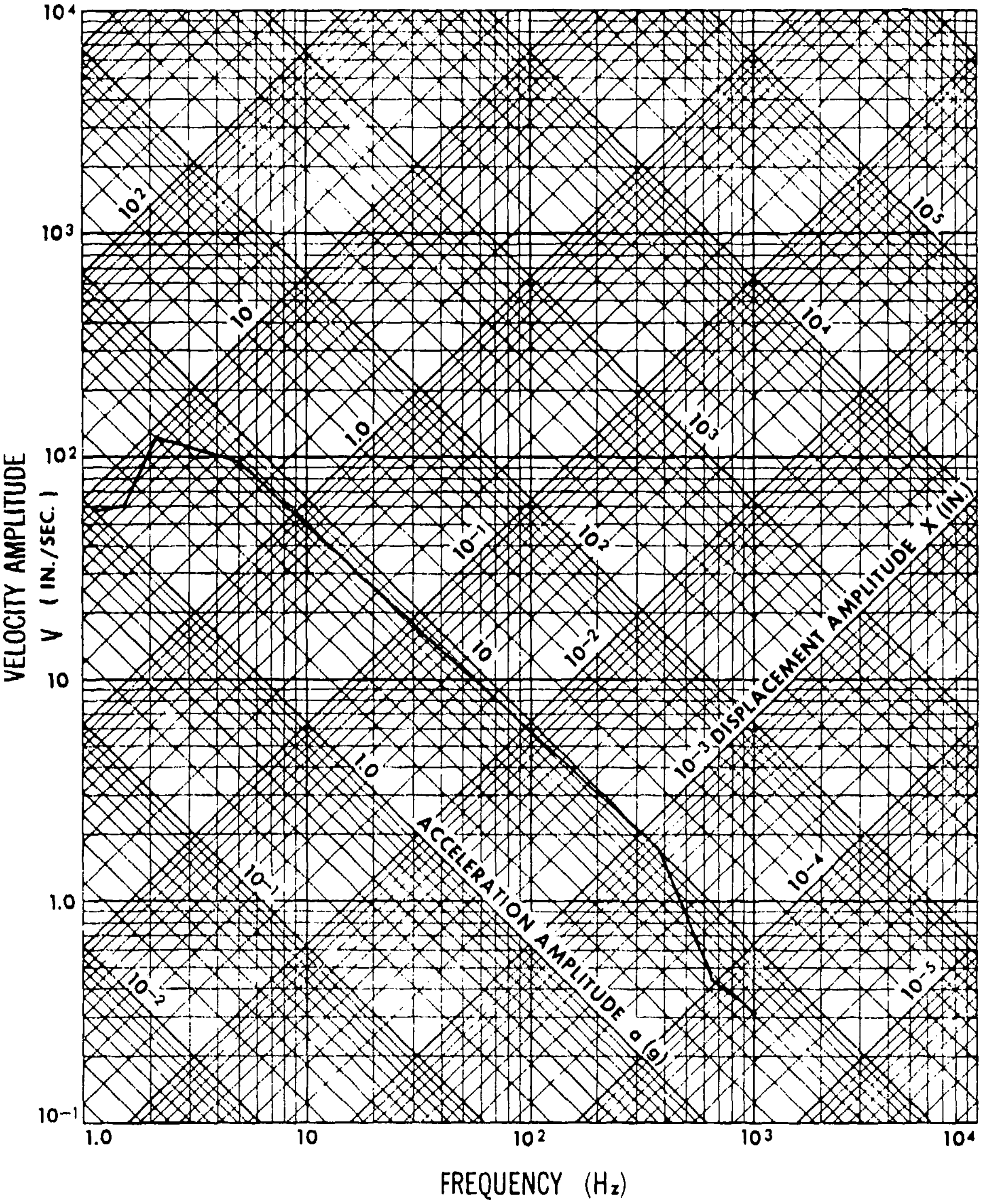

VELOCITY AMPLITUDE
V (IN./SEC.)
FREQUENCY (Hz)
ACCELERATION AMPLITUDE a (g)
DISPLACEMENT AMPLITUDE X (IN.)
10^4
10^3
10^2
10
1.0
10^-1
1.0
10
10^2
10^3
10^4

P83VE
Y-Axis


10⁴
10³
10²
10
1.0
10⁻¹
VELOCITY AMPLITUDE
V (IN./SEC.)
1.0
10
10²
10³
10⁴
FREQUENCY (Hz)
ACCELERATION AMPLITUDE a (g)
DISPLACEMENT AMPLITUDE X (IN.)

P83VE
Z-Axis

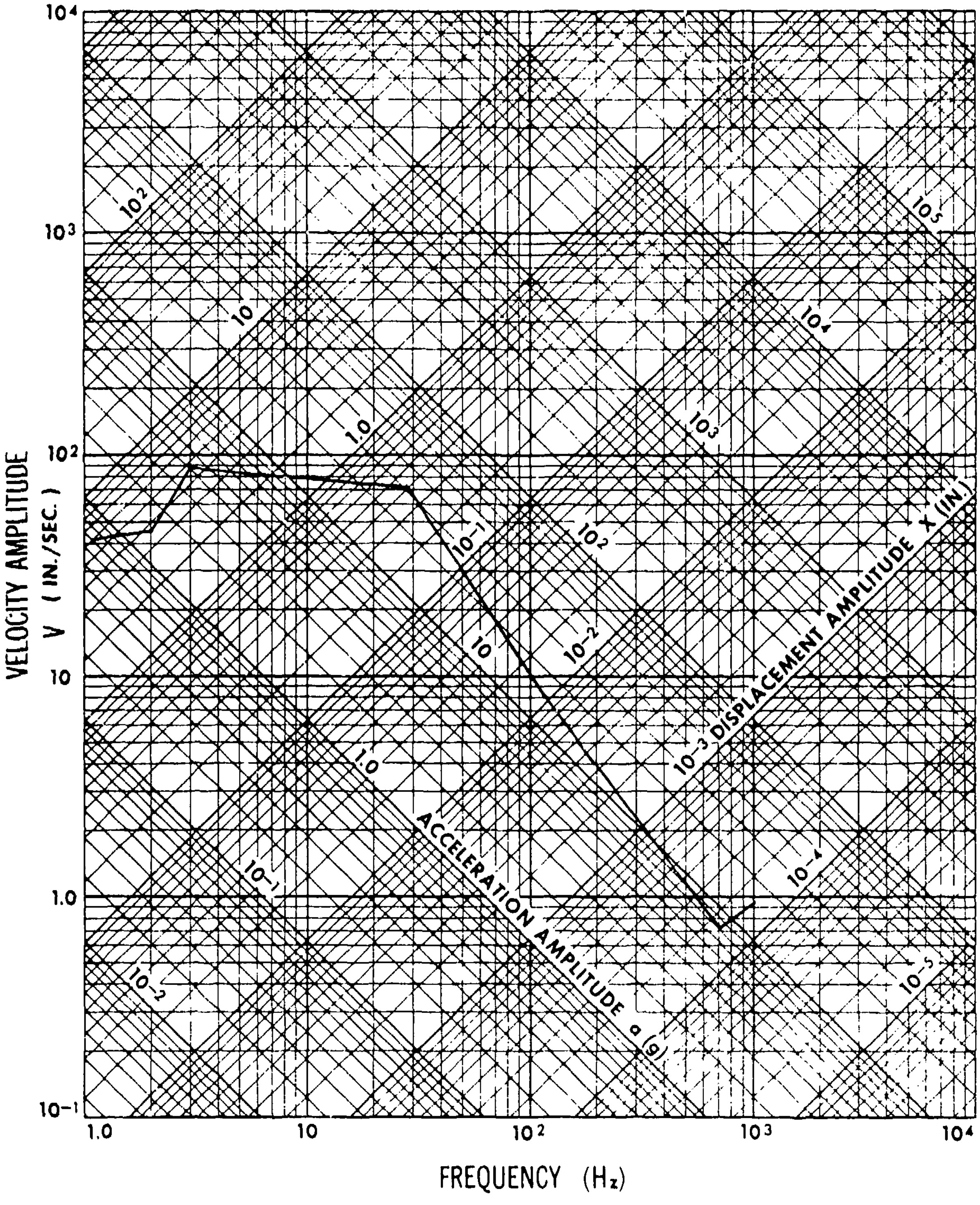
ITEM: Switch, Oil Shutdown, Air Compressor Control
REFERENCE: B-5
DESCRIPTION: for Air Compressor P01CR
MANUFACTURER: Chicago Pneumatic Tool Co.
P/N: 013905
STATISTICS: Weight: 10 lb; Size: 9 x 8 x 11 in.
AXIS IDENTIFICATION: x - longitudinal (perpendicular to mounting bracket); y - vertical; z - transverse
NOTE: No structural damage resulted. Momentary chatter on N.C. switch contacts occurred on the horizontal shocks.




P01CR Switch
X-Axis

NAVFAC / NCEL

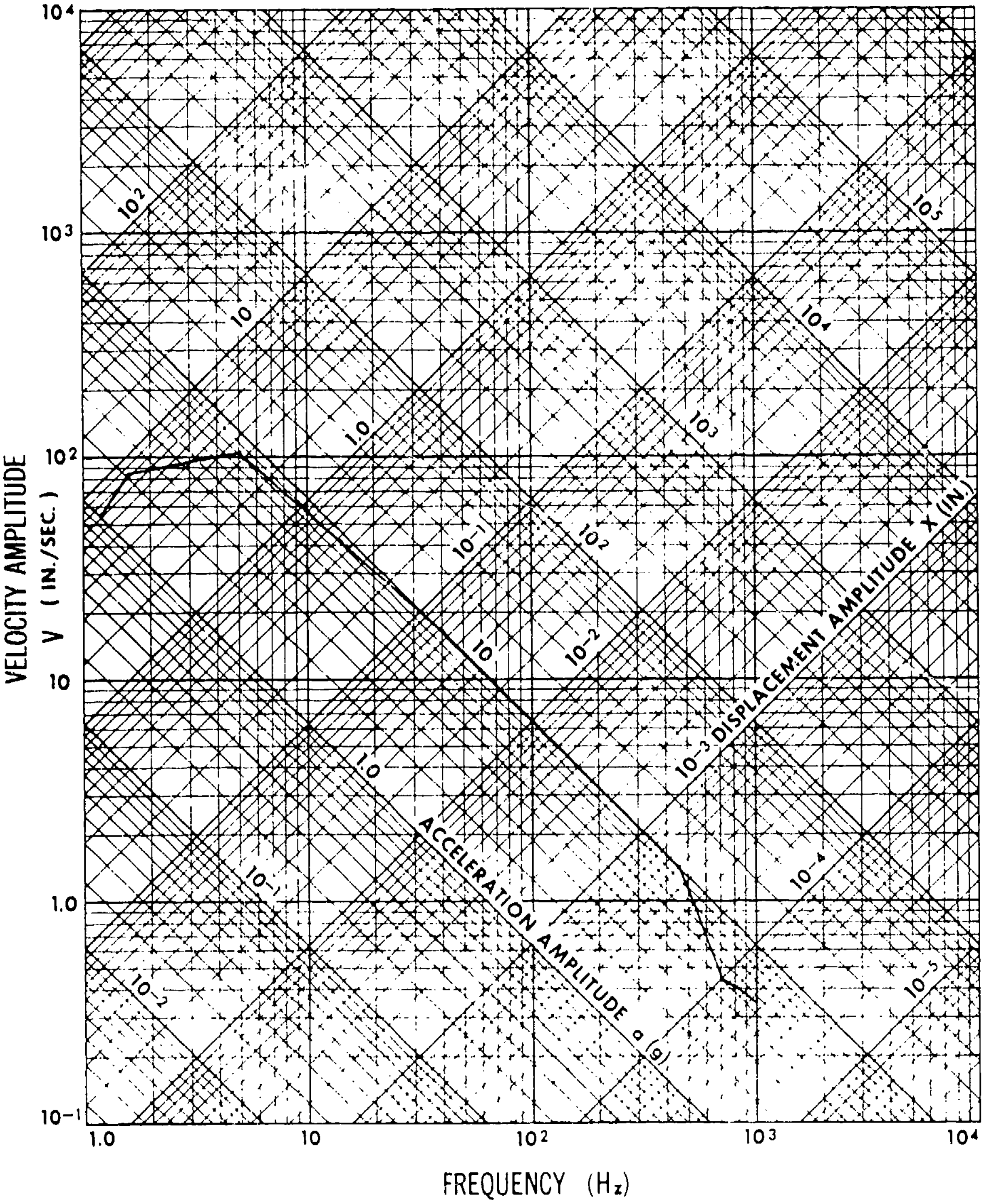
SHOCK DATA ANALYSIS




P01CR Switch
Y-Axis

NAVFAC / NCEL

SHOCK DATA ANALYSIS

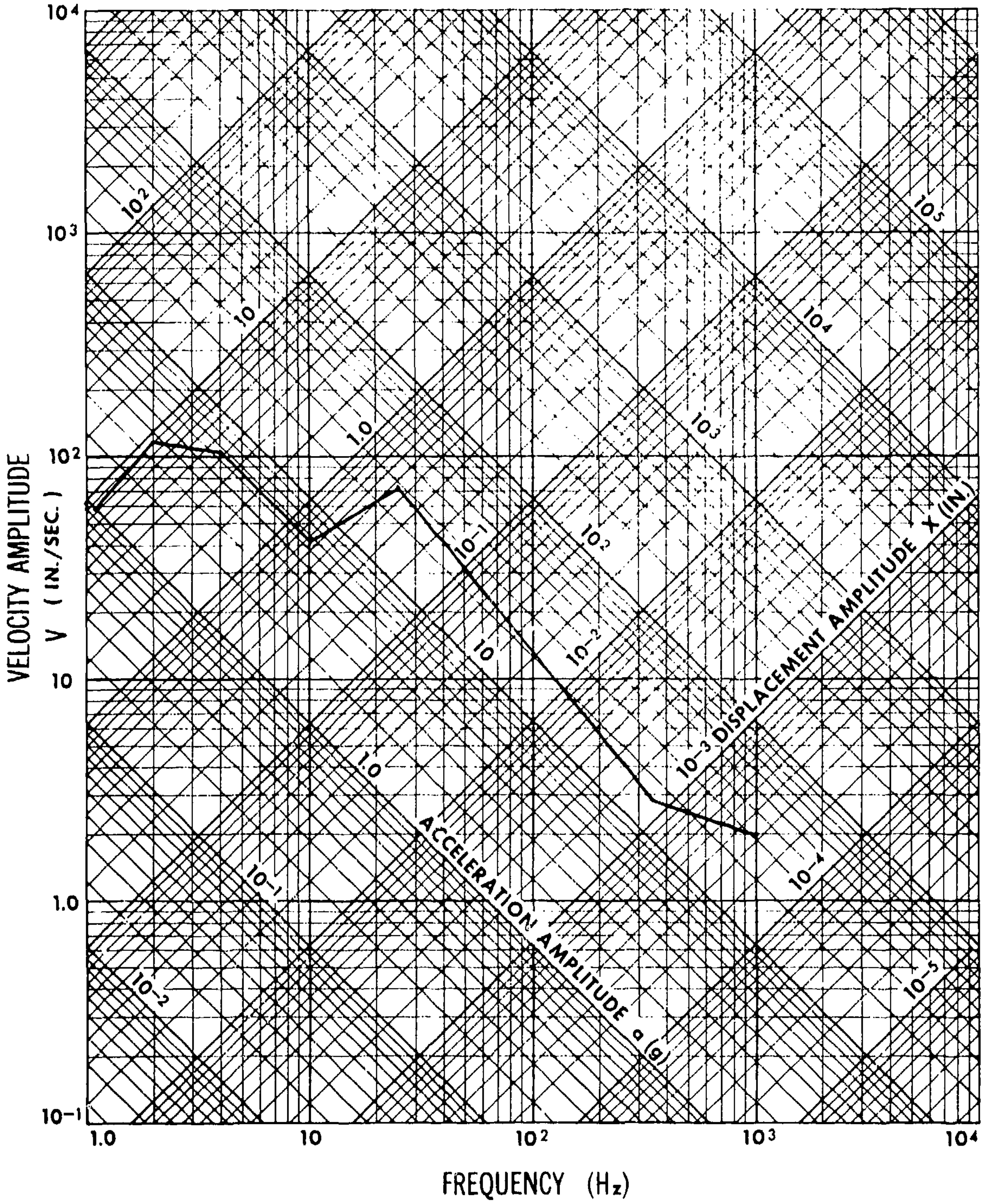



P01CR Switch
Z-Axis

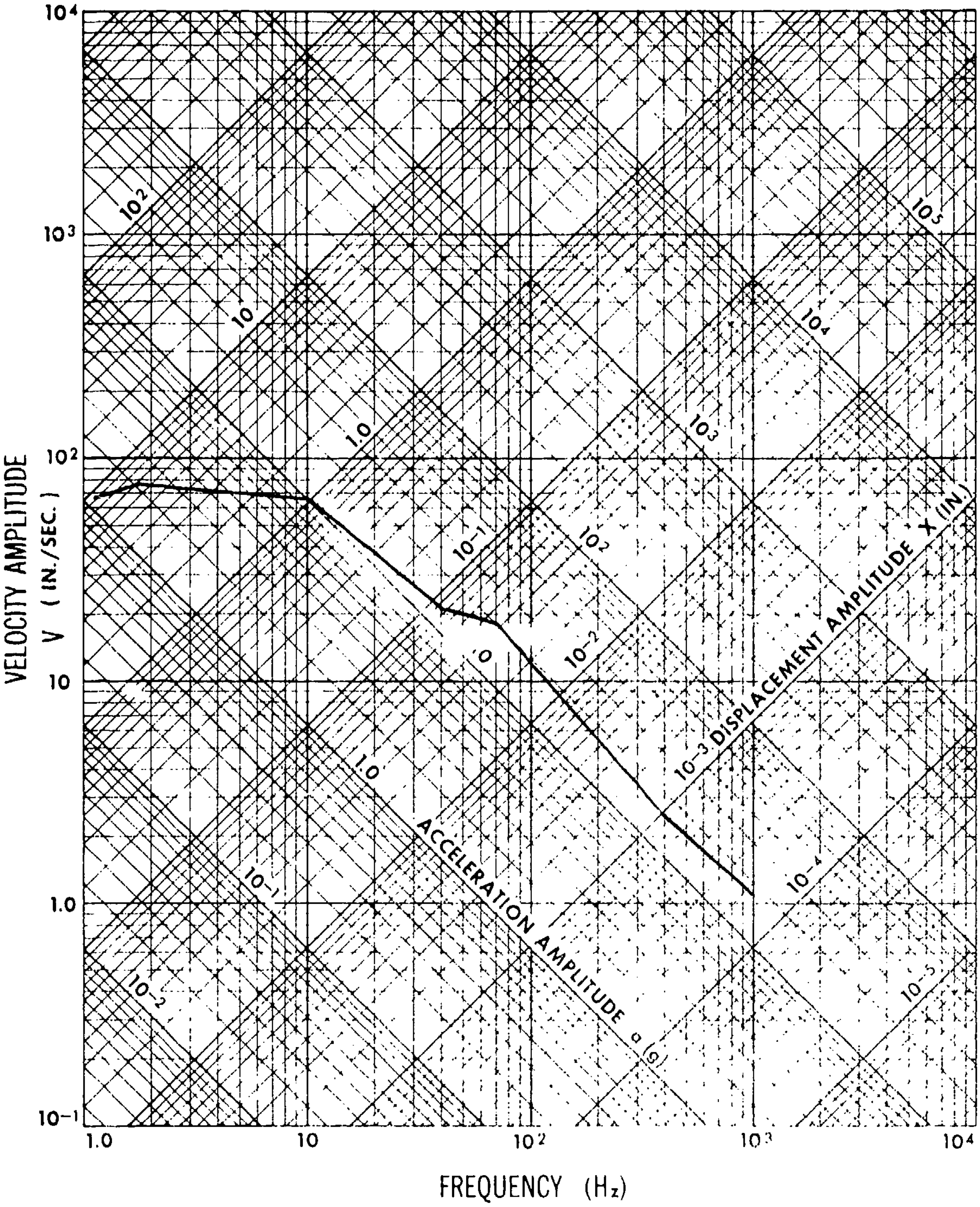
ITEM: Generator, Static Exciter Regulator
REFERENCE: B-6
DESCRIPTION: Specimen includes 5 kV switchgear cabinet which contains a circuitbreaker, potential transformer, current transformer, auxiliary protective relays, etc.
MANUFACTURER: General Electric Co.
P/N: 352020BE/02AL
STATISTICS: Weight: 880 lb; Size: 60 x 42 x 90 in.
AXIS IDENTIFICATION: x - transverse; y - vertical; z - longitudinal (along the longest dimension)
NOTE: No structural damage occurred during testing. Chatter in the protective auxiliary relays was experienced during the highest level of testing but was not considered degrading to the performance.

NAVFAC / NCEL

SHOCK DATA ANALYSIS



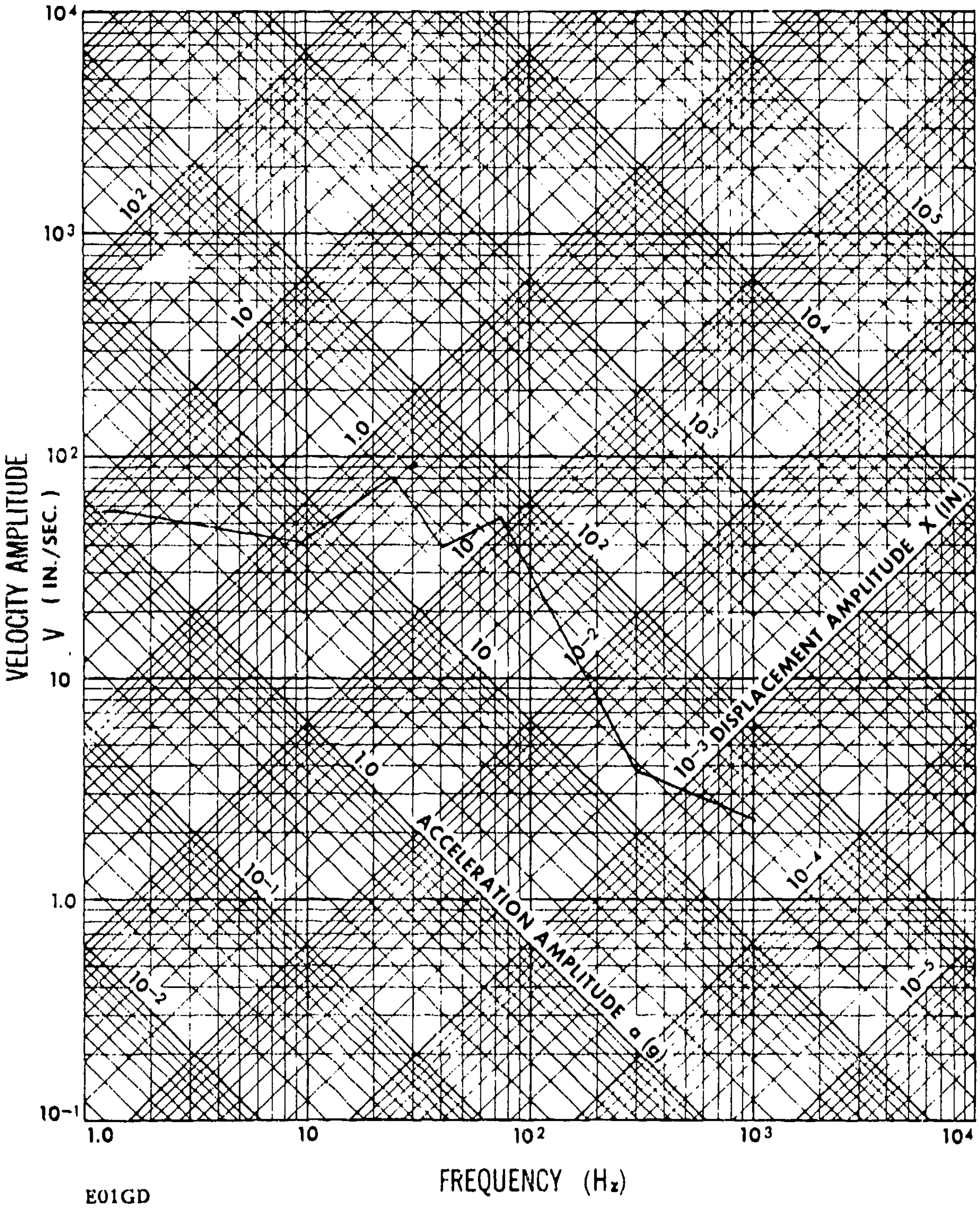

E01GD
X-Axis


VELOCITY AMPLITUDE
V (IN./SEC.)
FREQUENCY (Hz)
ACCELERATION AMPLITUDE a (g)
DISPLACEMENT AMPLITUDE X (IN.)

E01GD
Y-Axis

NAVFAC / NCEL

SHOCK DATA ANALYSIS

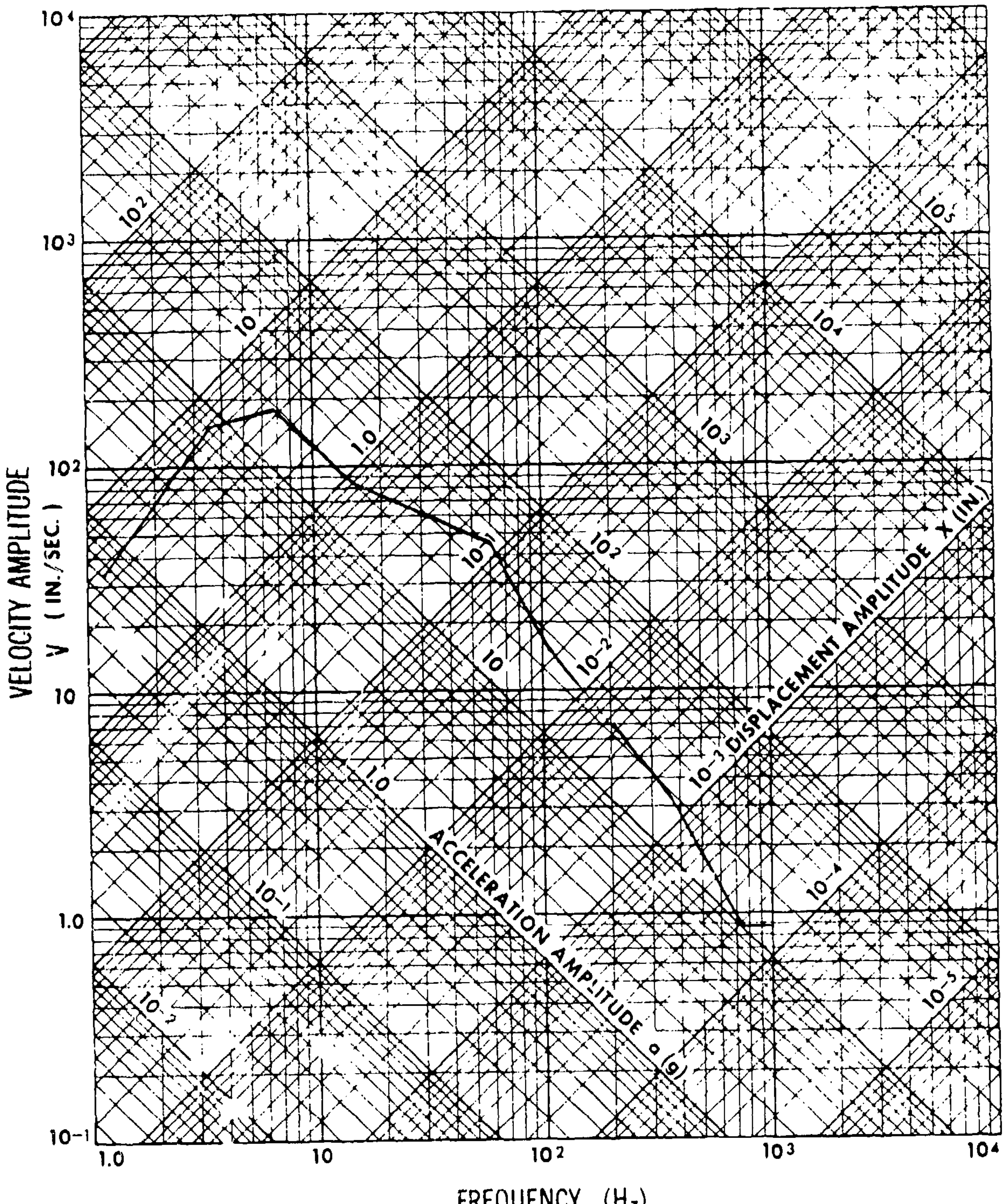



E01GD
Z-Axis

ITEM: Switch, Temperature
REFERENCE: B-7
DESCRIPTION: 250°F, 0.2 ampere
MANUFACTURER: United Electric Controls Co.
P/N: Model 2BSB, Type 302PD
STATISTICS: Weight: 3 lb; Size: 13 x 6-1/2 x 6-1/2 in.
AXIS IDENTIFICATION: x - longitudinal (longest dimension); y - vertical;
z - transverse

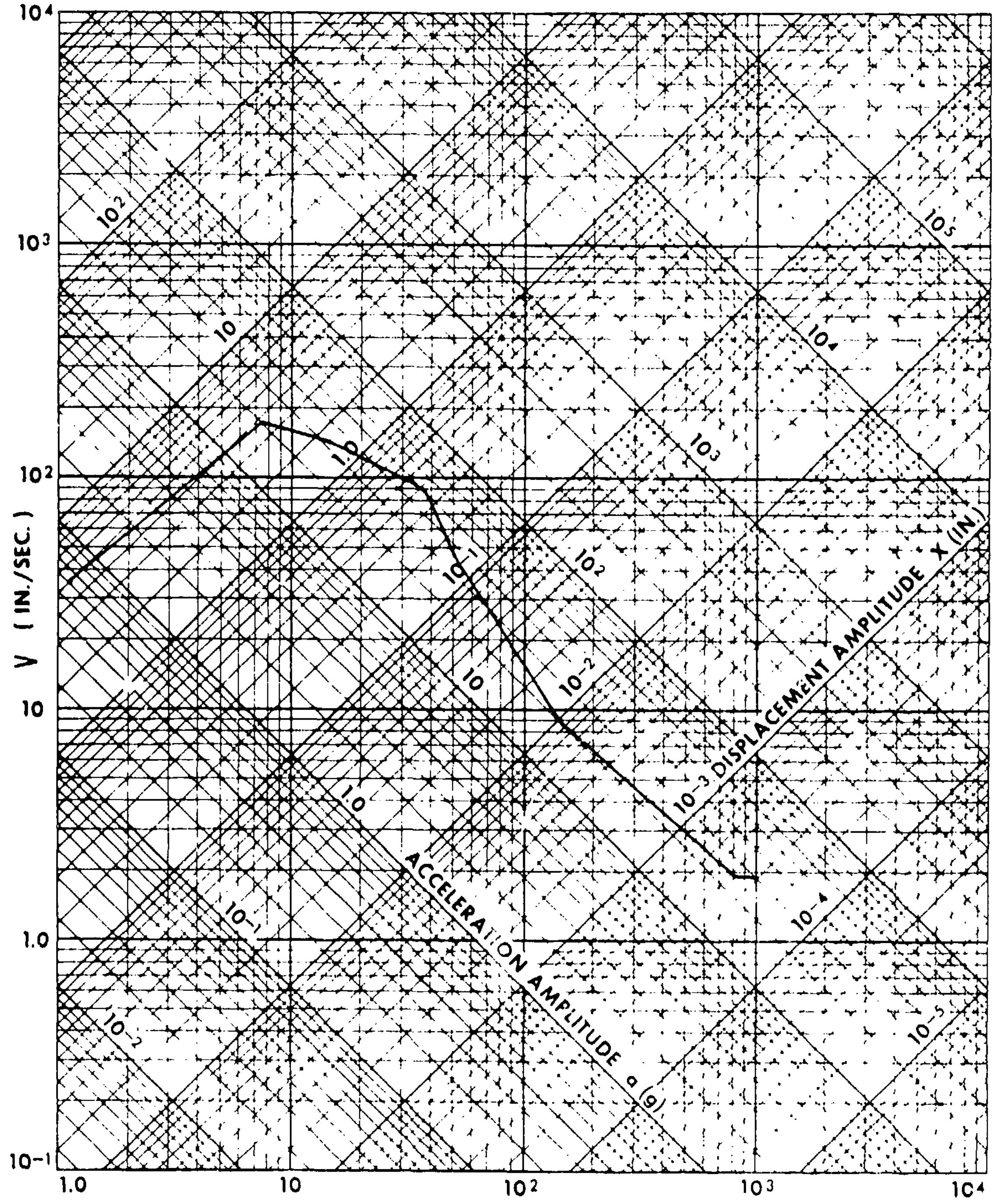
NAVFAC / NCEL

SHOCK DATA ANALYSIS




158TS
X-Axis

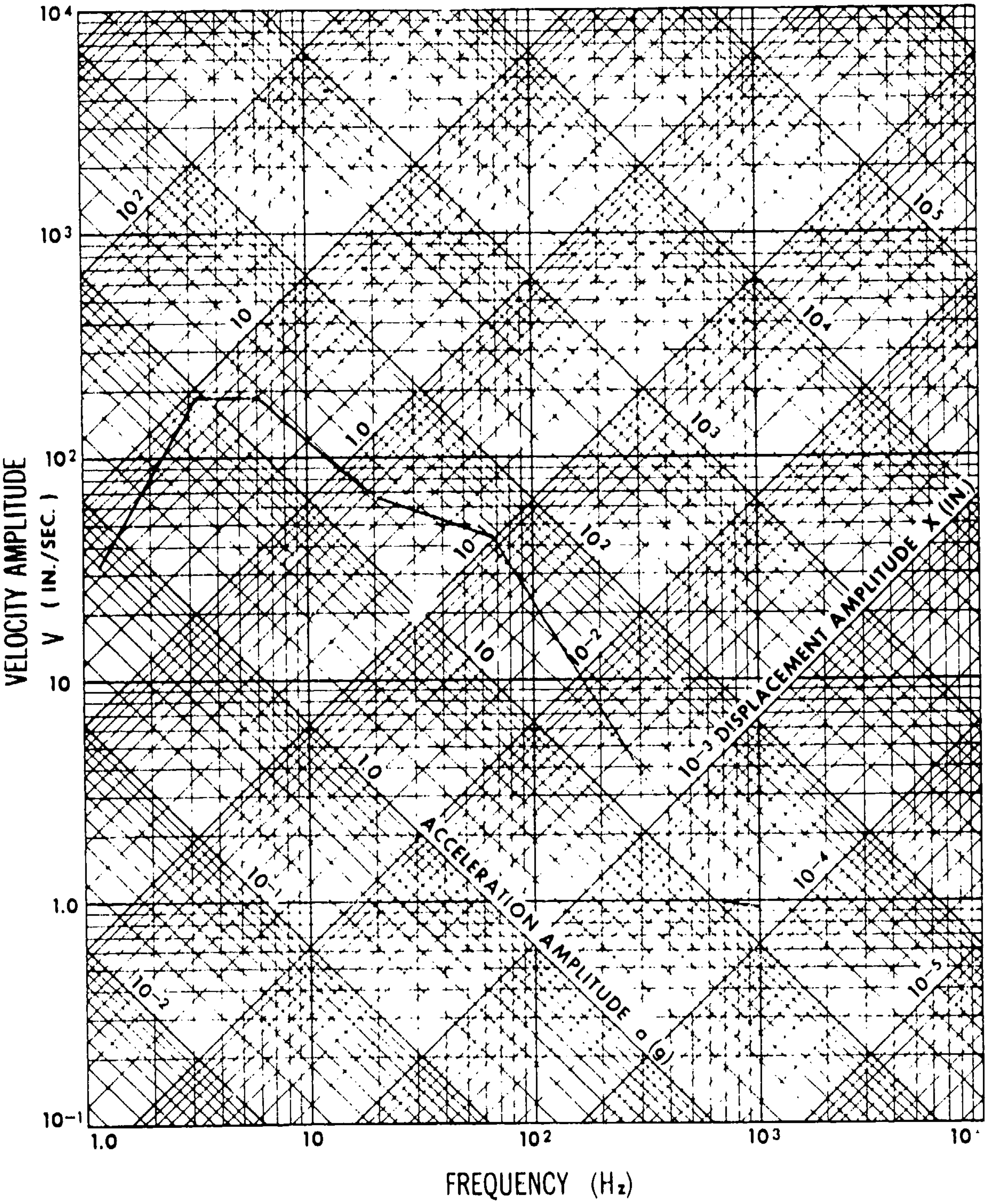
# NAVFAC / NCEL
# SHOCK DATA ANALYSIS




158TS
Y-Axis

NAVFAC / NCEL

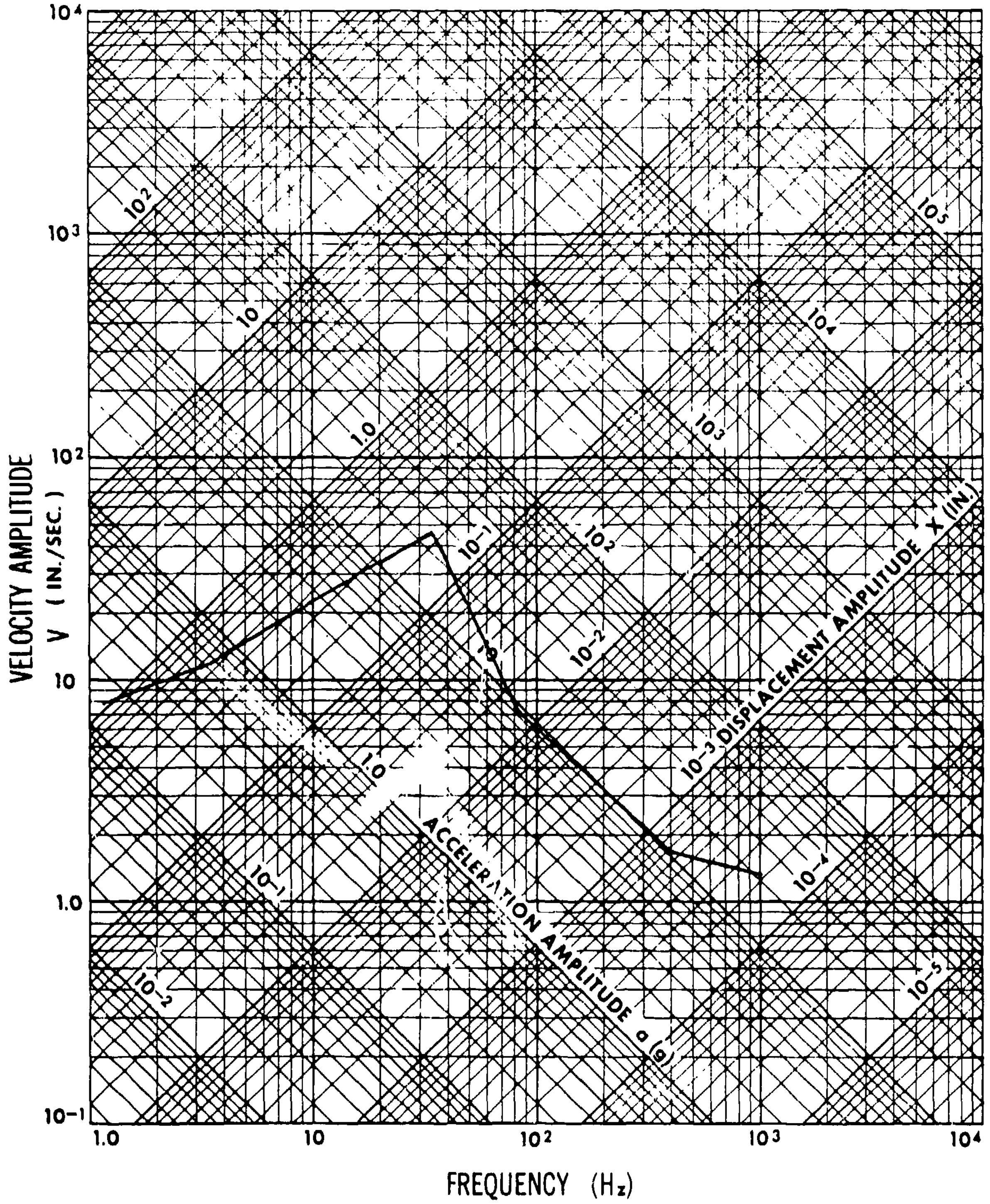
SHOCK DATA ANALYSIS



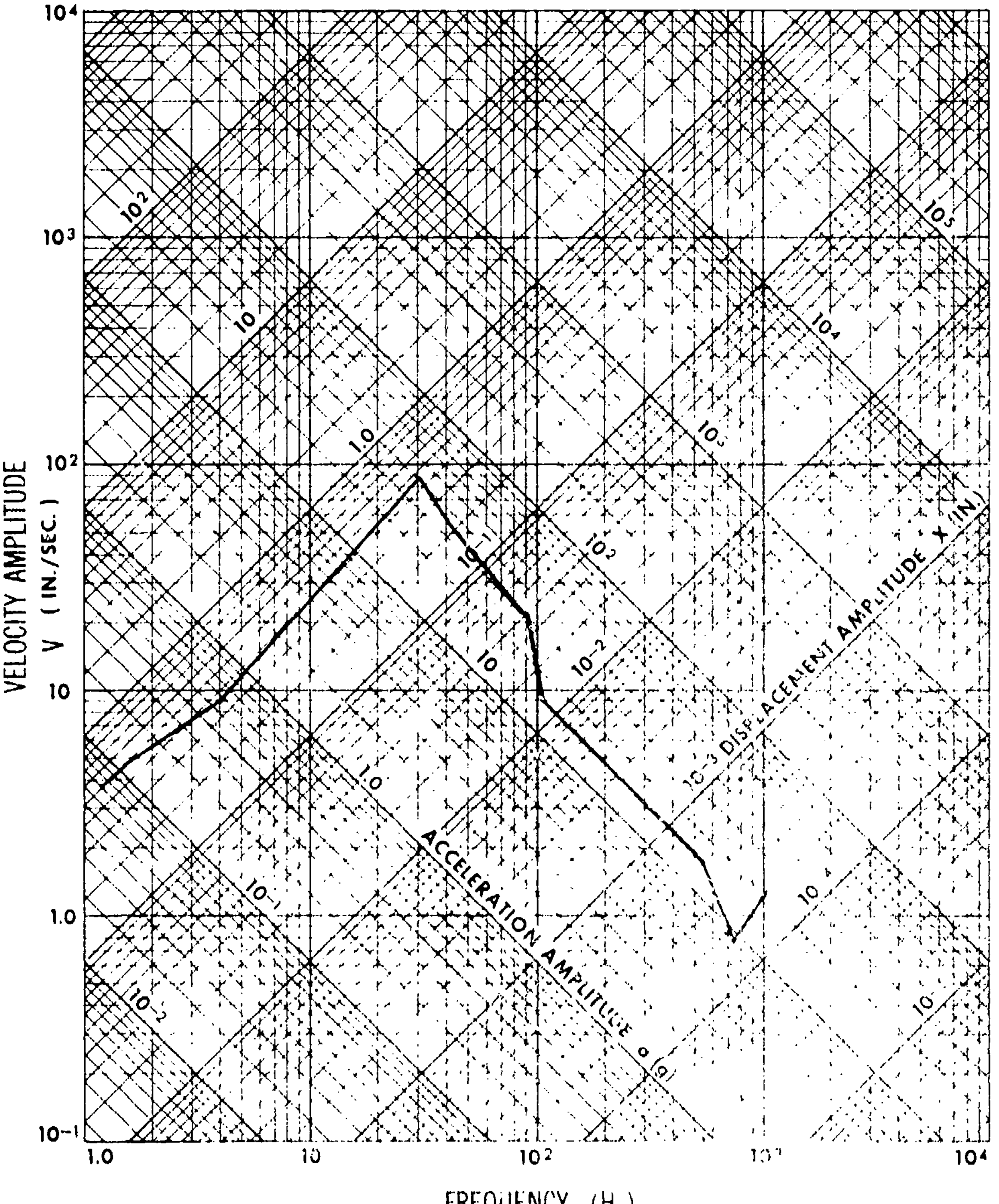

158TS
Z-Axis

ITEM: Valve, Thermal Water, for Air Compressor
REFERENCE: B-8
DESCRIPTION: 15 gpm, 80 psig, 125°F
MANUFACTURER: Powers
STATISTICS: Size: 17 x 6 x 6 in.
AXIS IDENTIFICATION: x - horizontal; y - vertical; z - transverse

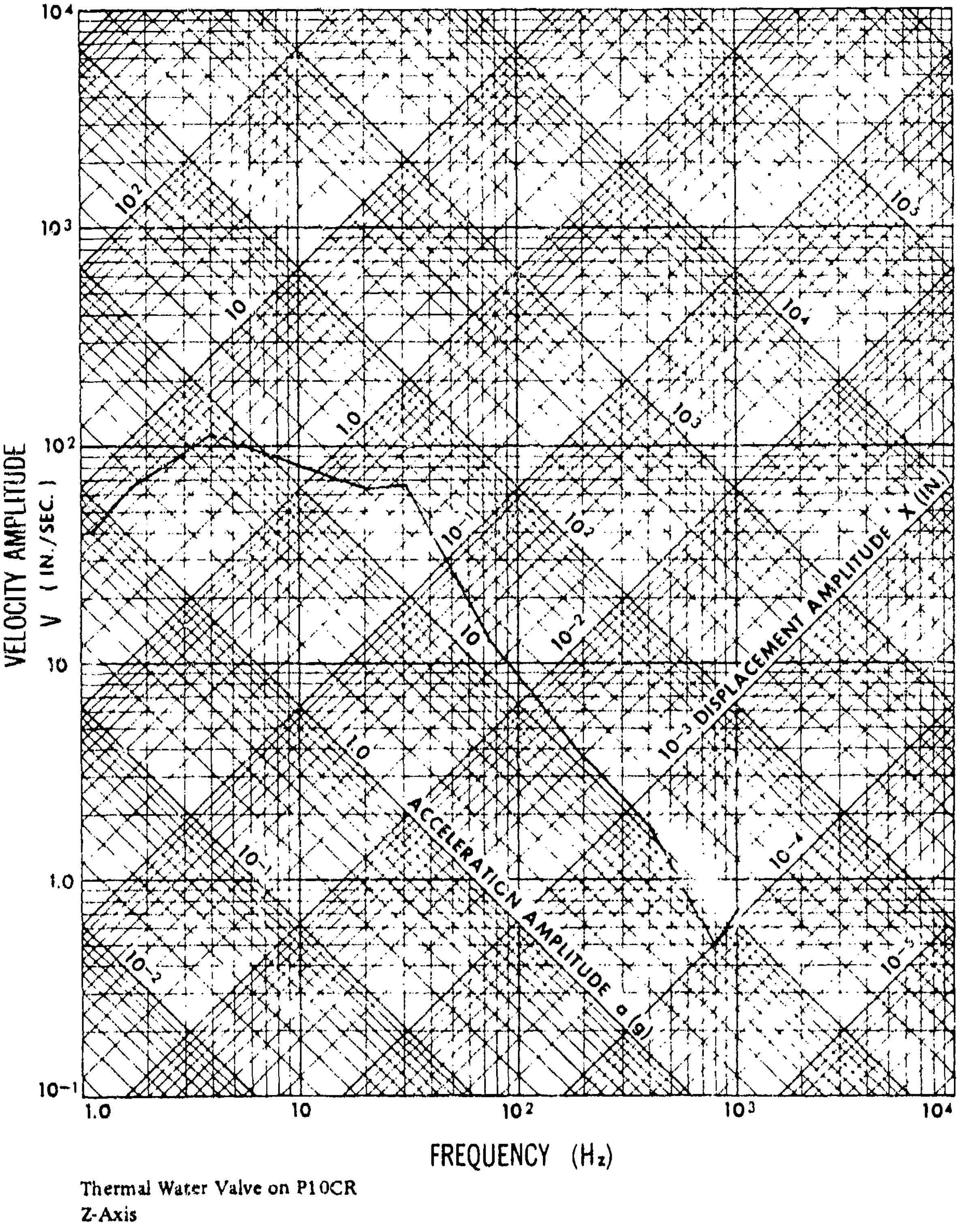
NAVFAC / NCEL
SHOCK DATA ANALYSIS




Thermal Water Valve on P10CR
X-Axis




Thermal Water Valve on P10CR
Y-Axis

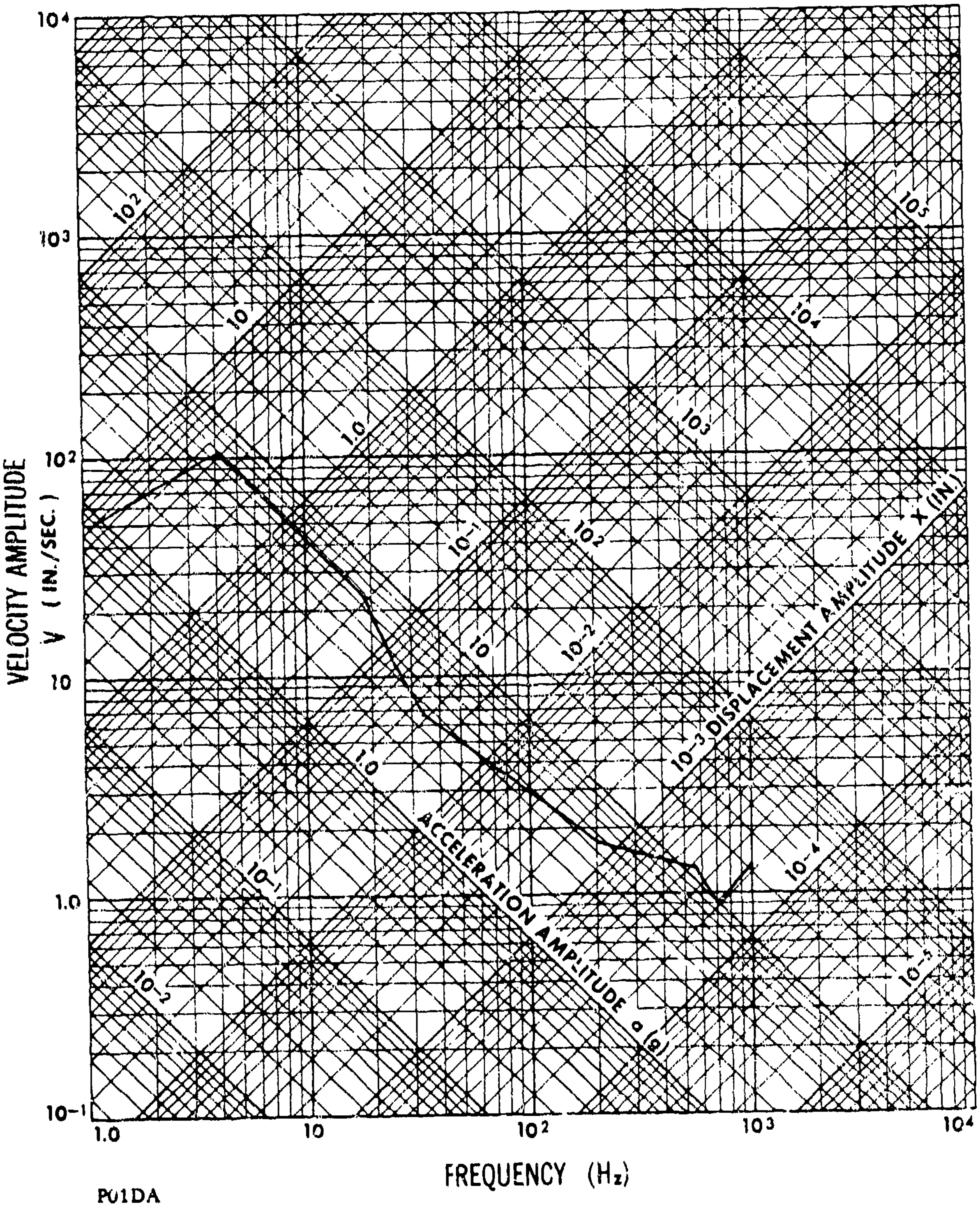



Thermal Water Valve on P10CR
Z-Axis

ITEM: Air Drier, Instrument
REFERENCE: B-9
DESCRIPTION: 250 psi, 10 cfm, 115 VAC, 1 phase
MANUFACTURER: Pall Trinity Micro Corp.
P/N: 5AE4-6AD 9811-7
STATISTICS: Weight: 400 lb; Size: 21 x 32 x 74 in.
AXIS IDENTIFICATION: x - longitudinal (depth); y - vertical (longest dimension); z - transverse (width)
NOTE: There was no structural damage during or after testing. One bulb burned out during testing.

NAVFAC / NCEL

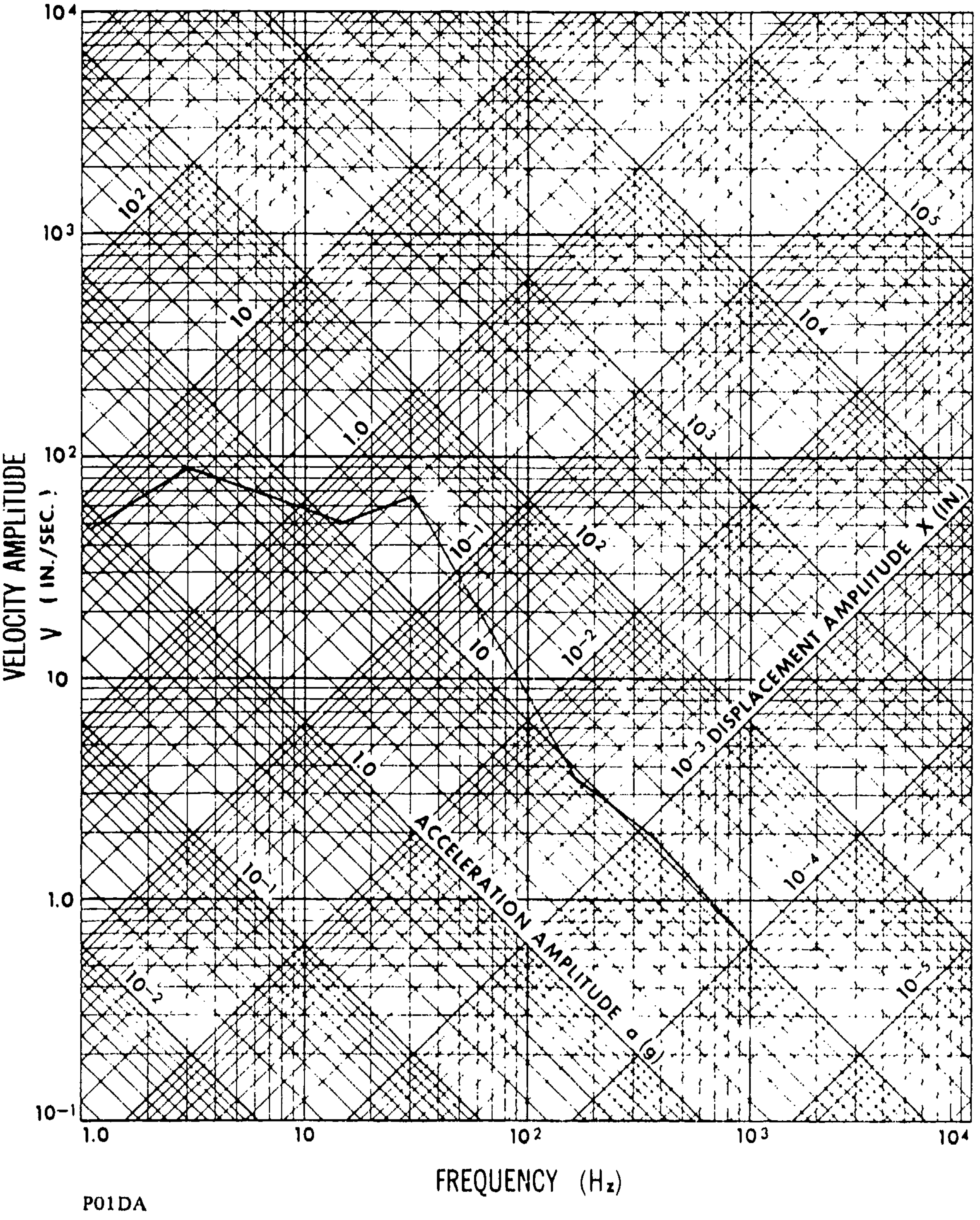
SHOCK DATA ANALYSIS



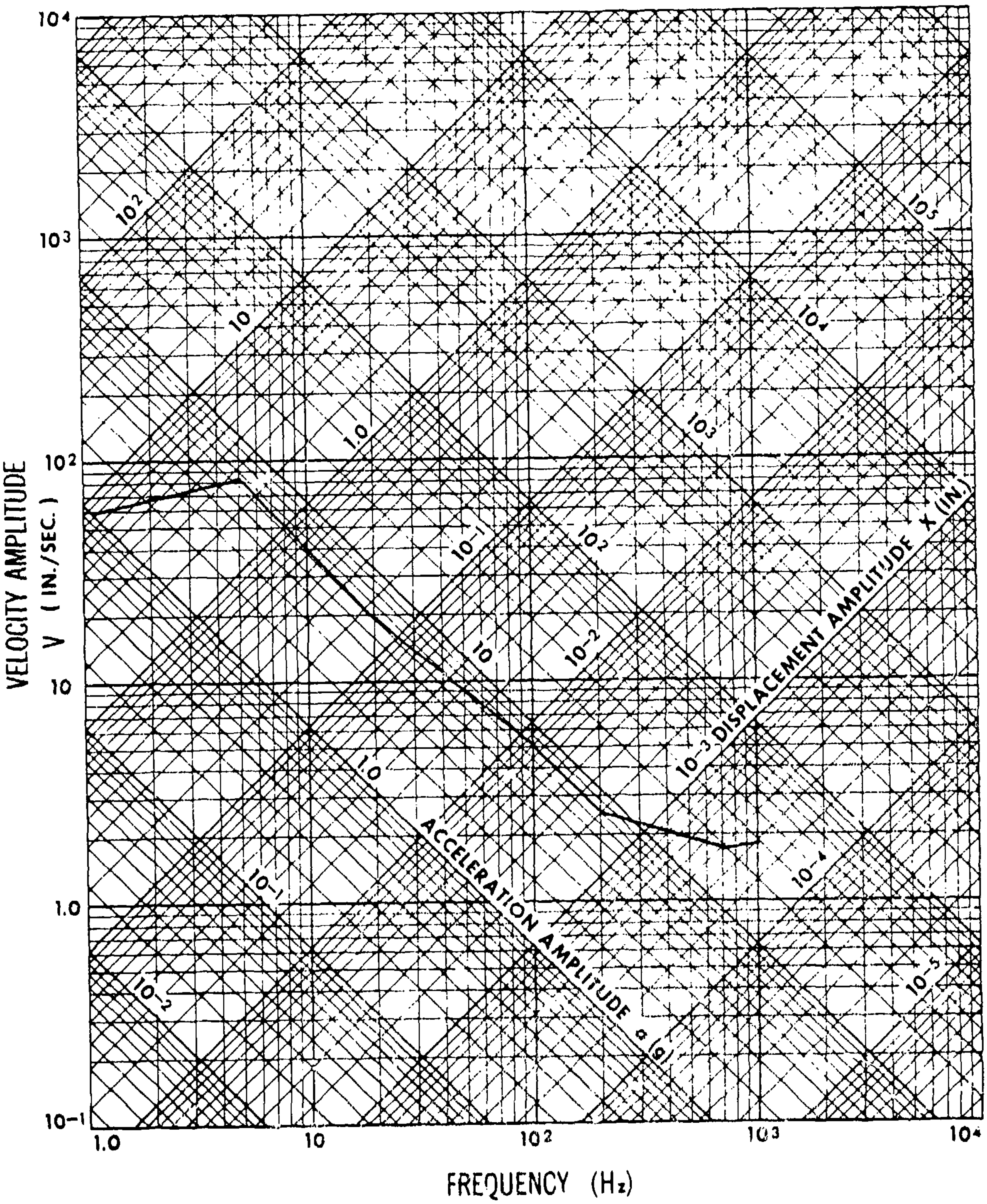

P01DA
X-Axis

**NAVFAC / NCEL**

**SHOCK DATA ANALYSIS**




P01DA
Y-Axis

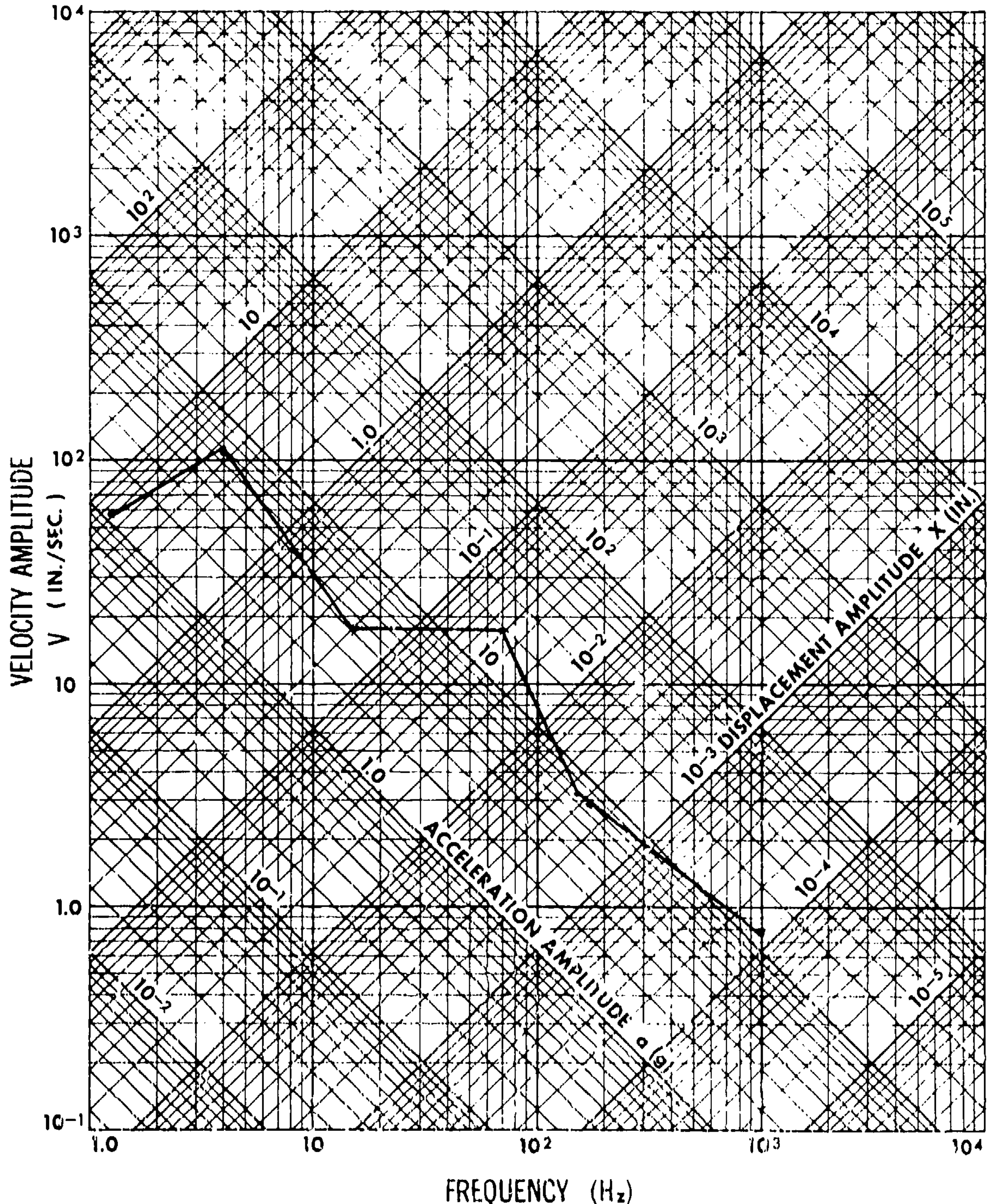

VELOCITY AMPLITUDE
V (IN./SEC.)
FREQUENCY (Hz)
ACCELERATION AMPLITUDE a (g)
DISPLACEMENT AMPLITUDE X (IN.)
1.0
10
$10^2$
$10^3$
$10^4$
$10^{-1}$

P01DA
Z-Axis

ITEM: Control Panel (for Air Compressor)
REFERENCE: B-10
DESCRIPTION: 120 VAC, 460 VAC, 0-250 psig
MANUFACTURER: Chicago Pneumatic Tool Co.
P/N: 0139CP
STATISTICS: Weight: 422 lb; Size: 12 x 35 x 63 in.
AXIS IDENTIFICATION: y - vertical; x - transverse; z - front to back
NOTE: Tests in y-axis caused vibration disable switch to open

ITEM: Drive Motor (for Compressor)
REFERENCE: B-10
DESCRIPTION: 460 VAC, 3 phase, 2,020 rpm, mounting (four 5/8-in. bolts)
MANUFACTURER: U.S. Electric Motor
P/N: 9104783
STATISTICS: Weight: 1,105 lb; Size: 33 x 30 x 25 in.

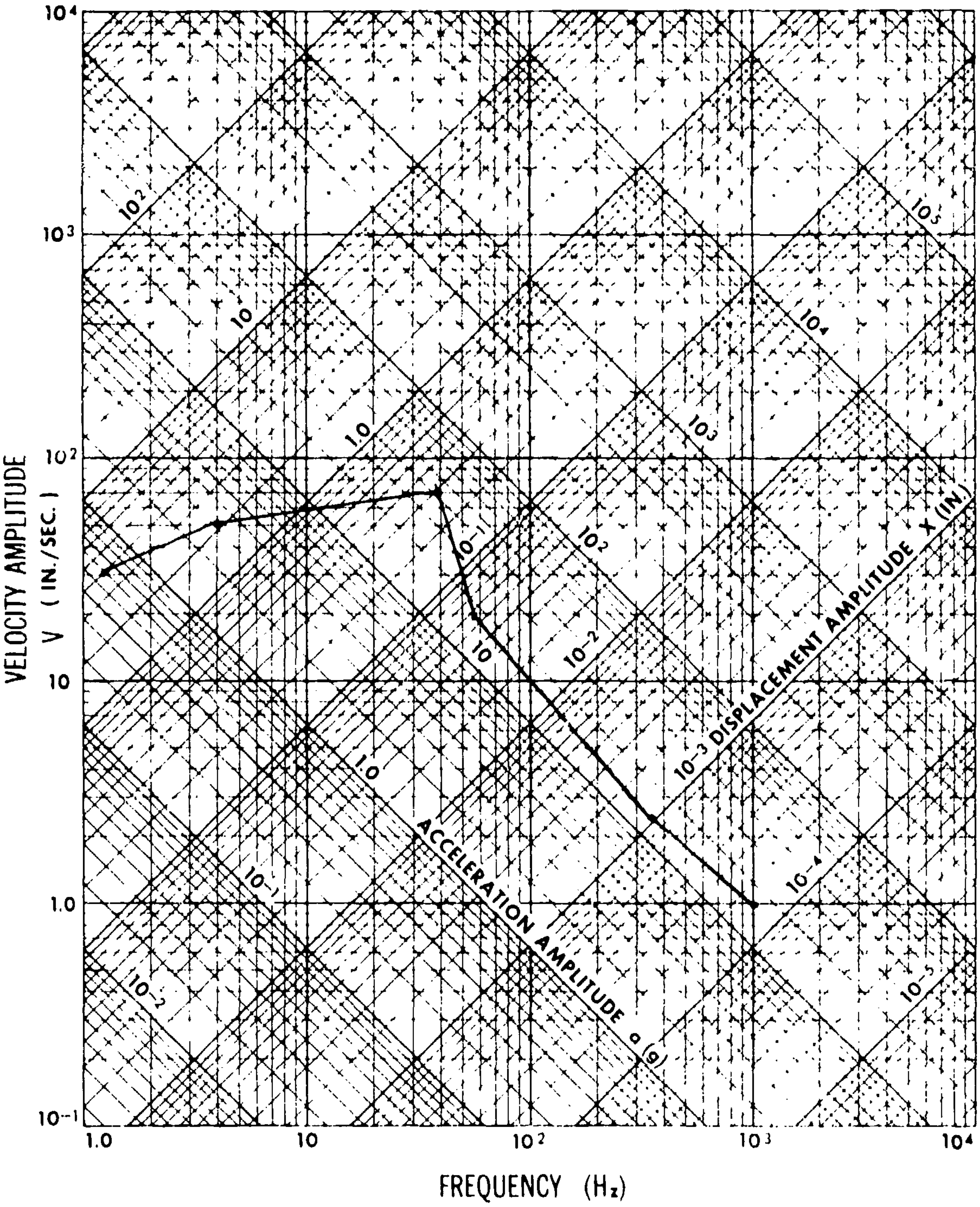
ITEM: Temperature Switch
REFERENCE: B-10
MANUFACTURER: United Electronics
P/N: 0139/TS
STATISTICS: Weight: 5 lb; Size 8 x 5 x 2 in.
NOTE: Mounted to the side of the control panel. Y-axis tests caused the switch contacts to momentarily open. This causes a shutdown, and the system must be restarted. No permanent damage occurred.


VELOCITY AMPLITUDE
V (IN./SEC.)
FREQUENCY (Hz)
ACCELERATION AMPLITUDE a (g)
DISPLACEMENT AMPLITUDE X (IN.)
10⁴
10³
10²
10
1.0
10⁻¹
10⁻²
10⁻³
10⁻⁴
10⁻⁵
10⁵

P01CR
X-Axis

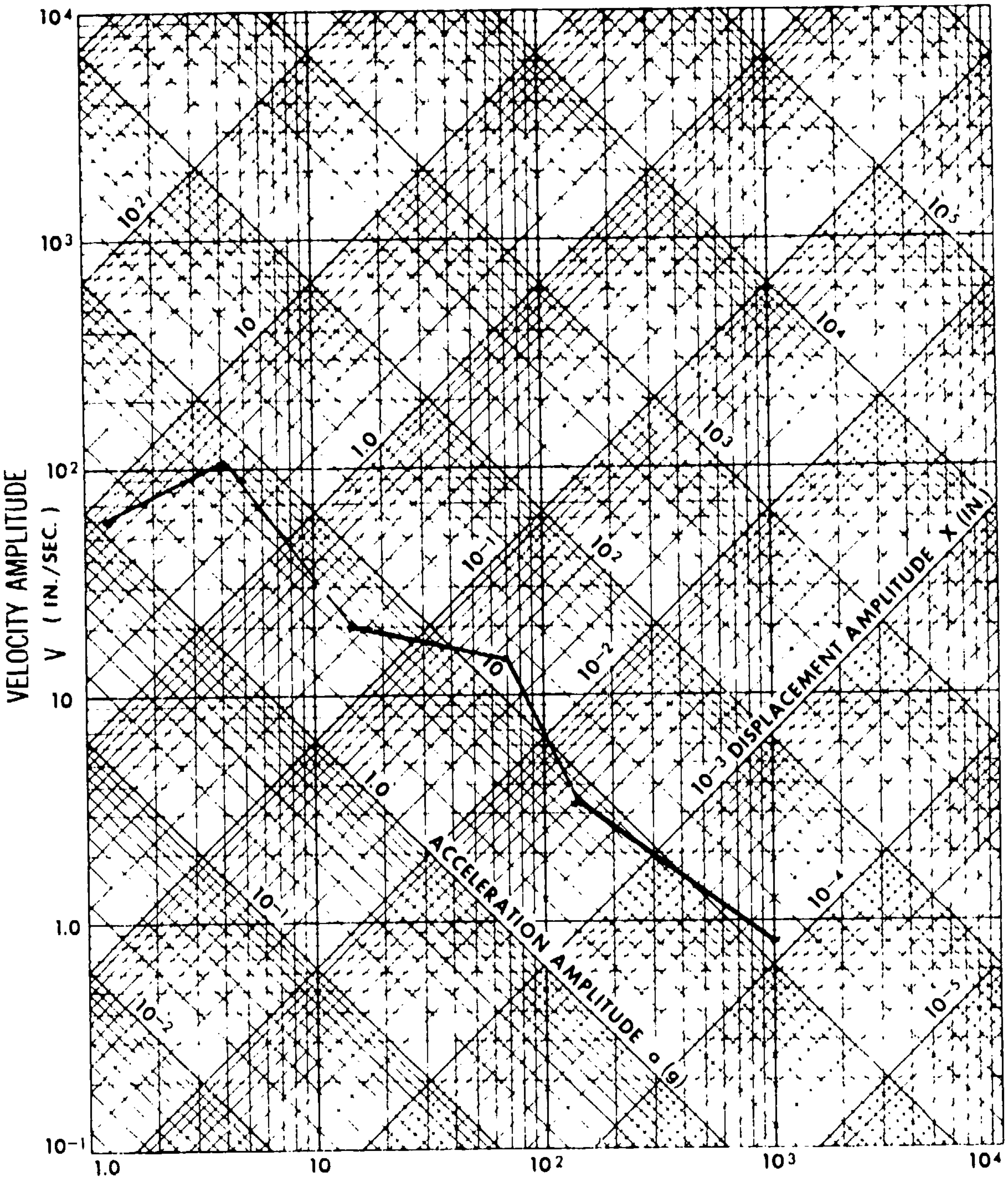
NAVFAC / NCEL

SHOCK DATA ANALYSIS




P01CR
Y-Axis

## NAVFAC / NCEL
## SHOCK DATA ANALYSIS

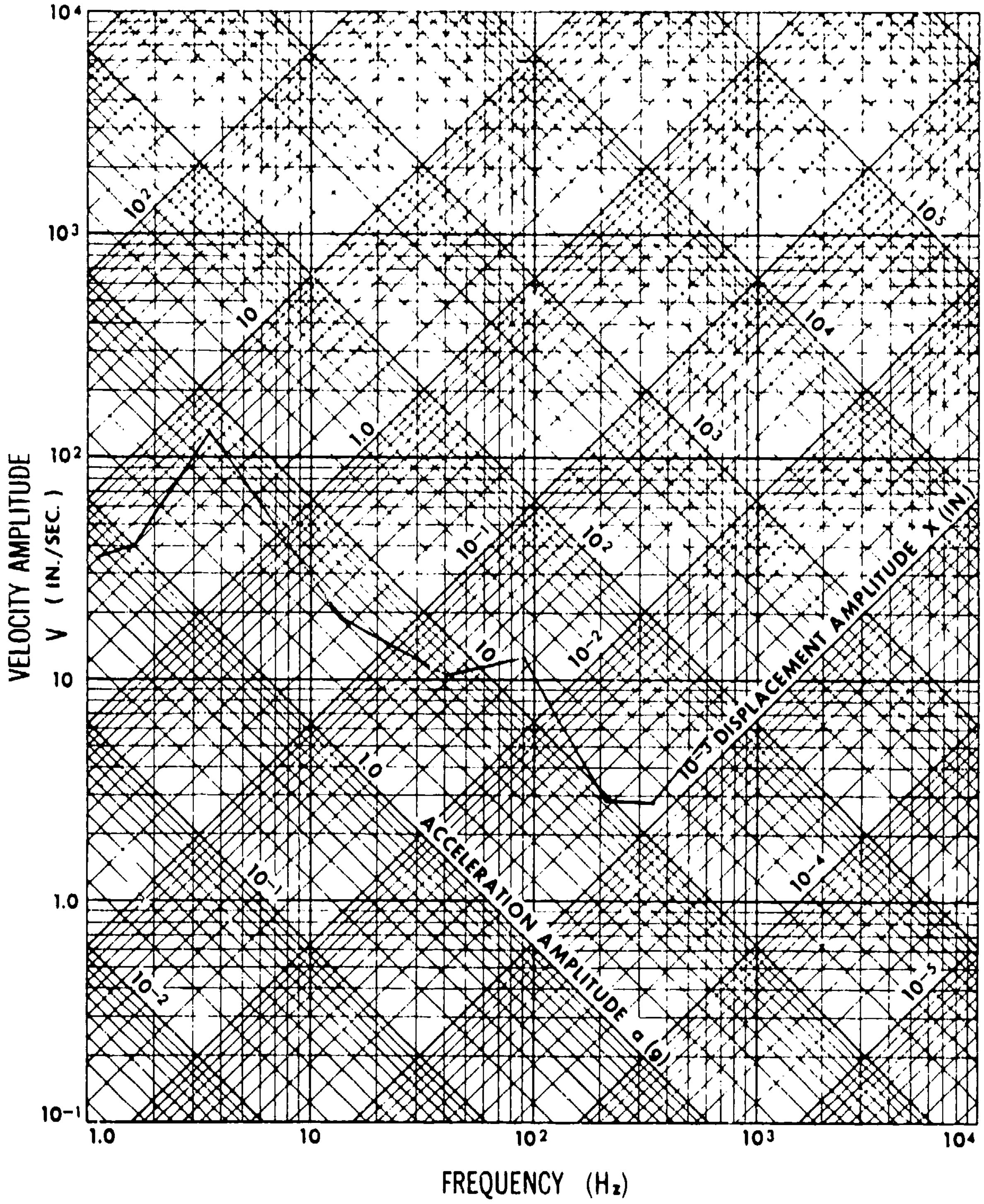



P01CR
Z-Axis

ITEM: Oil Pump and Relief Valve Assembly/Oil Cooler
REFERENCE: B-11
DESCRIPTION: Components for a 660-ton chille[illegible]
MANUFACTURER: Century
STATISTICS: Weight: 60 lb (pump and valve[illegible] [illegible] ooler);
Size: 19 x 15 in. diameter (pu[illegible] [illegible] valve[illegible] 30 x 7 x 7 in. (cooler)
AXIS IDENTIFICATION: x - longitudinal (along long[illegible] [illegible] sion);
y - vertical; z - transverse
NOTE: The oil pump and relief valve assembly [illegible] teste[illegible] a unit with the oil cooler.

NAVFAC / NCEL

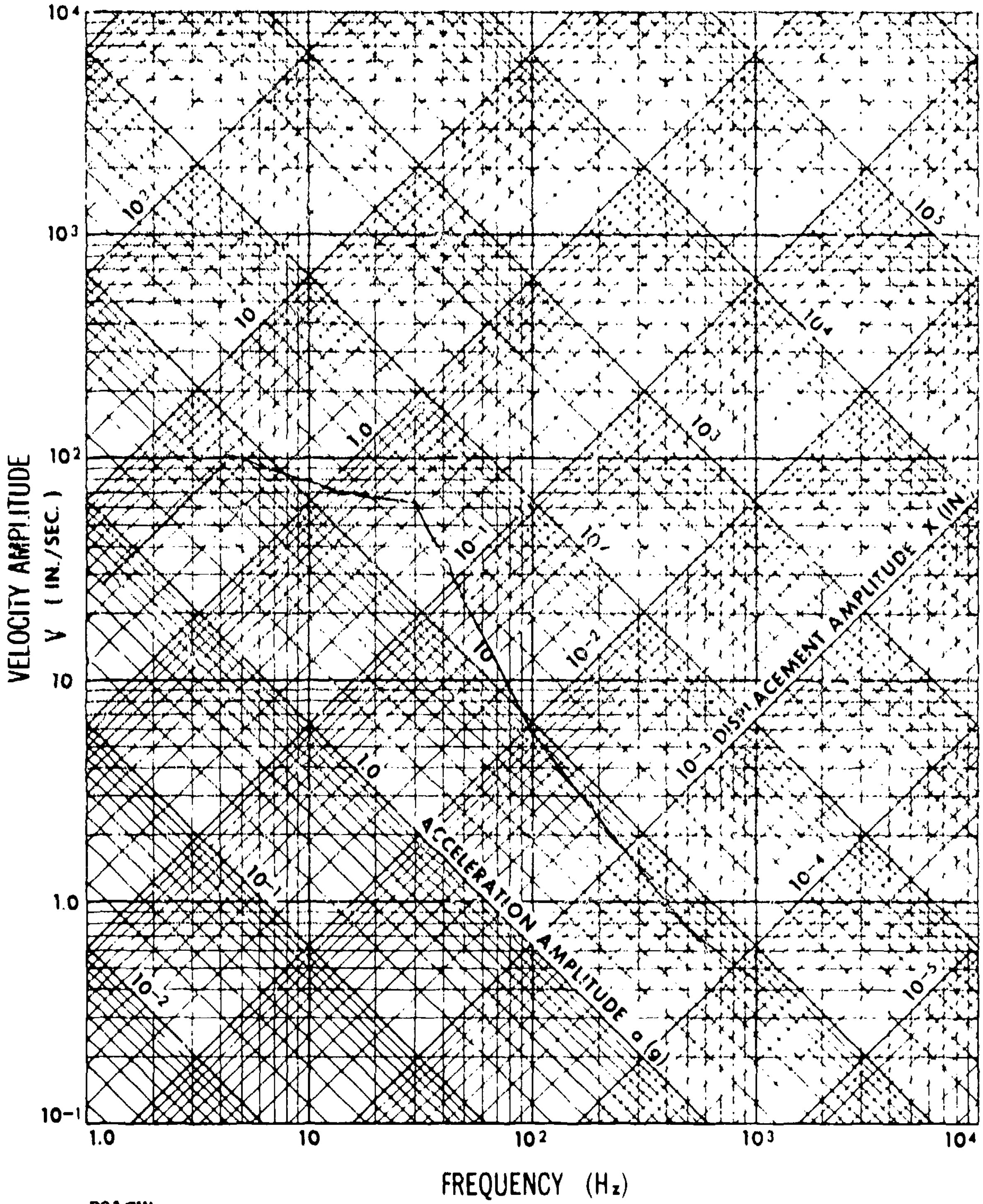
SHOCK DATA ANALYSIS




P03CW
X-Axis

NAVFAC / NCEL

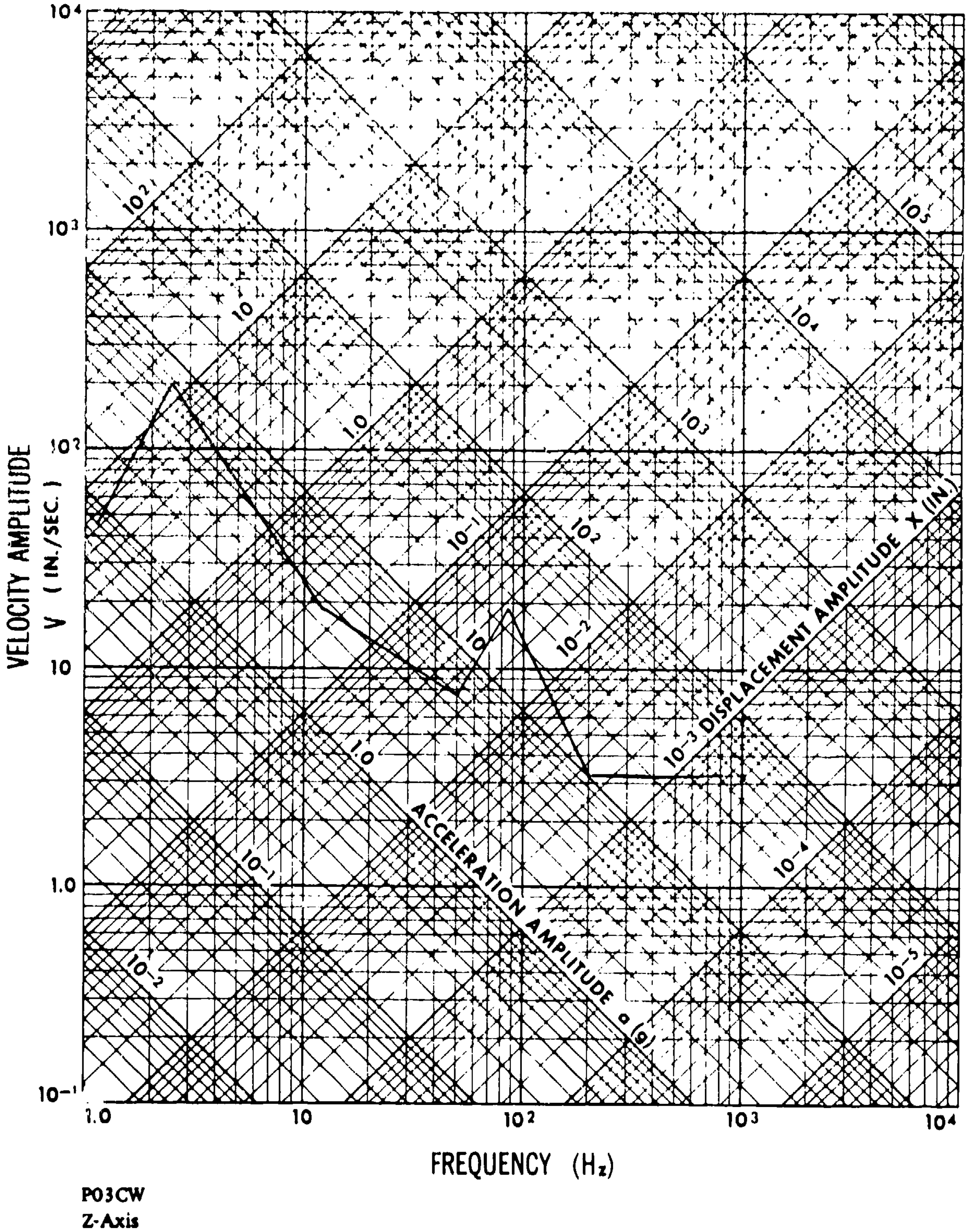
SHOCK DATA ANALYSIS




P03CW
Y-Axis

NAVFAC / NCEL

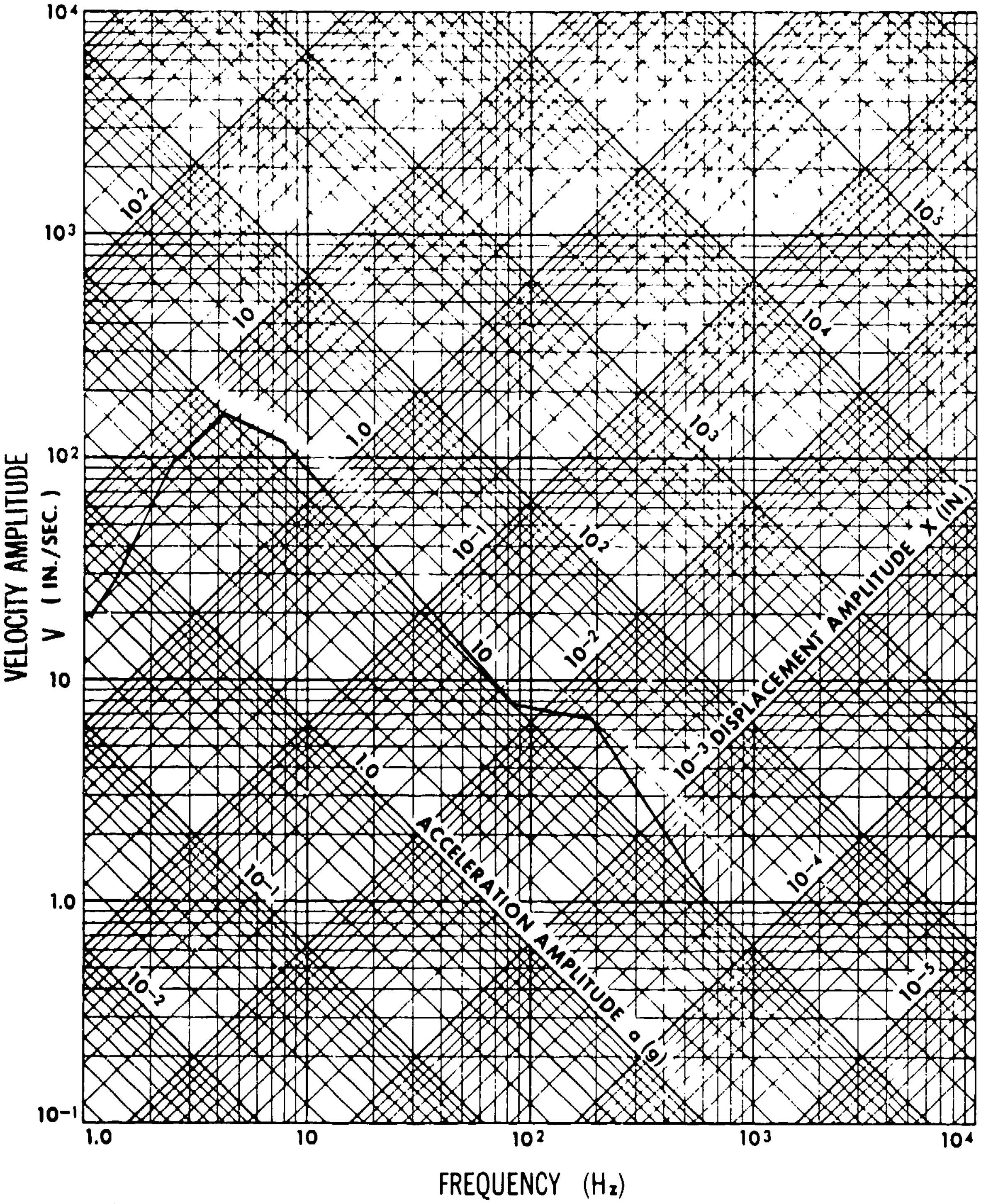
SHOCK DATA ANALYSIS




P03CW
Z-Axis

ITEM: Pump, Peripheral, Turbine
REFERENCE: B-12
DESCRIPTION: 3/4-hp, 1,750 rpm, 460 V, 1.3 amperes, 9 gpm
MANUFACTURER: General Signal Co. (Aurora)
P/N: Model CT-211-SS
STATISTICS: Weight: 200 lb; Size: 42 x 11-1/2 x 25 in.
AXIS IDENTIFICATION: x - vertical; y - longitudinal (along shaft axis); z - transverse

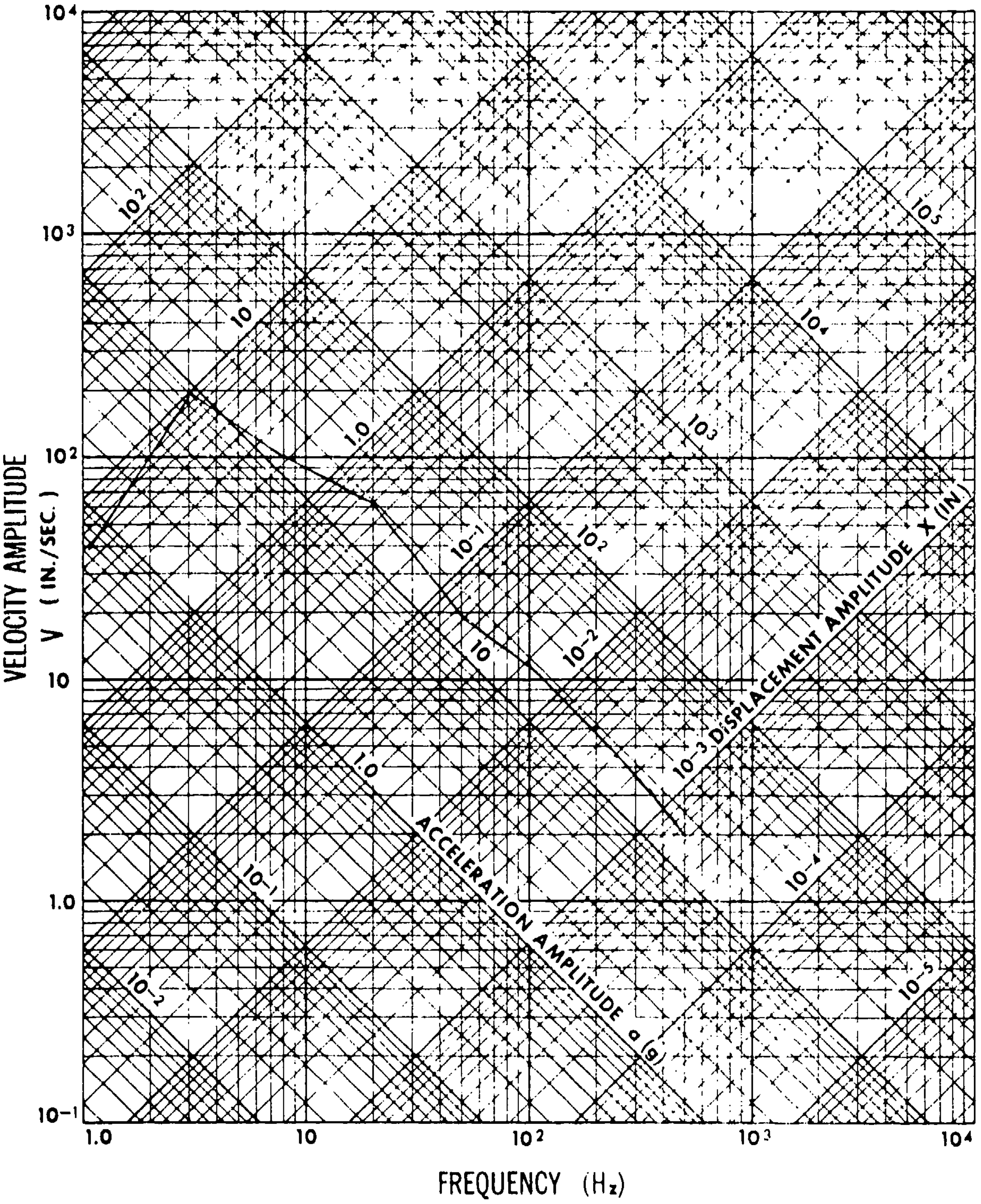
ITEM: Pump, Peripheral, Turbine
REFERENCE: B-12
DESCRIPTION: 7.5 hp, 3,600 rpm, 460 VAC
MANUFACTURER: General Signal Co. (Aurora)
STATISTICS: Weight: 580 lb; Size: 52 x 22 x 30 in.
AXIS IDENTIFICATION: x - vertical; y - longitudinal (along shaft axis); z - transverse




P01PT/P03PT
Z-Axis

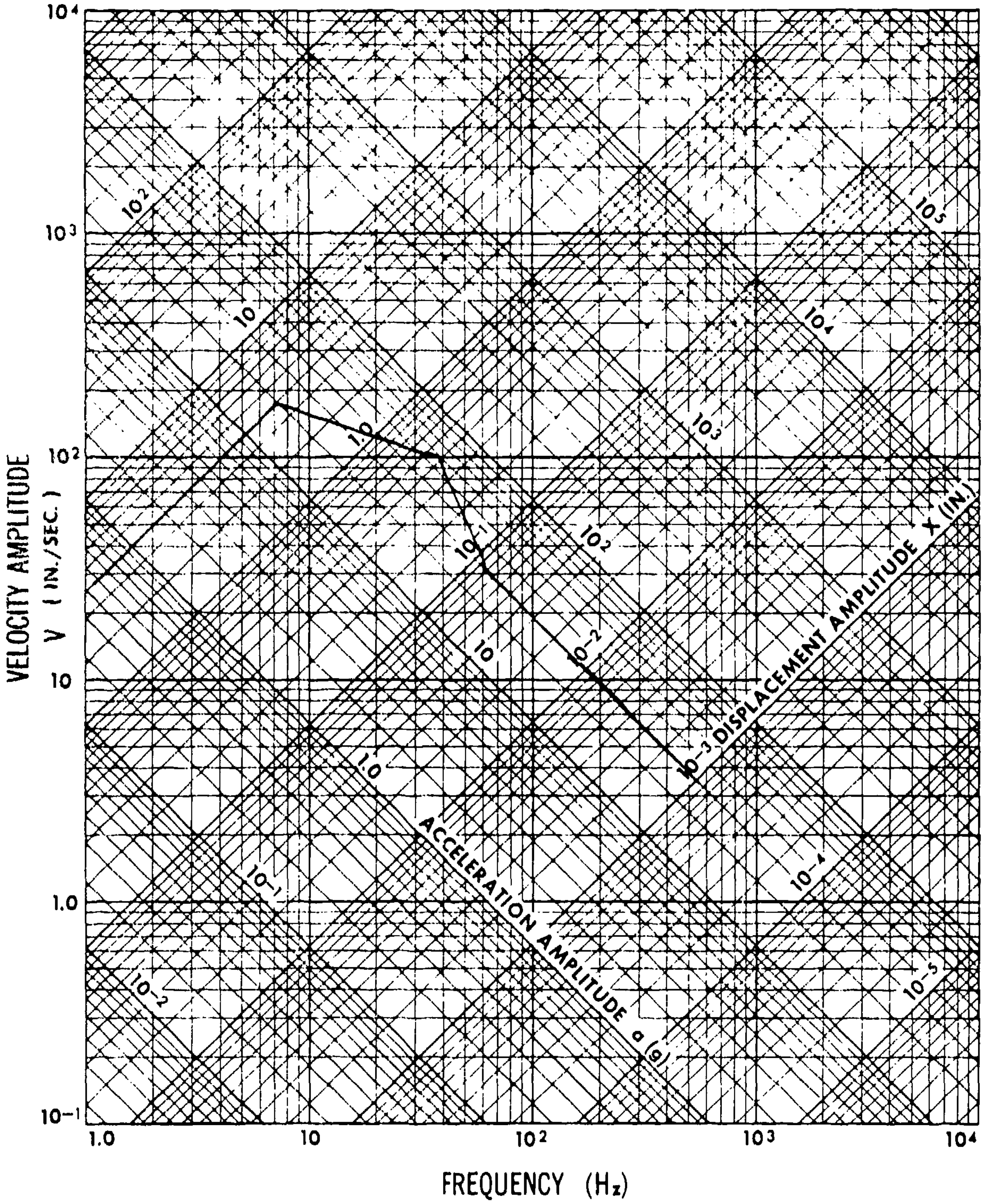
NAVFAC / NCEL

SHOCK DATA ANALYSIS



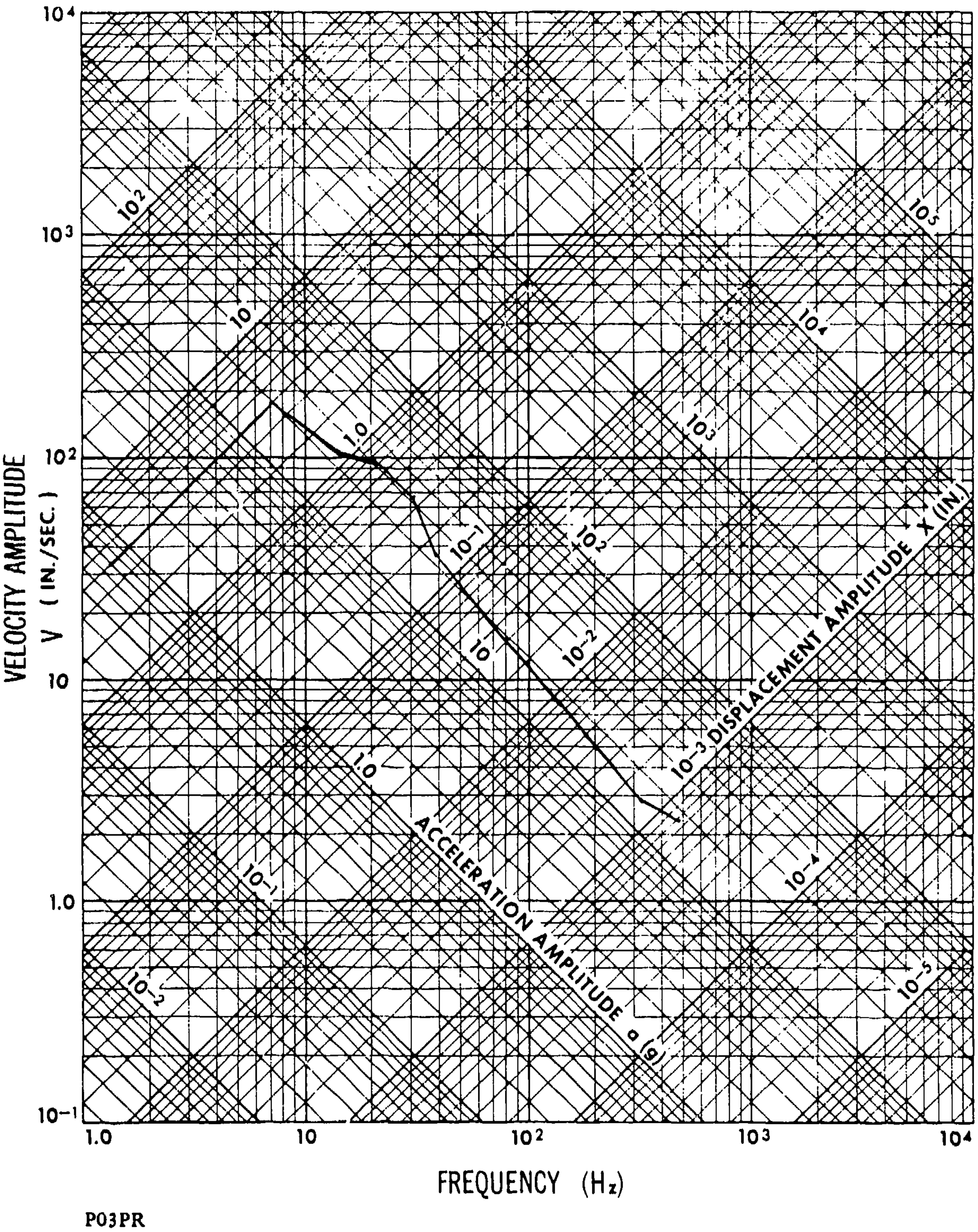

P01PT/P03PT
Horizontal X-Axis

## NAVFAC / NCEL
## SHOCK DATA ANALYSIS




P01PT/P03PT
Vertical

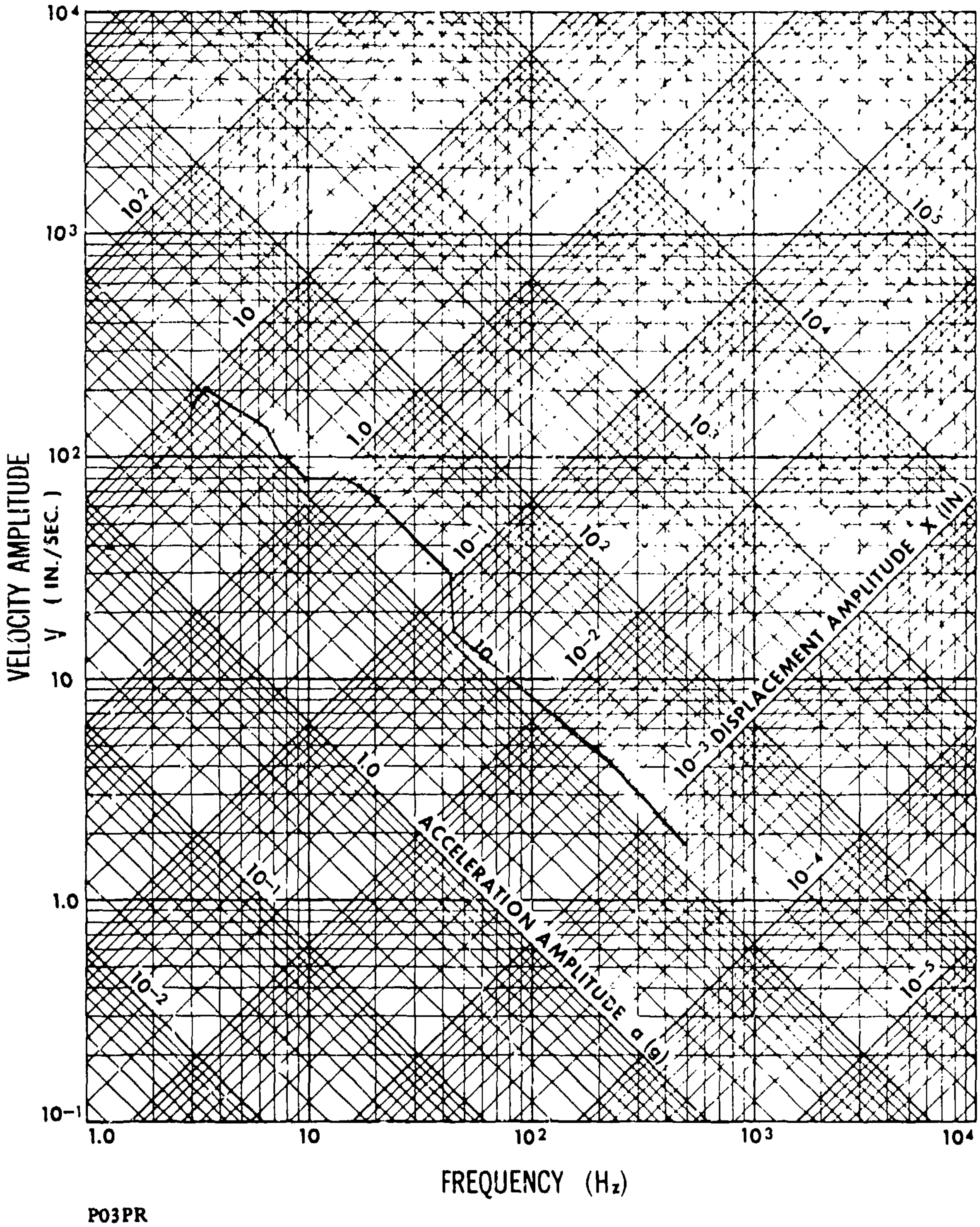
ITEM: Pump, Positive Displacement
REFERENCE: B-12
DESCRIPTION: 1-1/2 hp, 20 gpm, 1,745 rpm, 230/460 VAC, 5.4/2.7 amperes
MANUFACTURER: Haight Hydraulics, Inc.
P/N: 20DRM
STATISTICS: Weight: 126 lb; Size: 42 x 12 x 14.5 in.
AXIS IDENTIFICATION: y - vertical; x - horizontal (along shaft axis); z - transverse (to shaft)




P03PR
Y-Axis (Vertical)

NAVFAC / NCEL

SHOCK DATA ANALYSIS

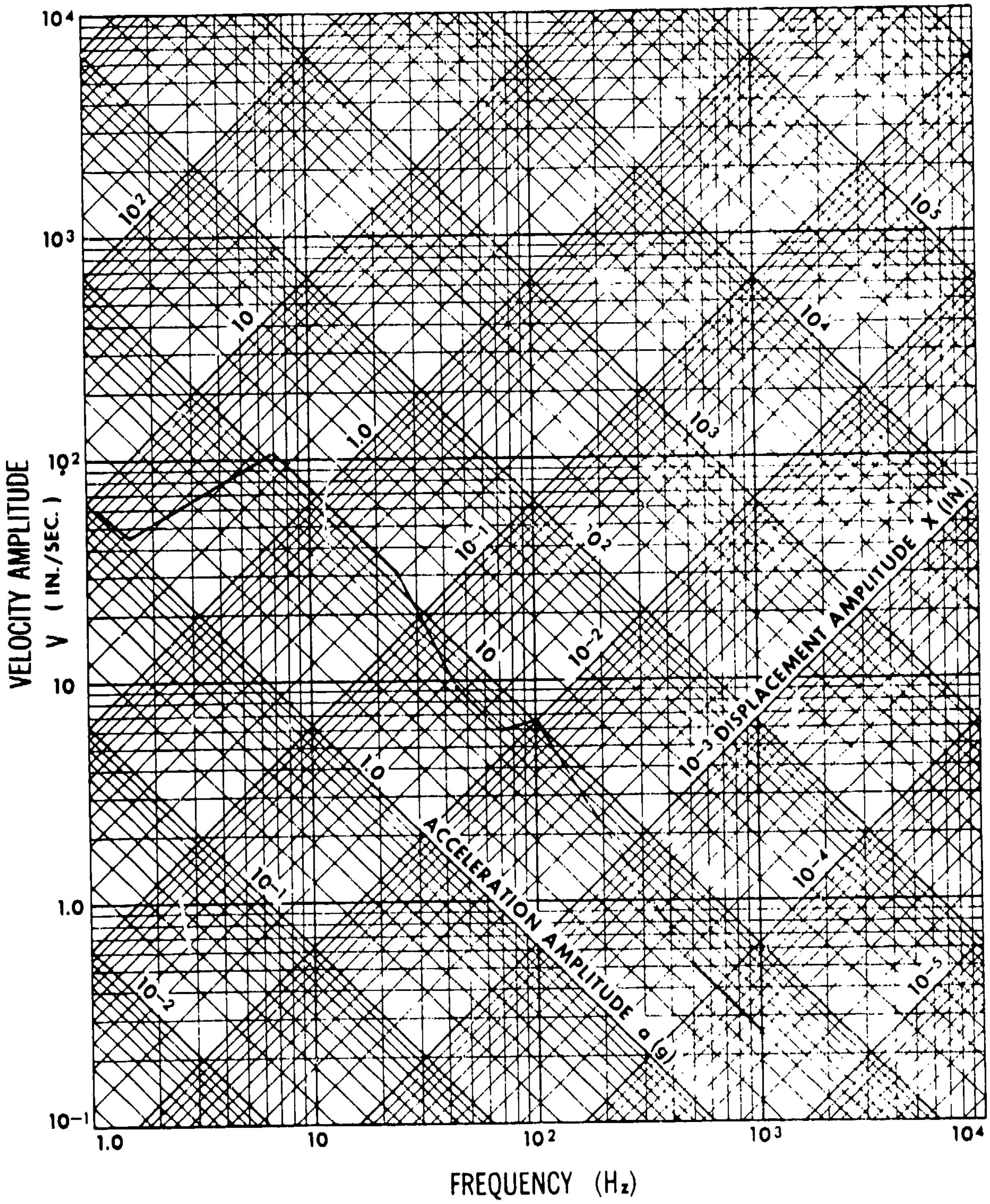



P03PR
X- and Z-Axis

ITEM: Fluorescent Light Fixture
REFERENCES: B-13 and B-14
DESCRIPTION: 4-ft tubes wth typical reflector suspended from rigid channel with two 18-in.-long by 3/8-in. threaded rods
MANUFACTURER: Stanley Electric Manufacturing Co., Altoona, Pa.
P/N: Type F10B
STATISTICS: Weight: 22 lb
AXIS IDENTIFICATION: x - longitudinal (along length of fixture); y - vertical; z - transverse
NOTE: No damage or degradation noted.

NAVFAC / NCEL

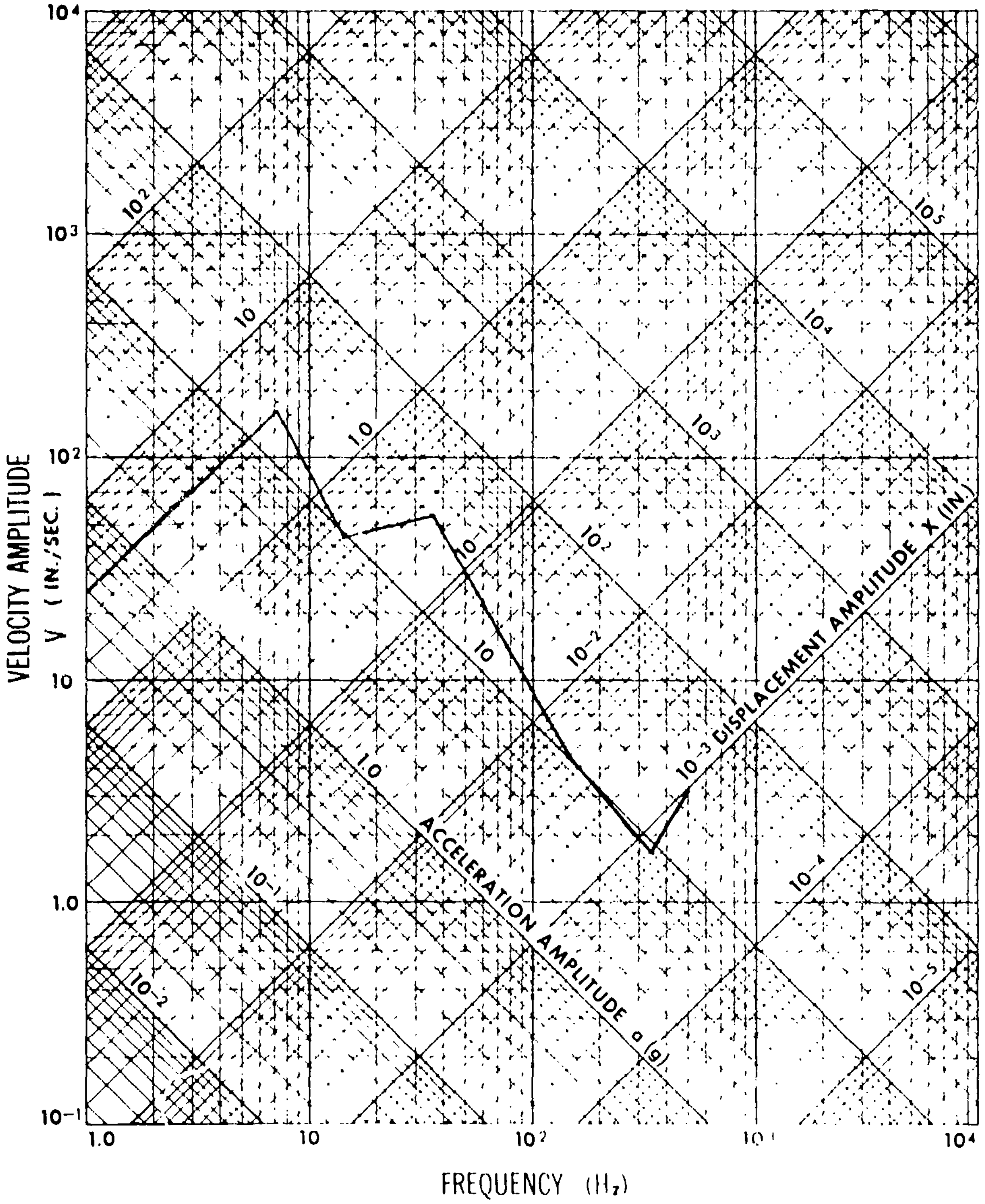
SHOCK DATA ANALYSIS




F10B
X-Axis

NAVFAC / NCEL

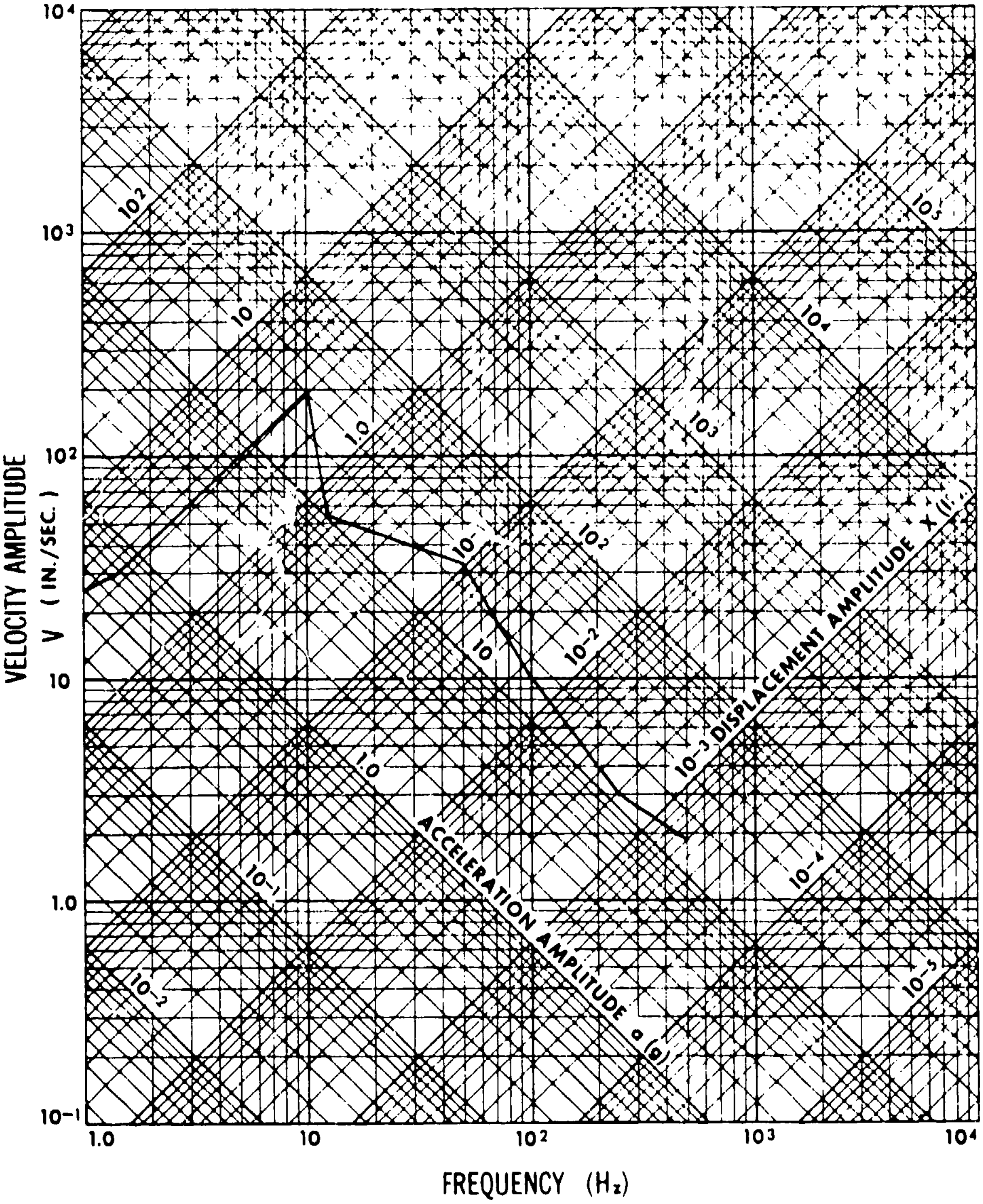
SHOCK DATA ANALYSIS




F10B
Y-Axis

NAVFAC / NCEL

SHOCK DATA ANALYSIS

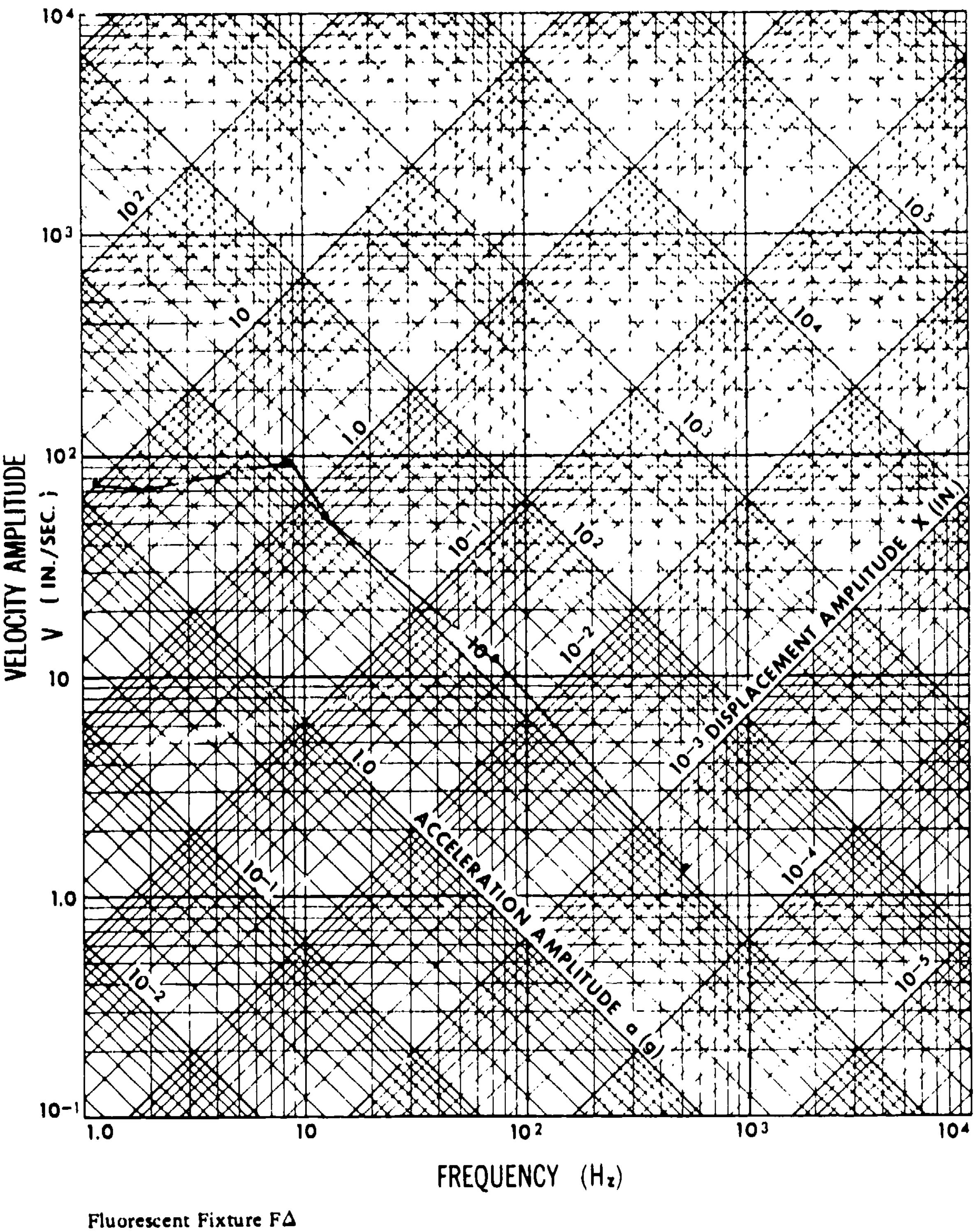



F10B
Z-Axis

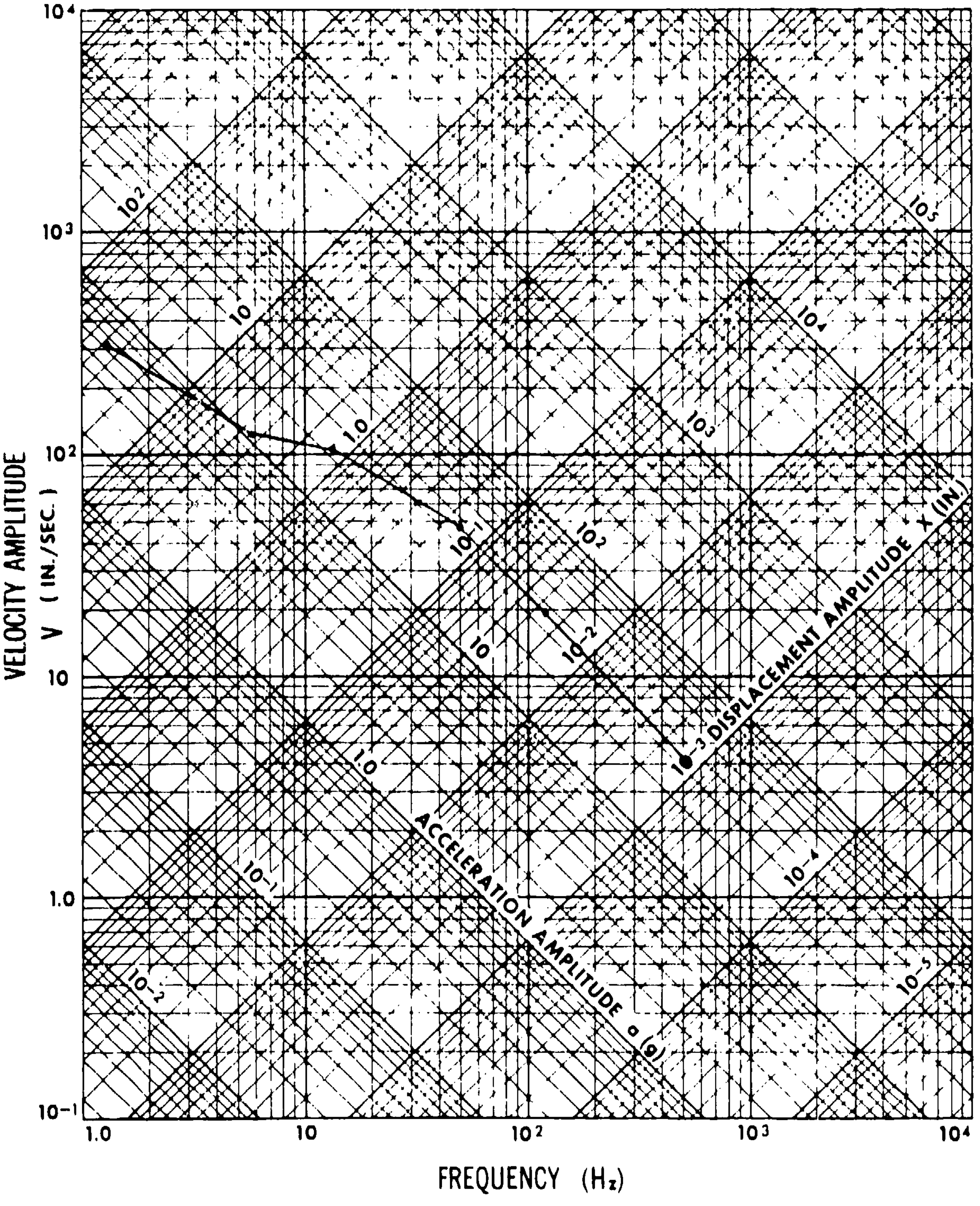
ITEM: Fluoresc^ t Light Fixture
REFERENCES: B-1 and B-14
DESCRIPTION: 110 VAC
MANUFACTURER: Stanley Electric Manufacturing Co., Altoona, Pa.
P/N: Type F4
STATISTICS: Weight: 10 lb
AXIS IDENTIFICATION: x - along tube length; y - vertical; z - transverse
NOTE: A hard-mounted prismatic box with two exposed tubes. No shades.

NAVFAC / NCEL

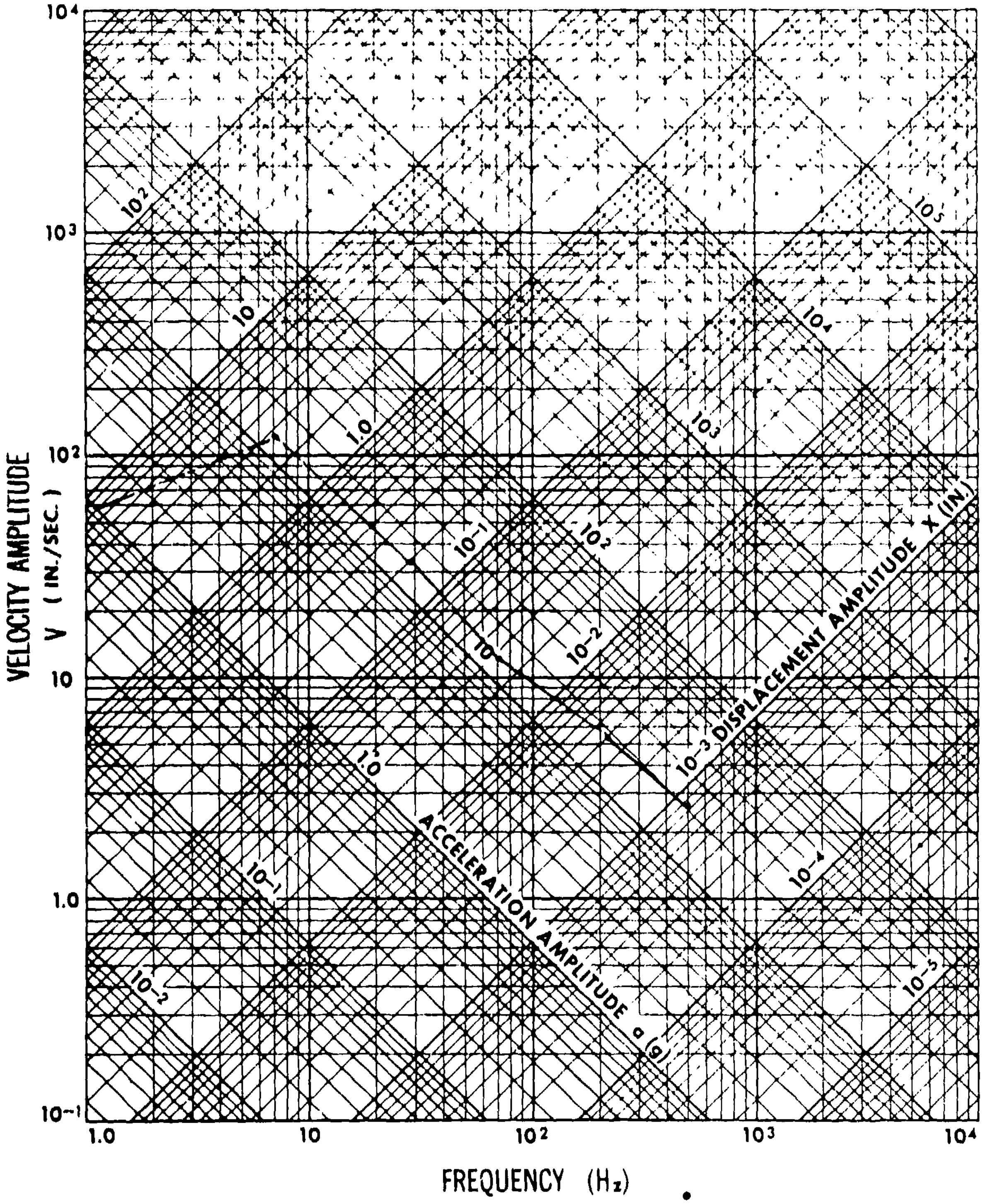
SHOCK DATA ANALYSIS




Fluorescent Fixture FΔ
X-Axis




Fluorescent Fixture FΔ
Y-Axis



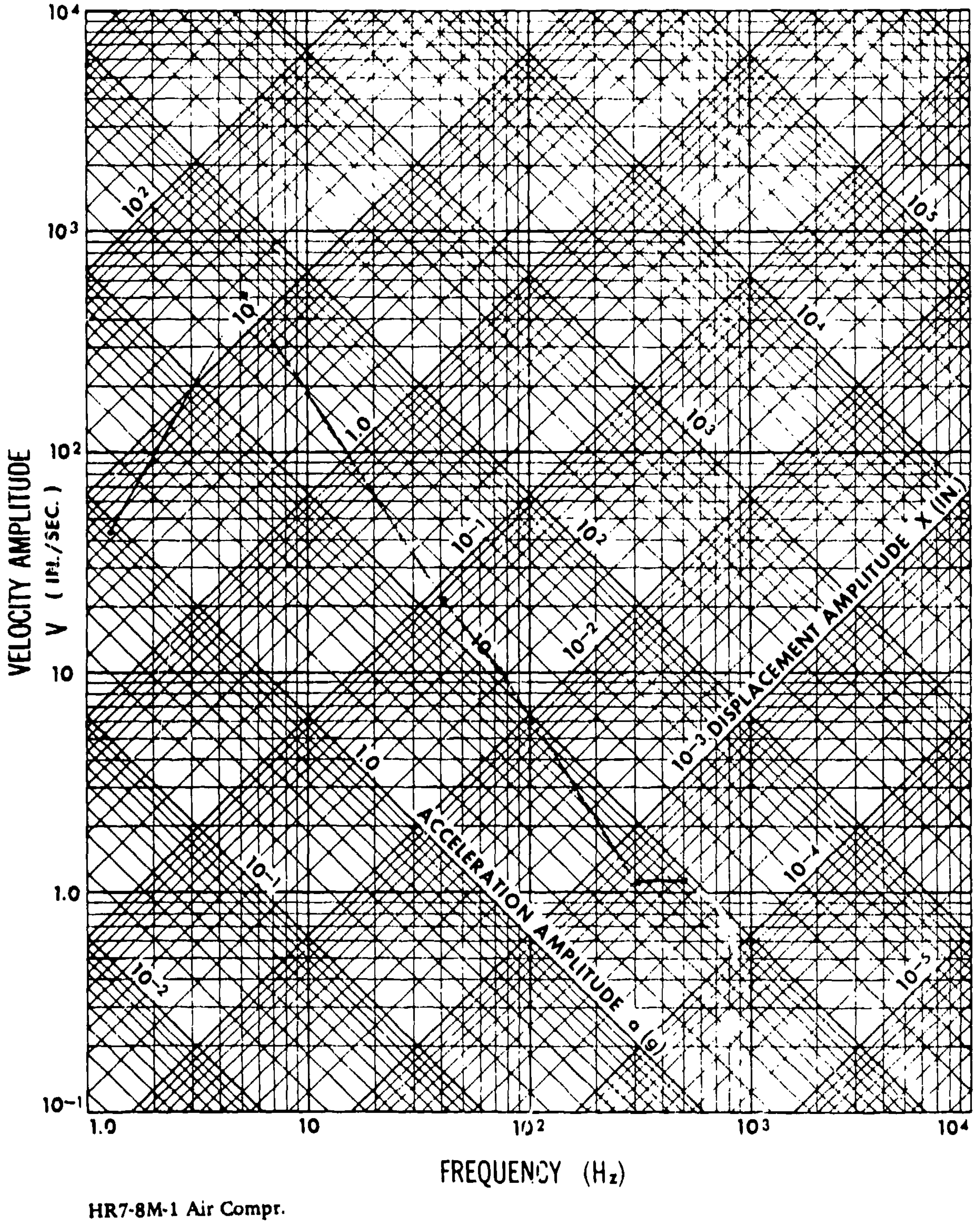

Fluorescent Fixture FΔ
Z-Axis

ITEM: Air Drier (for Compressor)
REFERENCE: B-15
DESCRIPTION: 120 psi, 20 cfm, 110 VAC, 1 phase, 7 amperes
MANUFACTURER: Kahn & Co.
STATISTICS: Weight: 300 lb; Size: 37 x 18 x 66 in.
AXIS IDENTIFICATION: x - longitudinal (along the longest horizontal dimension); y - vertical; z - transverse

ITEM: Air Compressor, Two Stage
DESCRIPTION: 250 psi, 300 cfm, 440 VAC, 3 phase, 11 amperes
MANUFACTURER: Champion Manufacturing Co.
P/N: HR7-8M-1
STATISTICS: Weight: 760 lb; Size: 66 x 22 x 50 in.
AXIS IDENTIFICATION: x - longitudinal (along the longest horizontal direction); y - vertical; z - transverse
NOTE: Only one horizontal direction was used at 45 degrees to x and z axes.

NAVFAC / NCEL

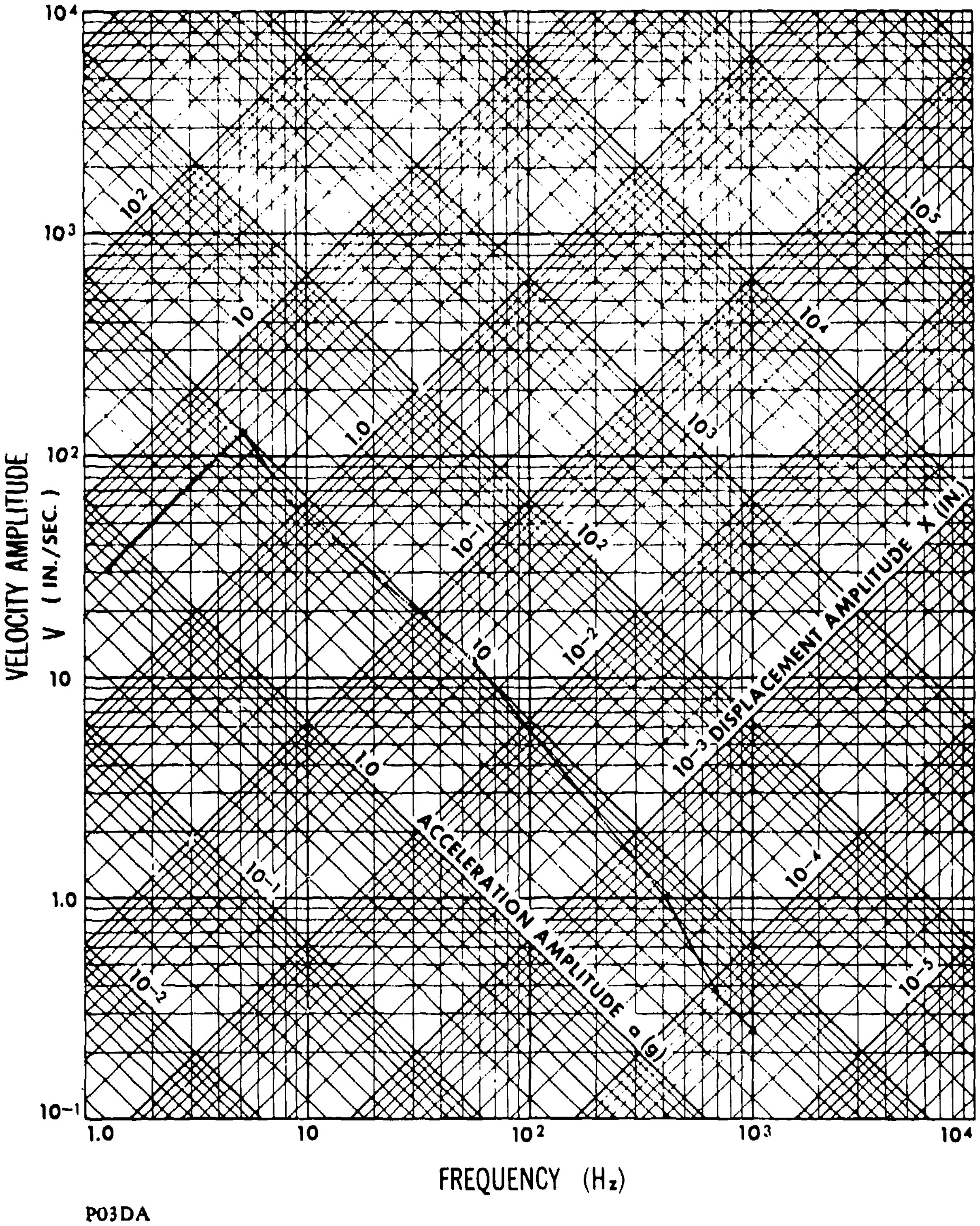
SHOCK DATA ANALYSIS




HR7-8M-1 Air Compr.
P03DA Air Drier
Horizontal Axis

NAVFAC / NCEL

SHOCK DATA ANALYSIS



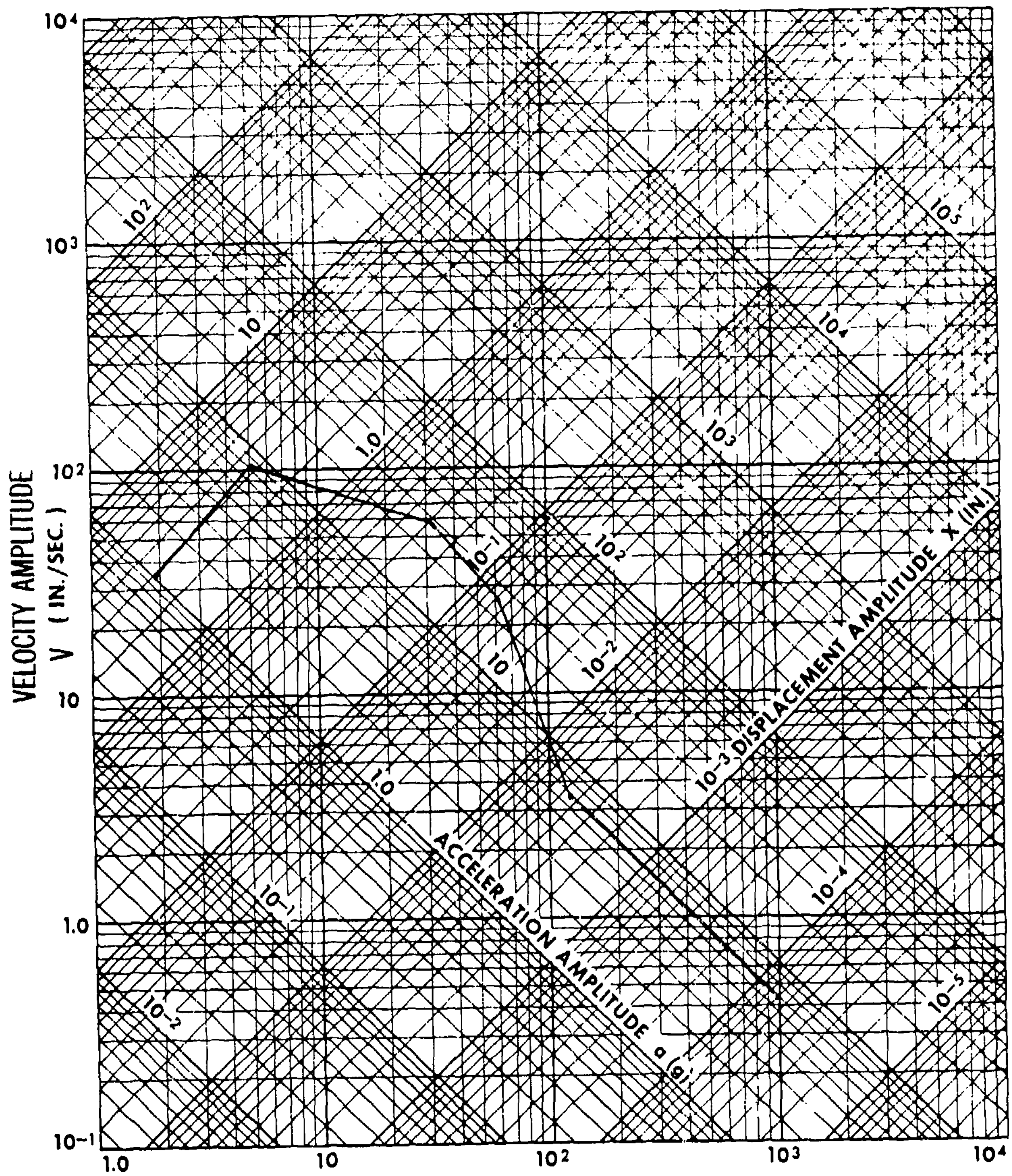

P03DA
HR7-8M-1
Vertical Axis

ITEM: Pumps, Waste Disposal (Sump Pump)
REFERENCE: B-16
DESCRIPTION: Sump pump with control panel, 30 hp, 1,800 rpm, 460 V, 3 phase, 500 gpm, 103 ft head, 1.5 ft suction
MANUFACTURER: Weil Pump Co.
P/N: S/N 213-053B Pump, and S/N 213-053 Control Panel
STATISTICS: Weight: 1,070 lb; Size: 33 in. diameter x 84 in. high
NOTE: The first test sample failed due to lubricant, but not considered due to shock. Spare replacement was tested without degradation or failure.

ITEM: Reverse Flow Sewage Ejector with Control Panel
REFERENCE: B-16
DESCRIPTION: 2 hp, 1,800 rpm, 400 VAC, 3 phase, 60 Hz, (120 VAC for controls), 100 gpm, 28 ft head, 1.5 ft suction
MANUFACTURER: Weil Pump Co.
P/N: Pump S/N 213-054B, Control Panel 213-054
STATISTICS: Weight: 840 lb (pump 770 lb, panel 80 lb); Size: 33 in. diameter x 84 in. high, panel 24 x 20 x 6-5/8 in.

NAVFAC / NCEL

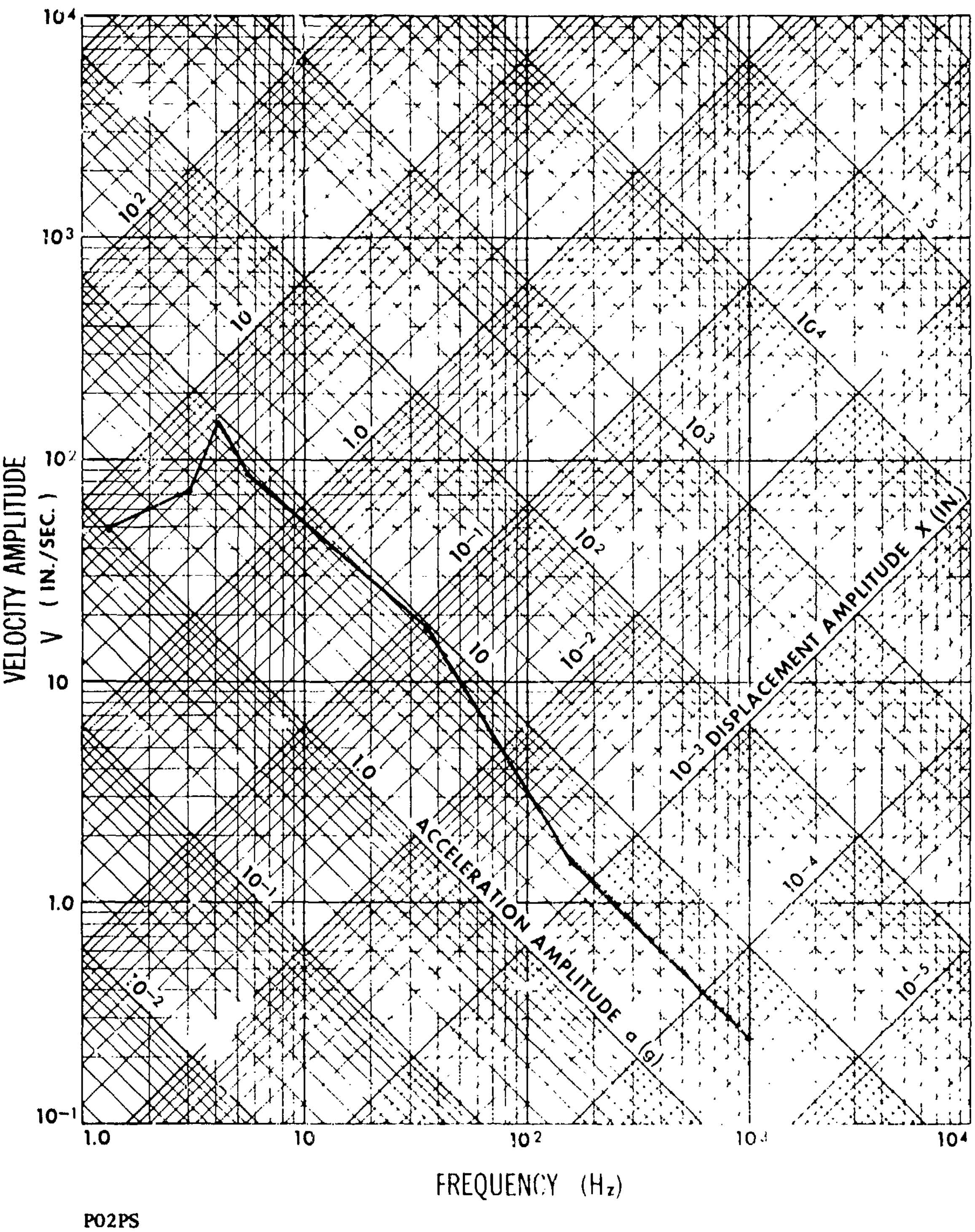
SHOCK DATA ANALYSIS




P02PS
P03PS
Vertical Axis

NAVFAC / NCEL

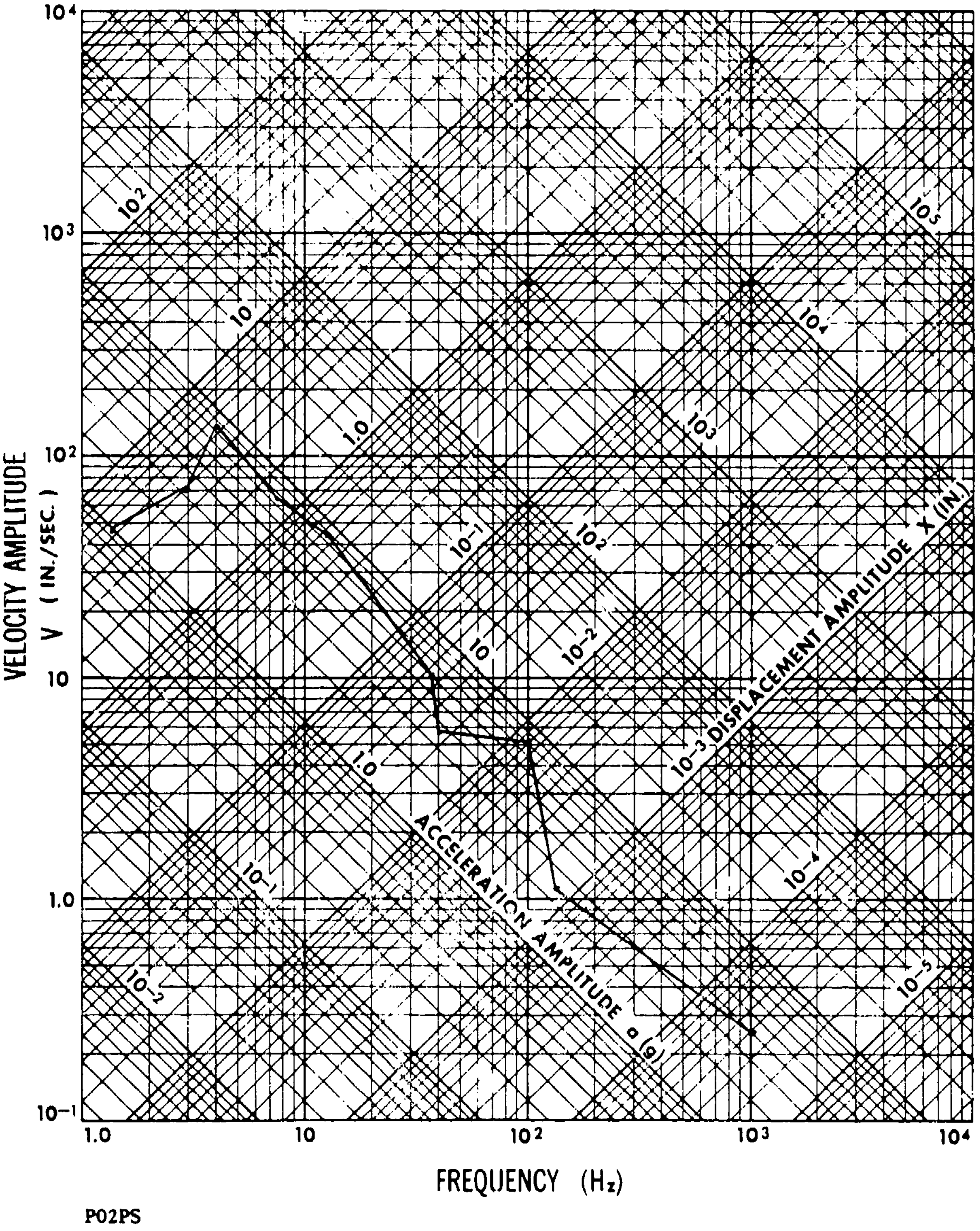
SHOCK DATA ANALYSIS




P02PS
P03PS
Longitudinal Axis

NAVFAC / NCEL

SHOCK DATA ANALYSIS

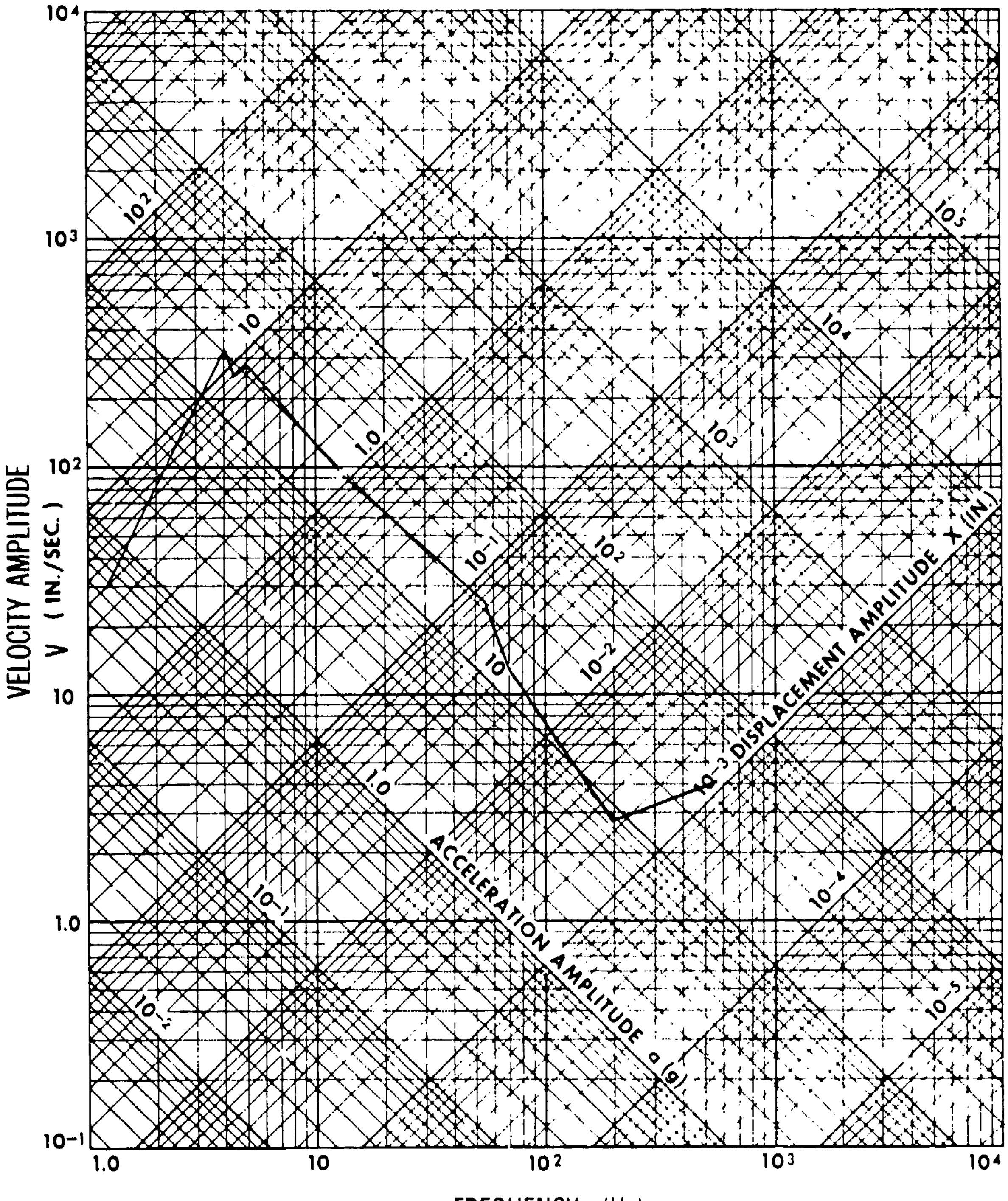



P02PS
P03PS
Lateral Direction

ITEM: Fan, Axial
REFERENCE: B-17
MANUFACTURER: American Air Flow Co.
P/N: HVO-3300-1725-H49FC
STATISTICS: Weight: 1,050 lb; Size: 48 x 60 x 45 in.
AXIS IDENTIFICATION: x - longitudinal (along direction of drive shaft);
y - vertical; z - transverse
NOTE: Test specimen experienced no function degradation during testing but sustained some structural damage: belt guard housing bent, leg cross braces borken, transition duct cracked, interference between blades and housing.

NAVFAC / NCEL

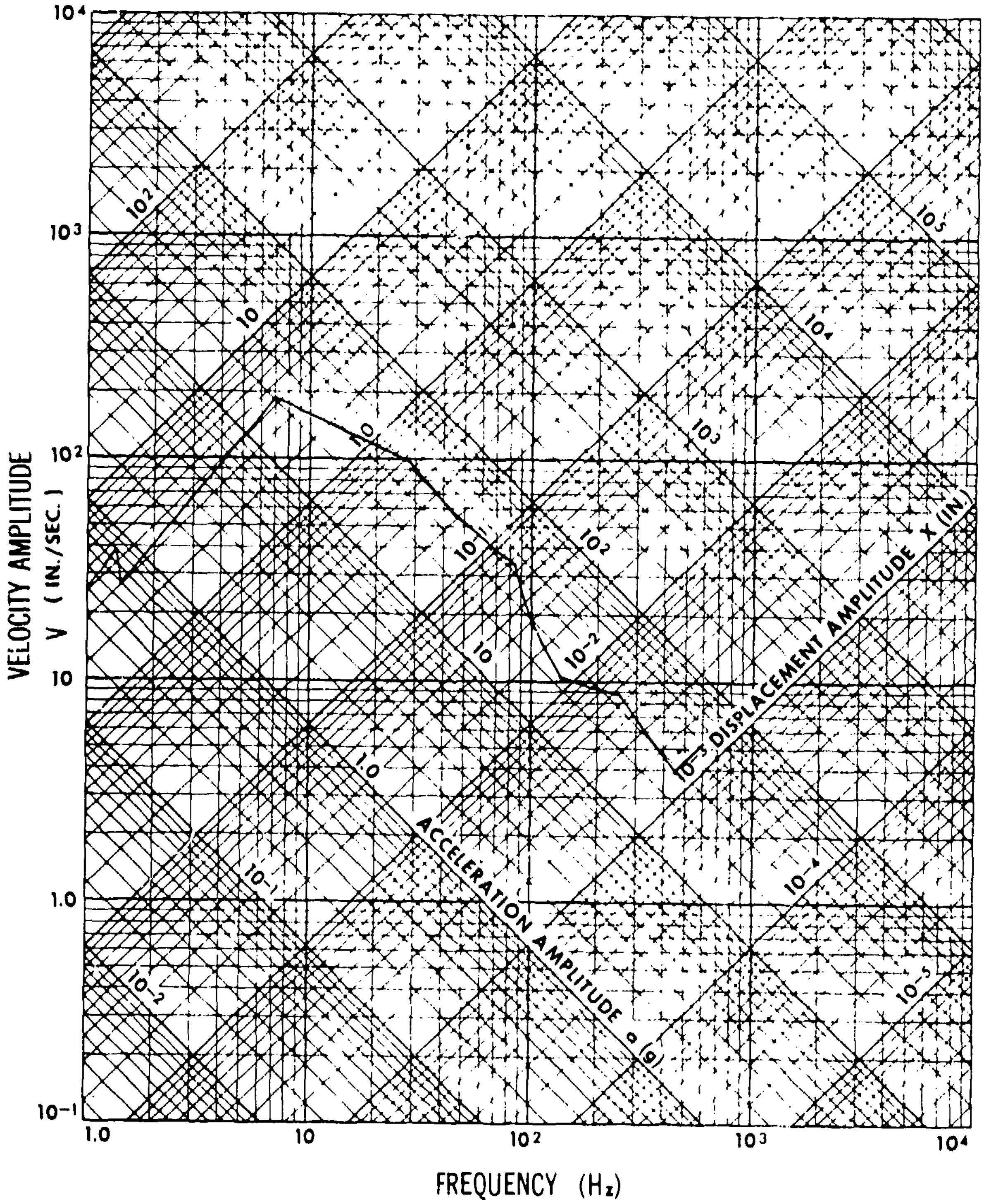
SHOCK DATA ANALYSIS




H49FC
X-Axis

NAVFAC / NCEL

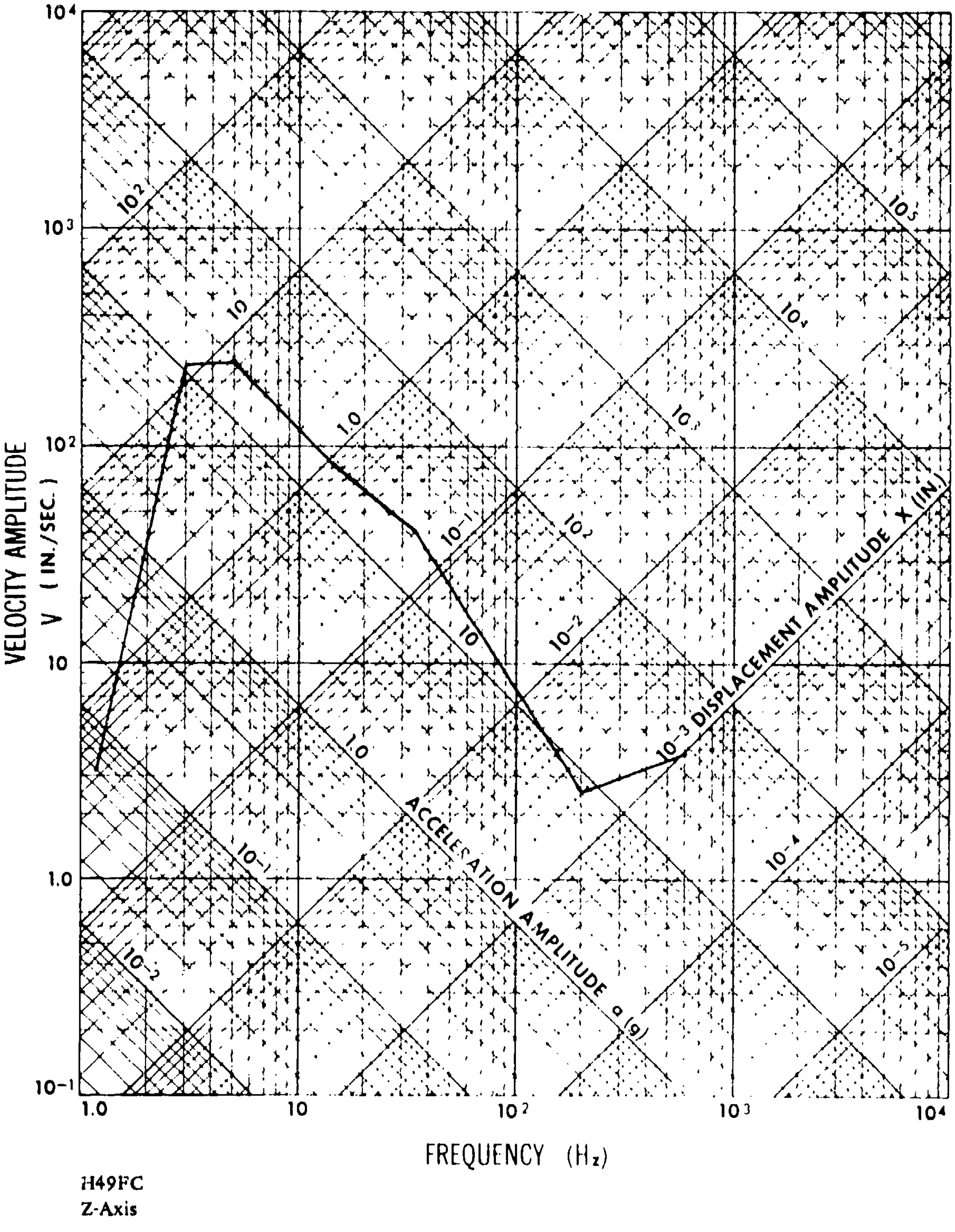
SHOCK DATA ANALYSIS




H49FC
Y-Axis

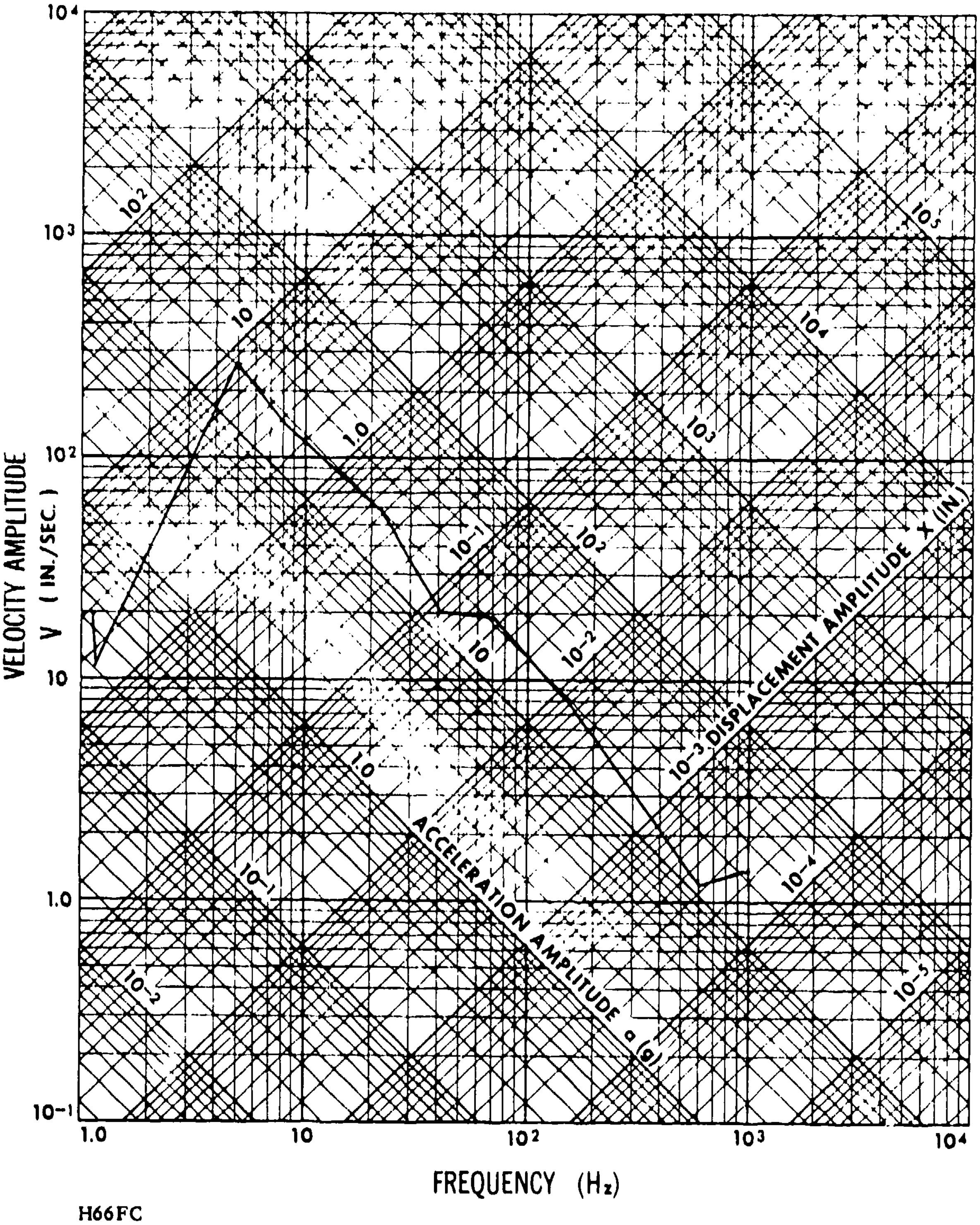
NAVFAC / NCEL

SHOCK DATA ANALYSIS

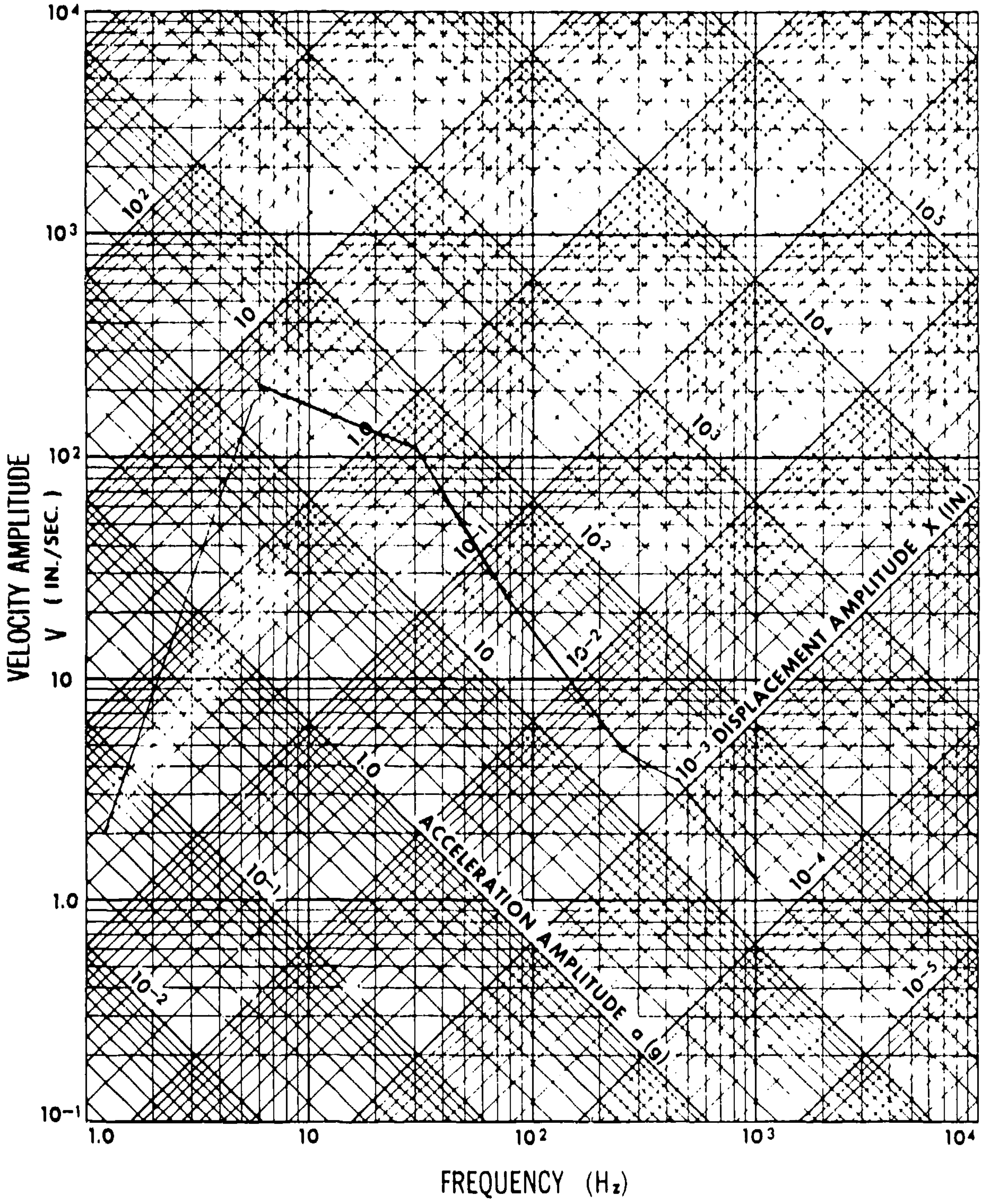



H49FC
Z-Axis

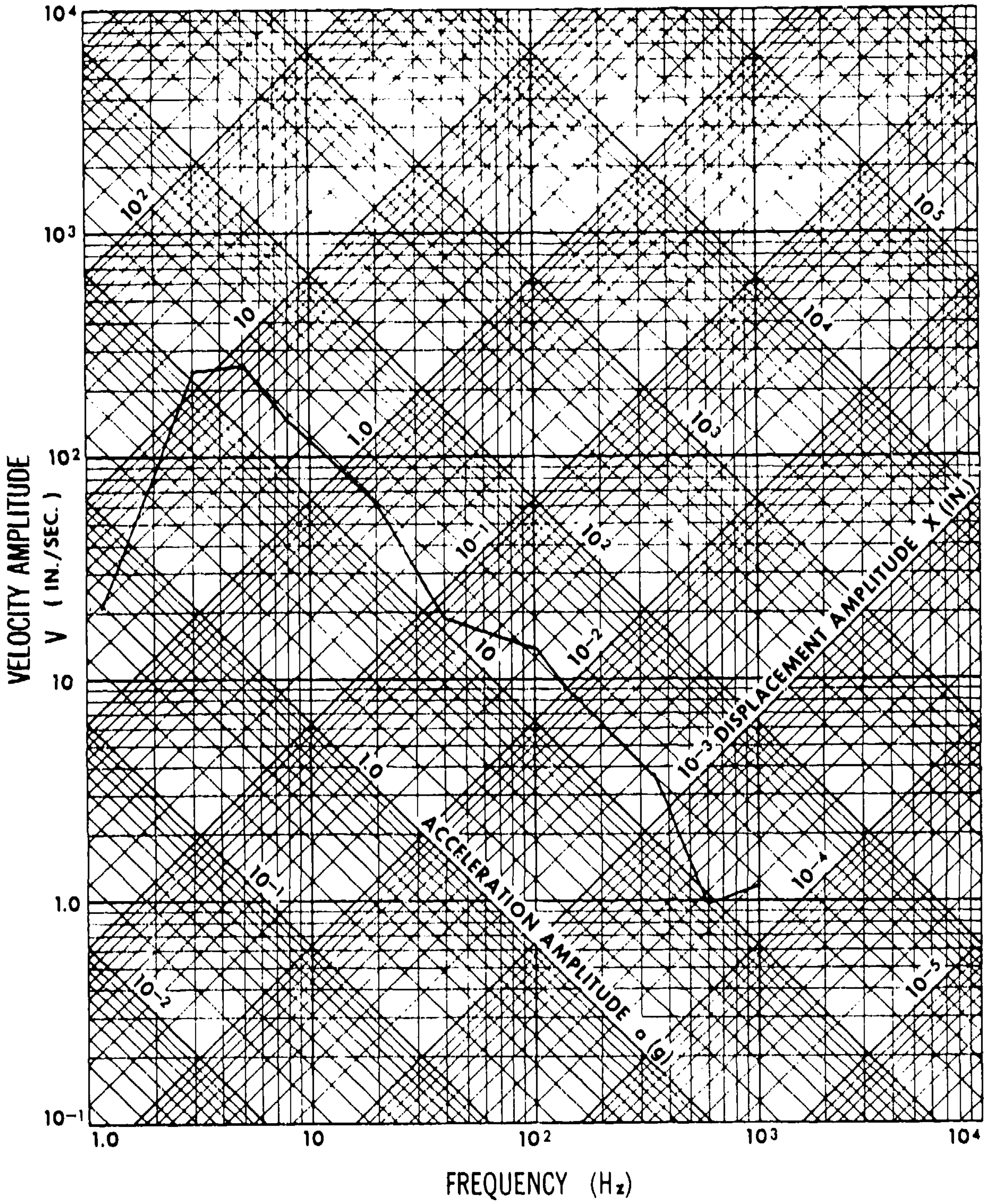
ITEM: Fan, Centrifugal
REFERENCE: B-17
DESCRIPTION: 8,400 cfm
MANUFACTURER: Carrier Transicold Co.
P/N: 27CC212
STATISTICS: Weight: 900 lb; Size: 60 x 48 x 40 in.
AXIS IDENTIFICATION: x - transverse (direction of belt travel); y - vertical; z - longitudinal (along length of drive shafts)




H66FC
X-Axis (Transverse)




H66FC
X-Y Axis Control Accel. on Y-Axis (Vertical)

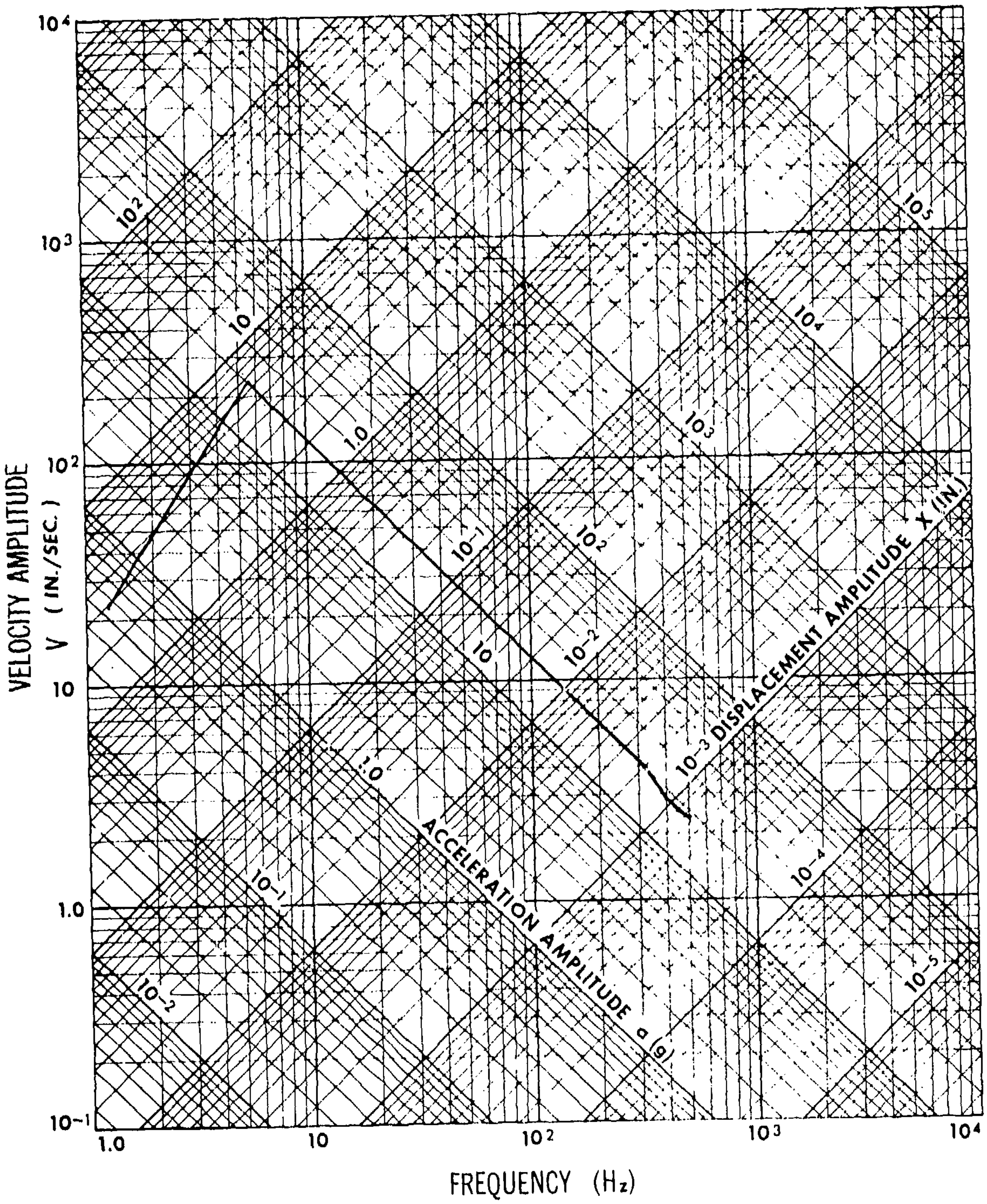

10⁴
10³
10²
10
1.0
10⁻¹
1.0
10
10²
10³
10⁴
VELOCITY AMPLITUDE
V (IN./SEC.)
FREQUENCY (Hz)
ACCELERATION AMPLITUDE a (g)
DISPLACEMENT AMPLITUDE X (IN.)

H66FC
Z-Axis

ITEM: Fan, Centrifugal, Centrifugal Axial Fan
REFERENCE: B-17
DESCRIPTION: 1 hp, variable inlet blades, 150 cfm
MANUFACTURER: Carrier Corp., motor manufactured by Reliance Corp.
P/N: 506D (motor PB 56)
STATISTICS: Weight: 110 lb; Size: 34 x 35 x 30 in.
AXIS IDENTIFICATION: x - longitudinal (along shaft); y - vertical;
z - transverse

NAVFAC / NCEL

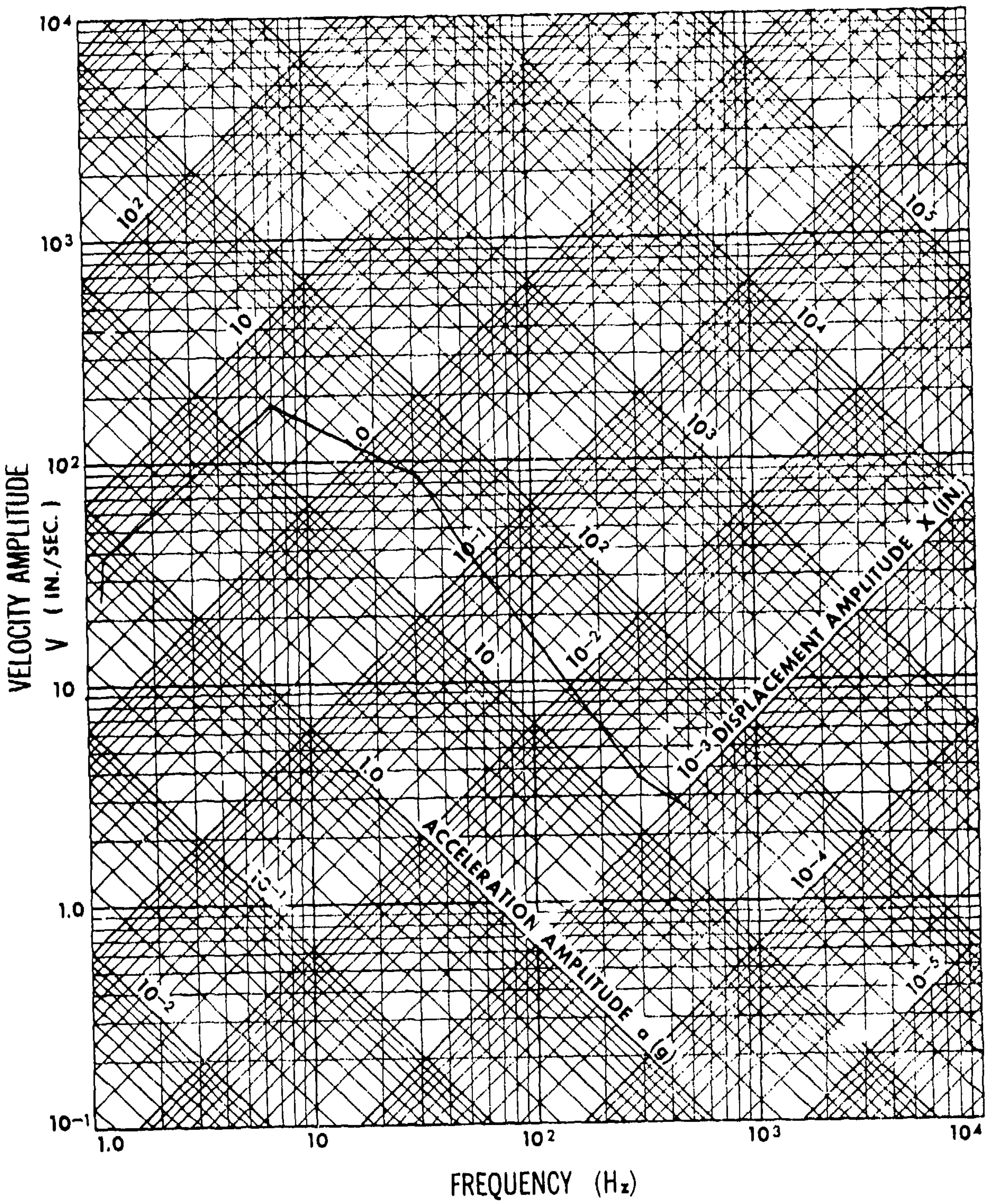
SHOCK DATA ANALYSIS



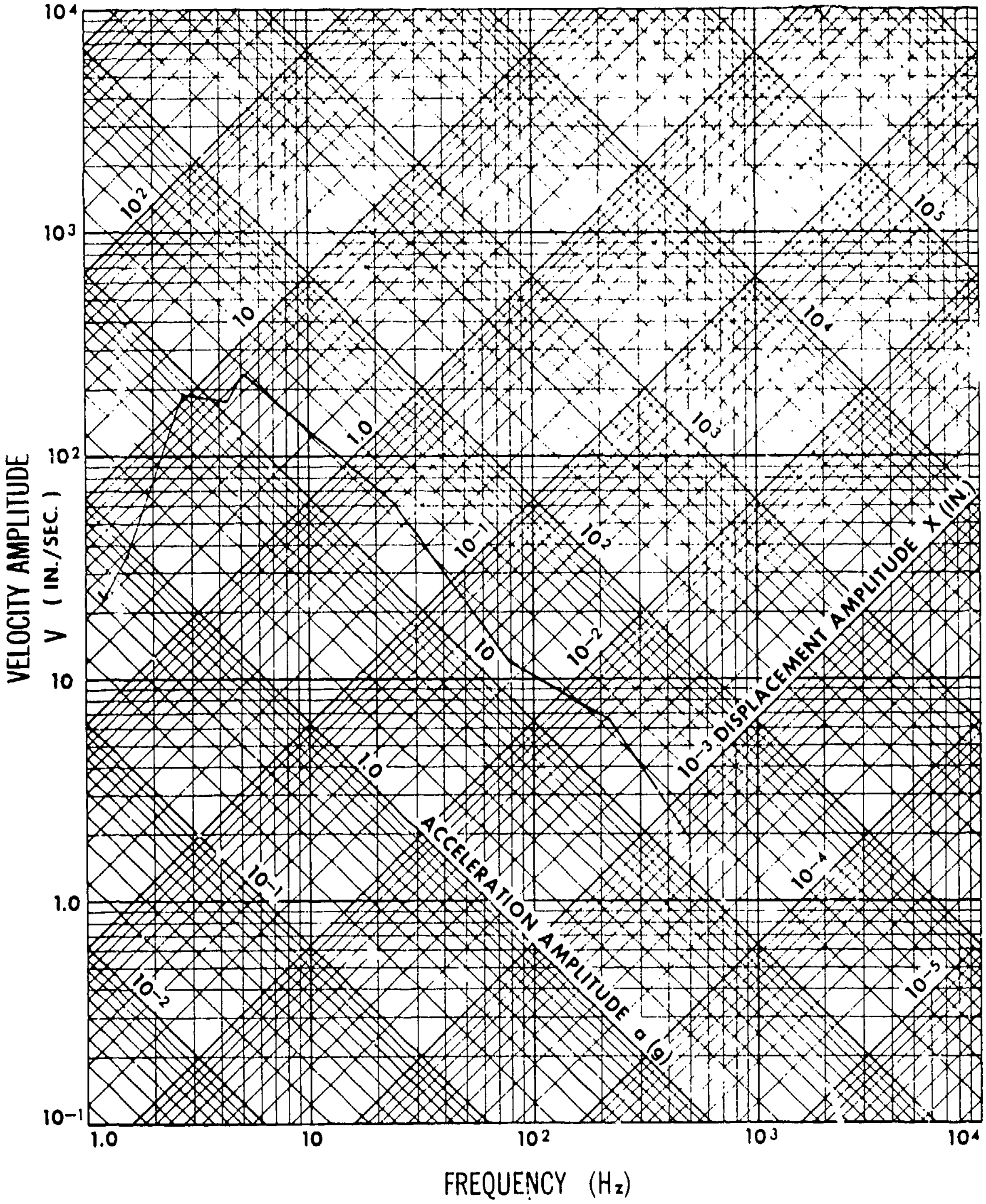

H45FD
X-Axis

**NAVFAC / NCEL**

**SHOCK DATA ANALYSIS**




H45FD
Y-Axis

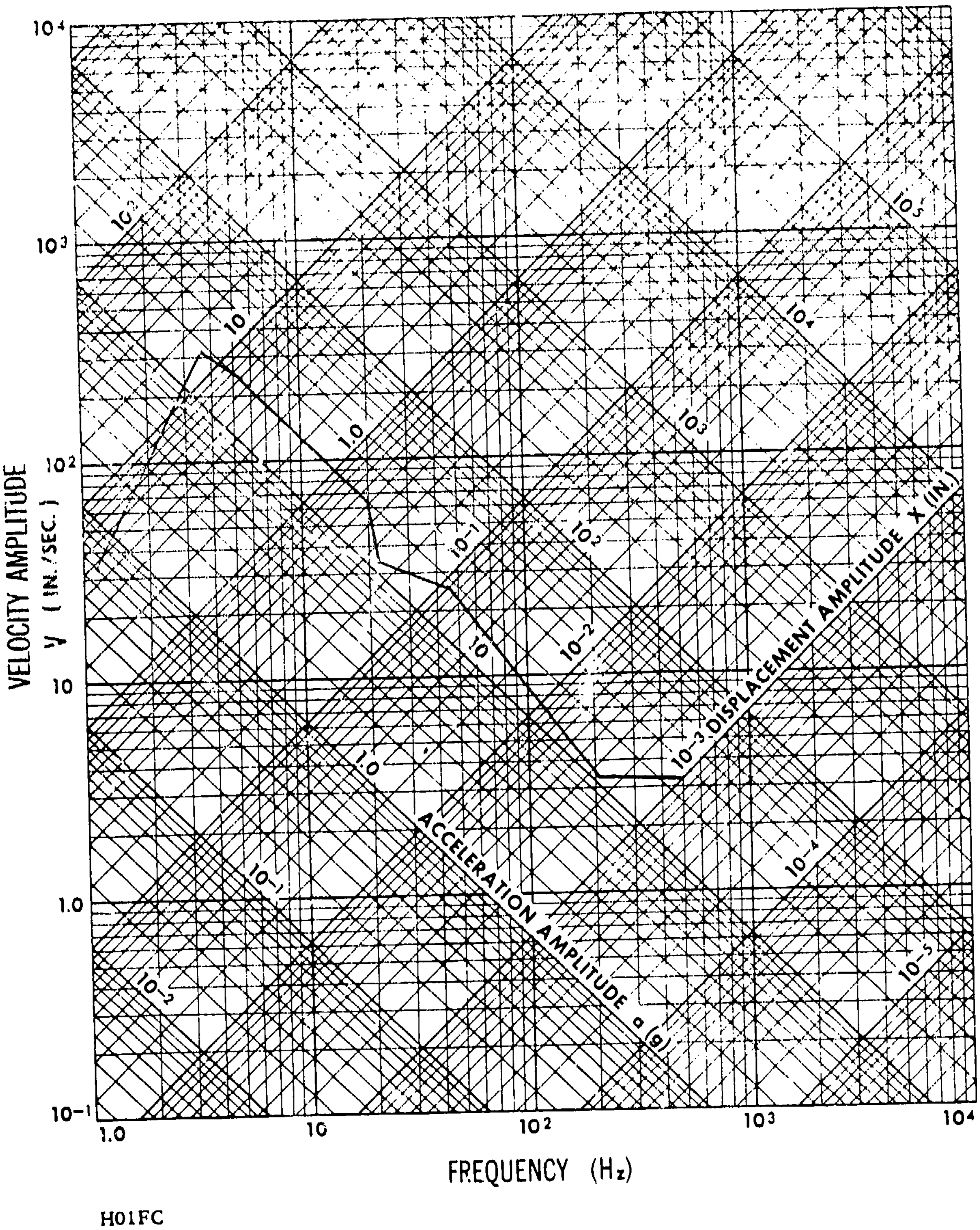

VELOCITY AMPLITUDE
V (IN./SEC.)
FREQUENCY (Hz)
ACCELERATION AMPLITUDE a (g)
DISPLACEMENT AMPLITUDE X (IN.)
10⁴
10³
10²
10
1.0
10⁻¹
1.0
10
10²
10³
10⁴

H45FD
Z-Axis

ITEM: Fan, Centrifugal
REFERENCE: B-17
DESCRIPTION: 40 hp, 460 V, 3 phase, 26,000 cfm
MANUFACTURER: Carrier Transicold Co.
P/N: 27CC490
STATISTICS: Weight: 2,050 lb; Size: 108 x 92 x 72 in.
AXIS IDENTIFICATION: x - transverse; y - vertical; z - longitudinal (in direction of motor drive shaft)

NOTE: Test specimen experienced no functional degradation during testing but was damaged slightly due to test table attachment, which was different than in-service attachment. Damage was repaired and test proceeded normally.

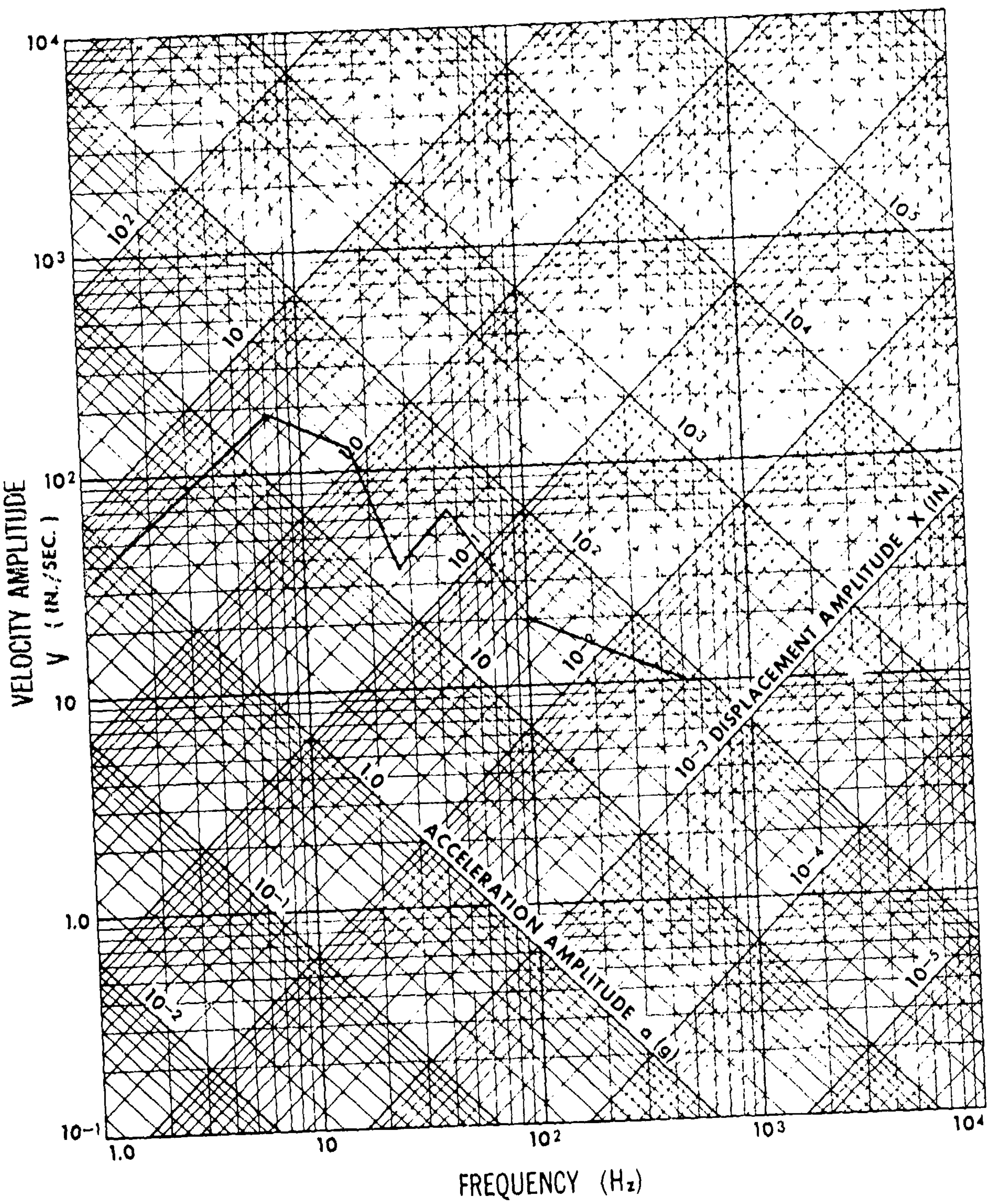
## NAVFAC / NCEL
## SHOCK DATA ANALYSIS




H01FC
X-Axis

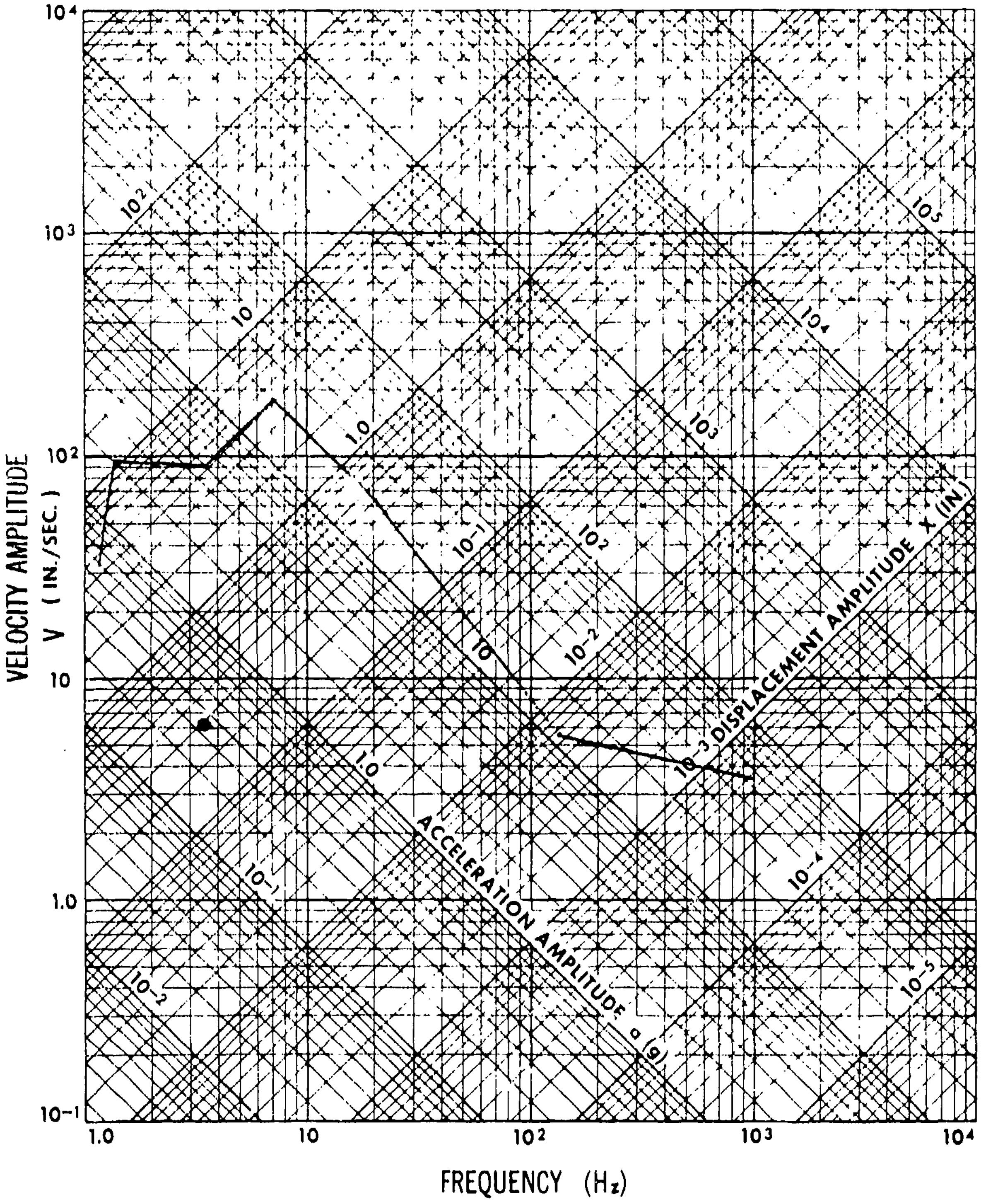
**NAVFAC / NCEL**

**SHOCK DATA ANALYSIS**




H01FC
Y-Axis

# NAVFAC / NCEL
## SHOCK DATA ANALYSIS

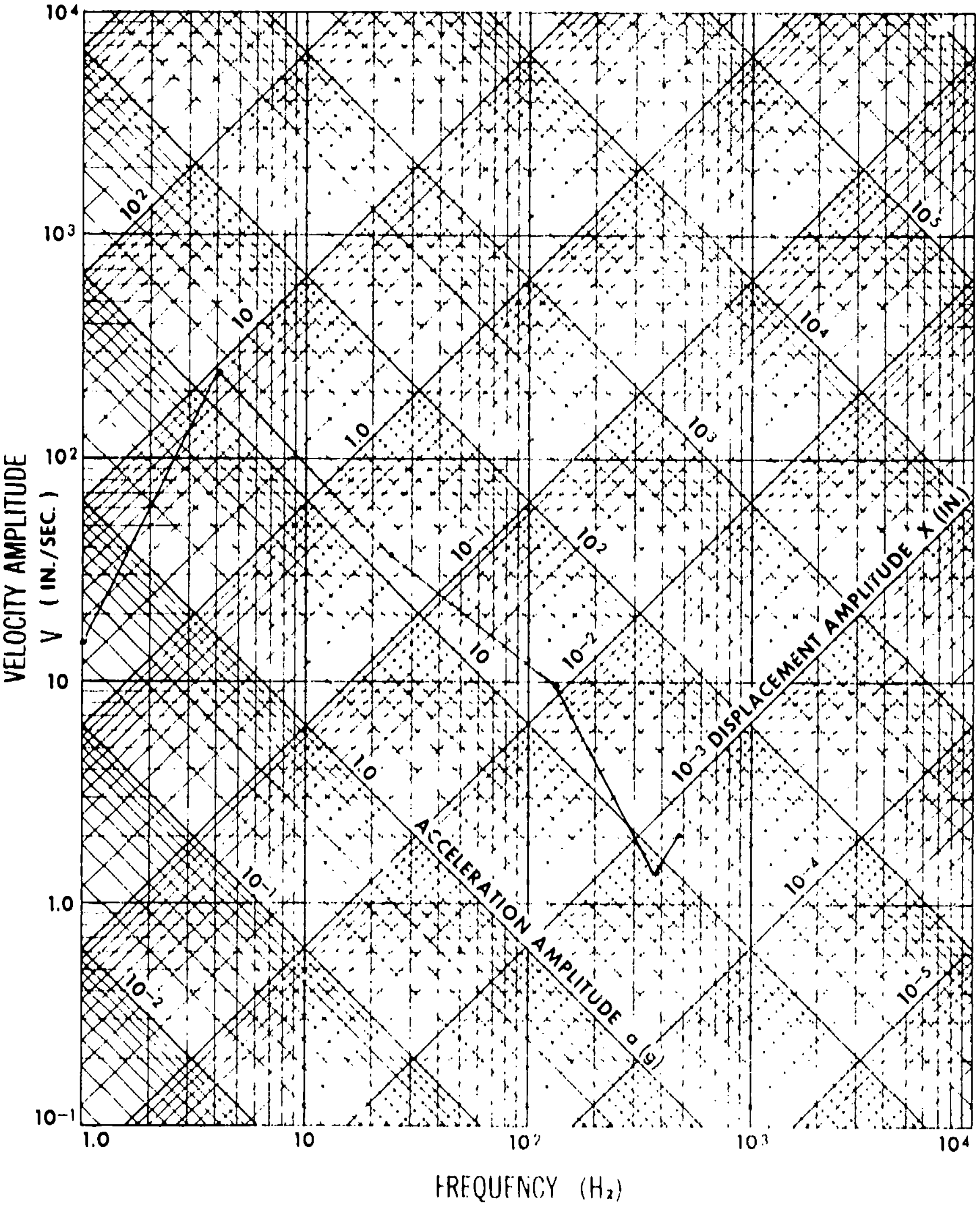



H01FC
Z-Axis

ITEM: Heat Exchanger
REFERENCE: B-18
DESCRIPTION: 312,000 Btu, 470 gpm shell side flow, 125 gpm tube side flow, 150 psi operating pressure. Chilled water makes two passes through the tube side. Water to be cooled is pumped through the shell side, which is cross-baffled so that the water flows back and forth across the tubes. The shell side provides for one pass only.
MANUFACTURER: Aqua-Chemical Inc.
P/N: 24-78DH
STATISTICS: Weight: 1,040 lb; Size: 96-1/4 x 15-3/4 x 24 in.
AXIS IDENTIFICATION: x - longitudinal (along the longest dimension); y - vertical; z - transverse
NOTE: No damage or degradation of performance was observed.

NAVFAC / NCEL

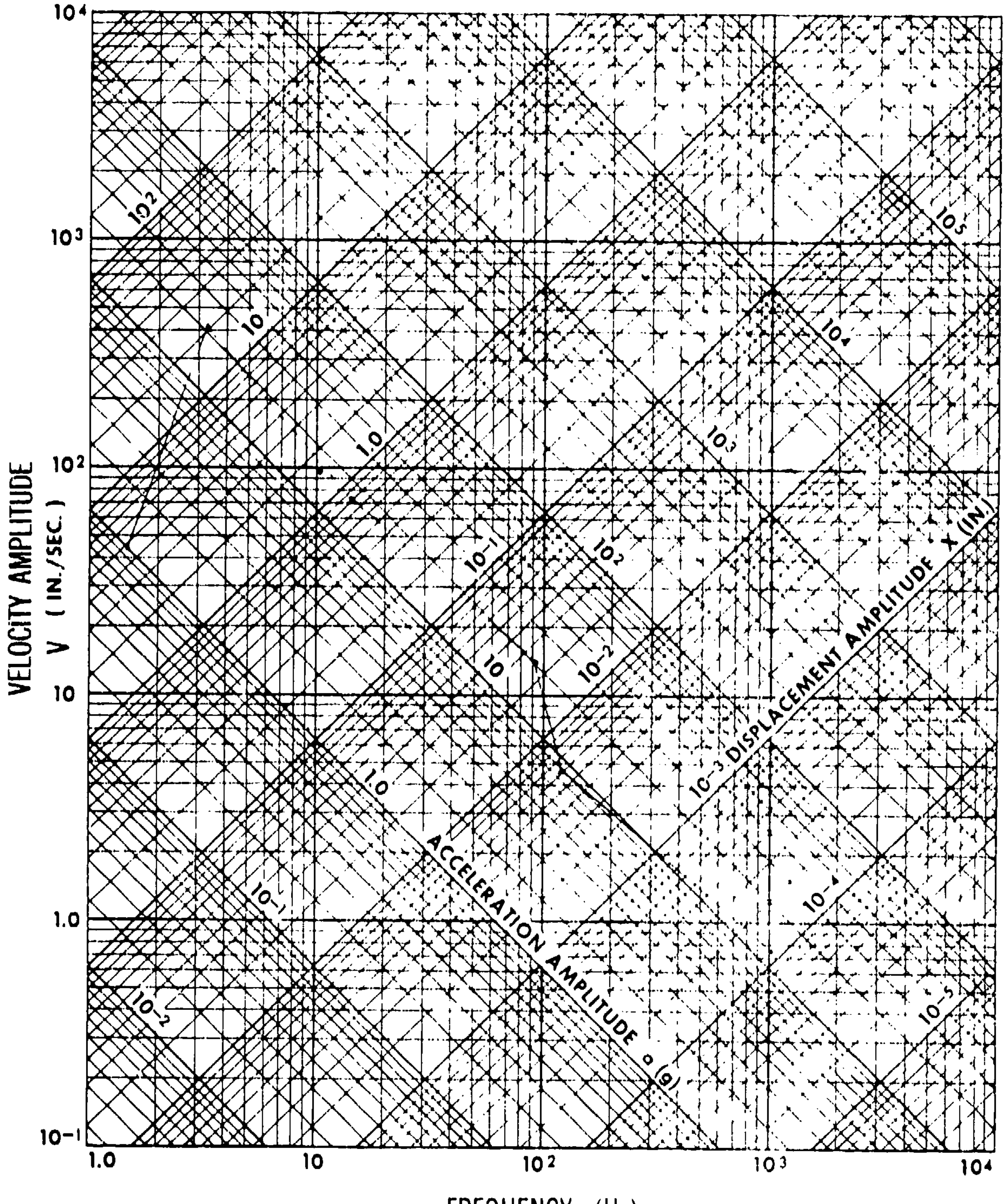
SHOCK DATA ANALYSIS




P01HE
Longitudinal Axis

NAVFAC / NCEL

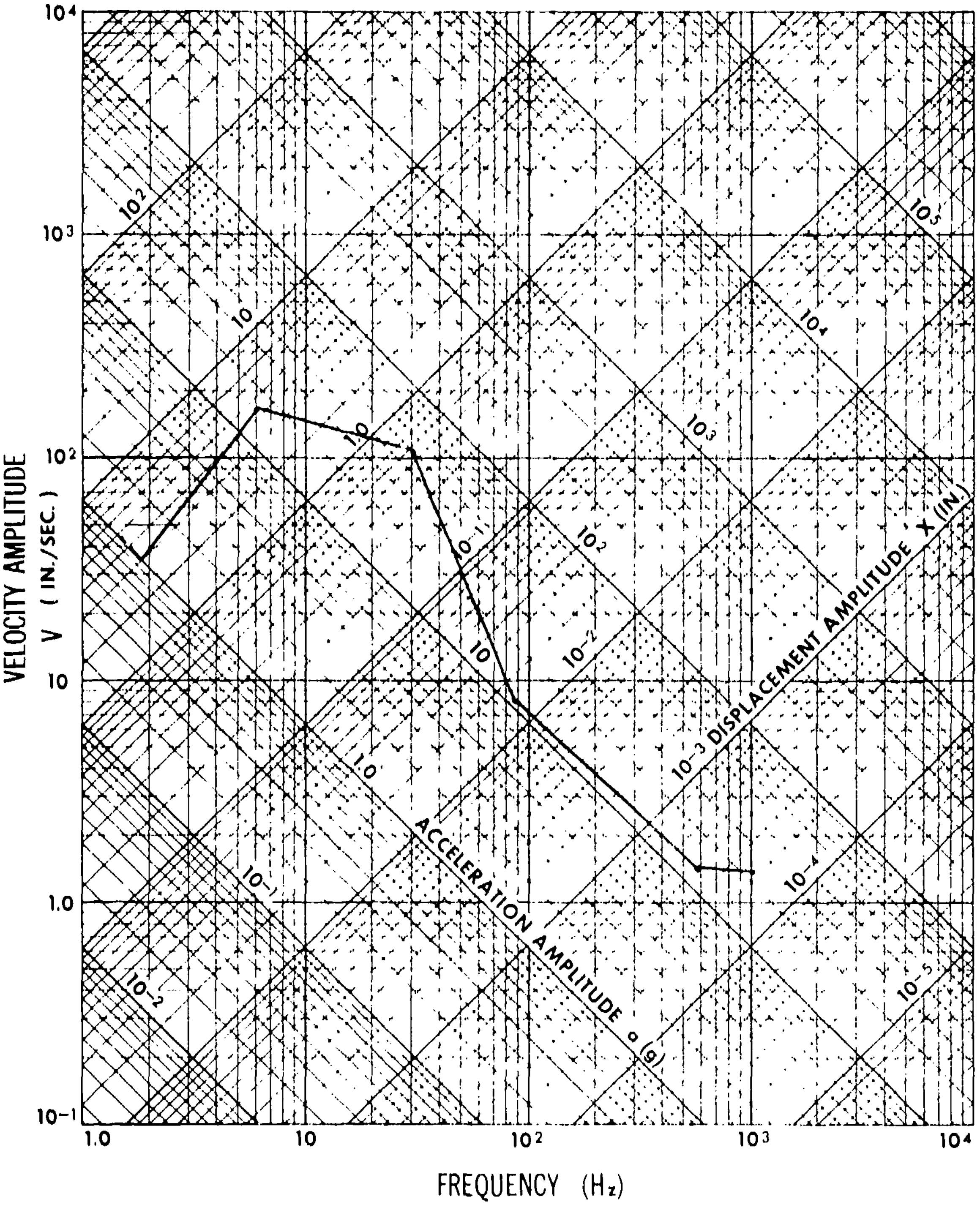
SHOCK DATA ANALYSIS




P01HE
Lateral Axis

NAVFAC / NCEL

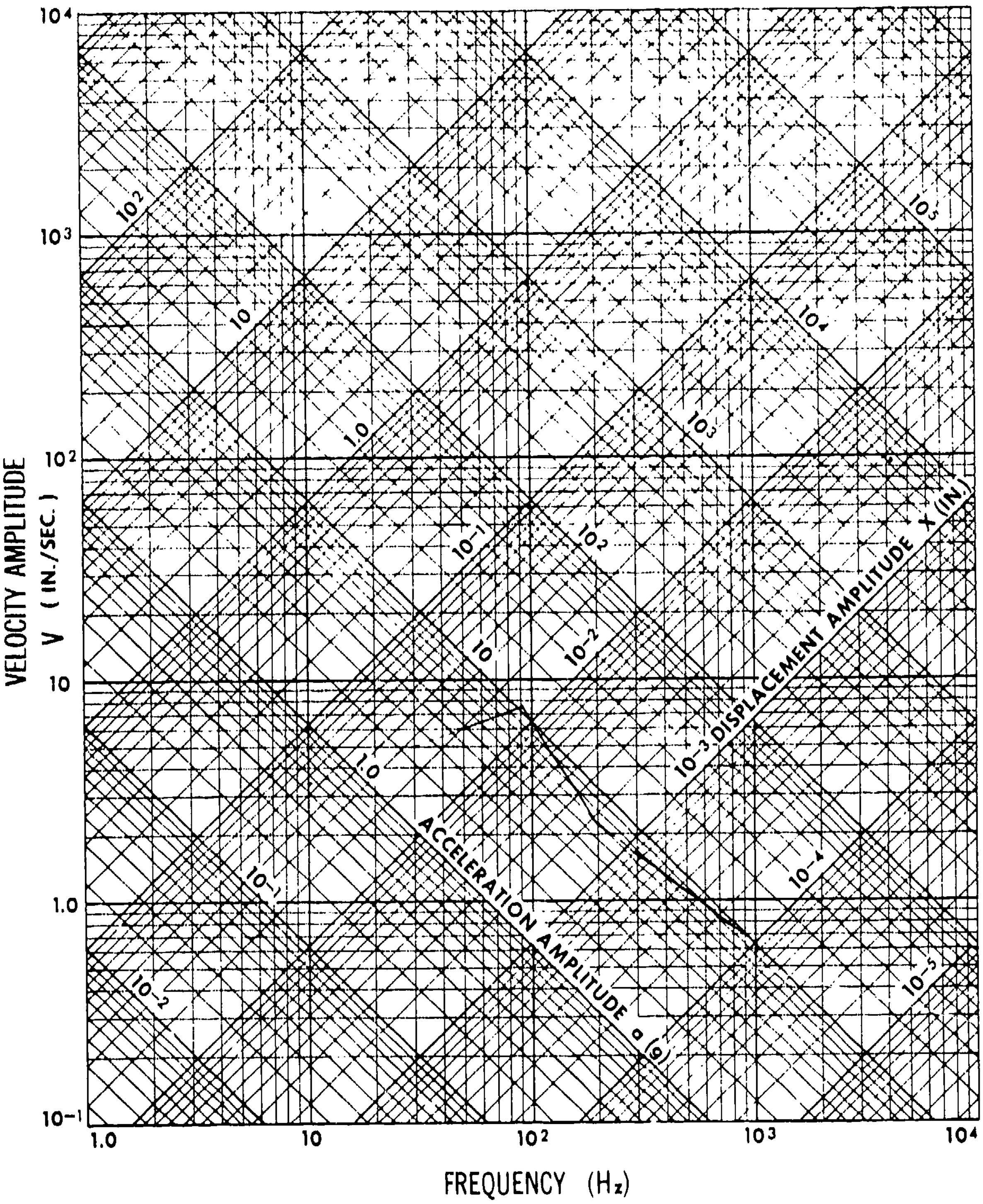
SHOCK DATA ANALYSIS




P01HE
Vertical Axis

ITEM: Air Handling Unit (part of air conditioning unit)
REFERENCE: B-20
DESCRIPTION: 1/4 hp, 1,075 rpm, 120 VAC, 1 phase, 500 cfm, 60 Hz
MANUFACTURER: Carrier Transicold Co.
P/N: 76FC4-103
STATISTICS: Weight: 115 lb; Size: 20-5/8 x 42 x 17-1/2 in.
AXIS IDENTIFICATION: x - front to rear; y - vertical; z - transverse
NOTE: Wellnut fasteners (which secure motor and blower in the air handling unit) must be replaced with bulkhead grommets using 1/4-in. washers on each side of grommets with existing 1/4-in. bolts.

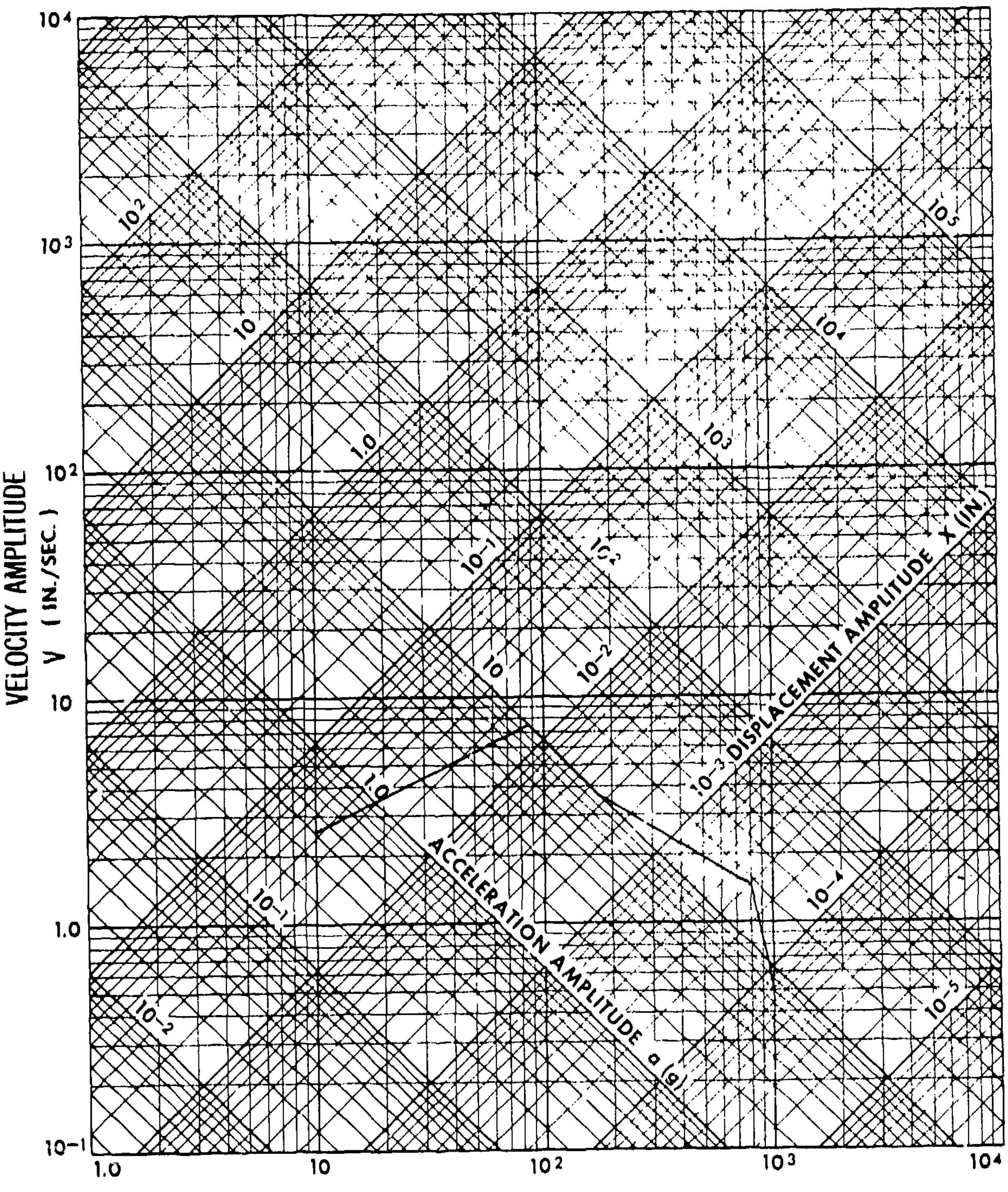
ITEM: Condensing Unit, Refrigeration (part of air conditioning unit)
DESCRIPTION: uses R12 refrigerant, 1 hp, 3,500 rpm, 10,350 Btu, 230 VAC
MANUFACTURER: Copeland Refrigeration Corp.
P/B: FSAH-0100-CAV-001
STATISTICS: Weight: 120 lb; Size: 16-7/8 x 12-7/8 x 24 in.
AXIS IDENTIFICATION: x - perpendicular to flat surface of radiator;
y - vertical; z - transverse



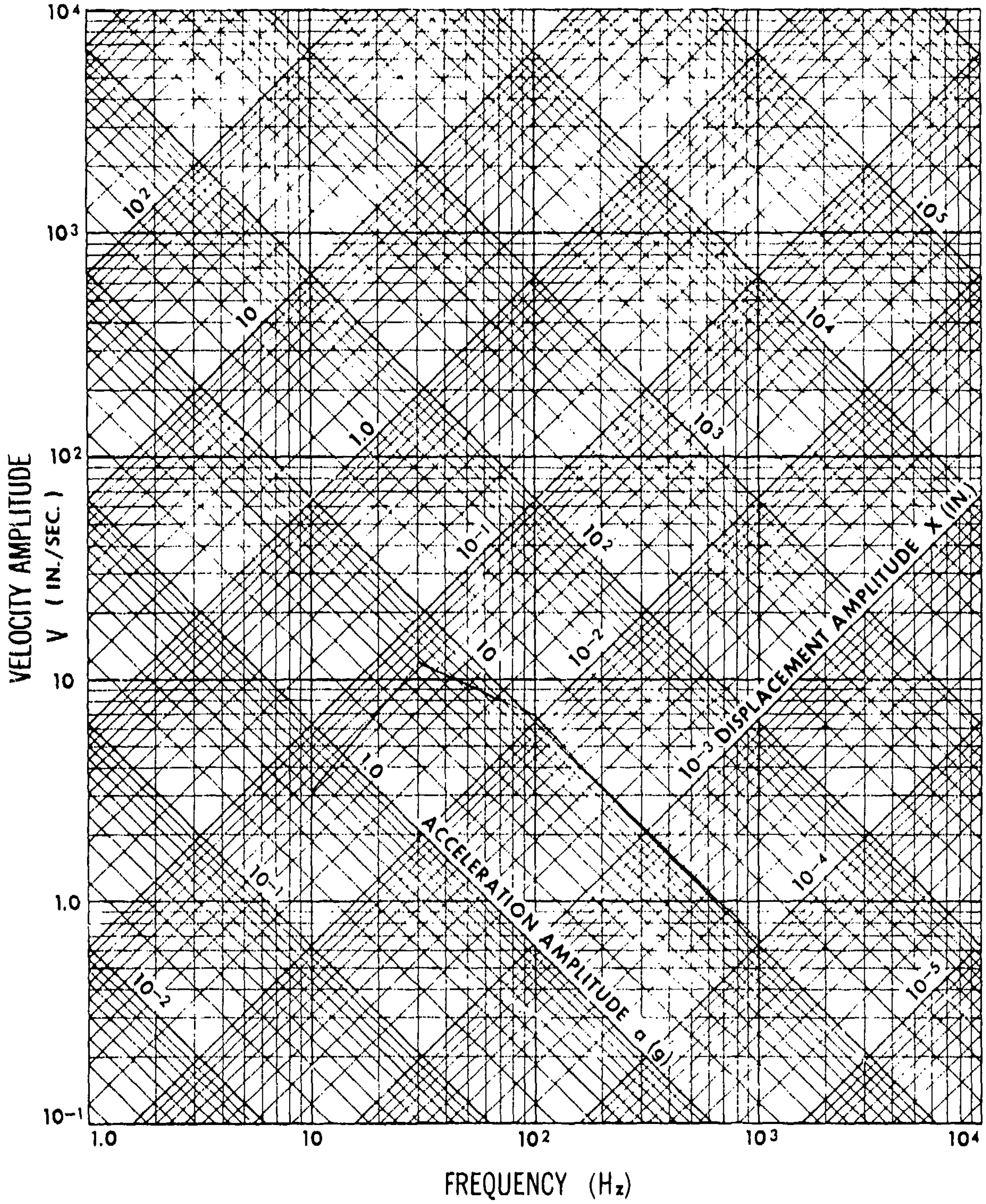

H14AU/H05CR
X-Axis

NAVFAC / NCEL

SHOCK DATA ANALYSIS




H14AU/H05CR
Y-Axis

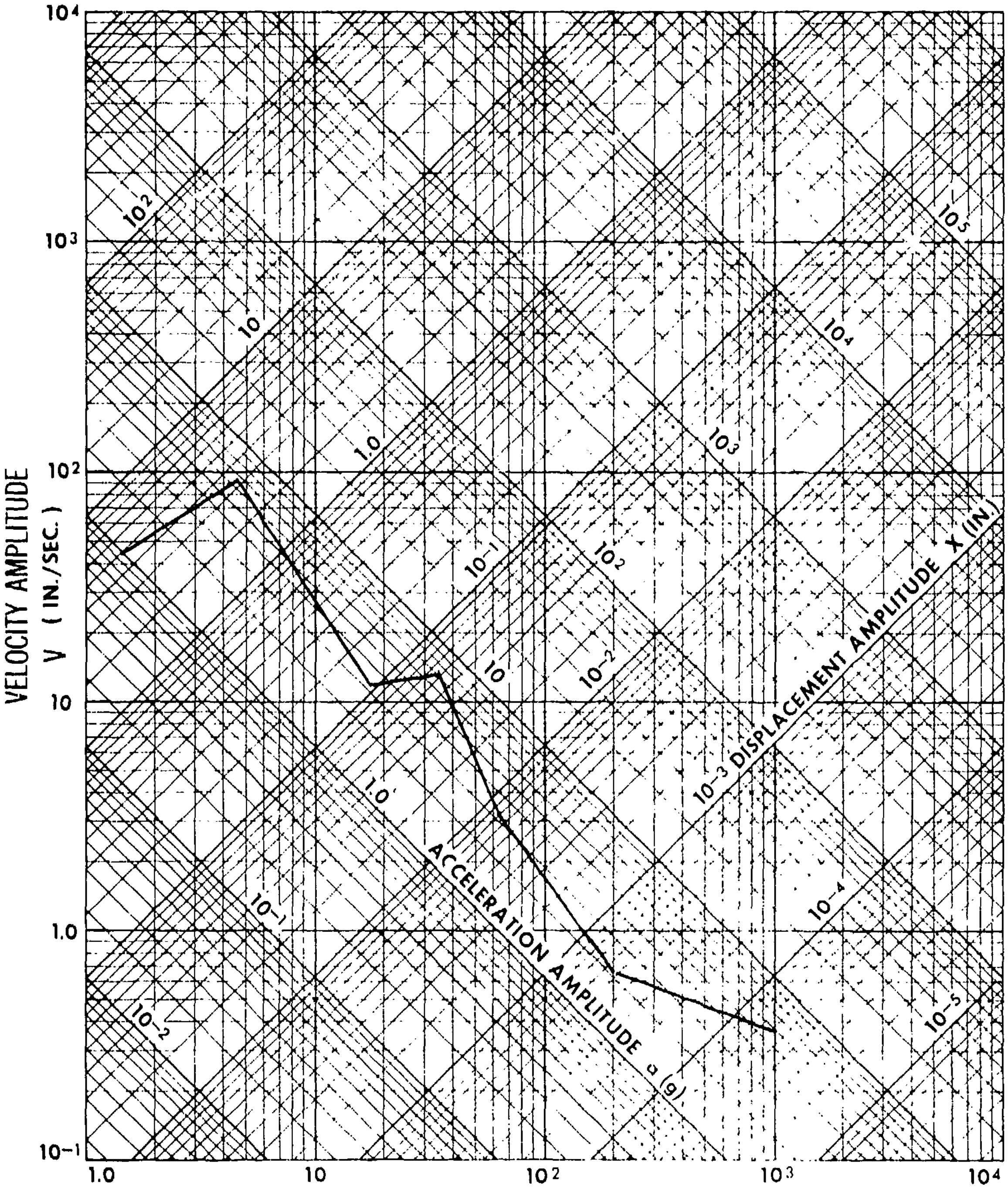



H14AU/H05CR
Z-Axis

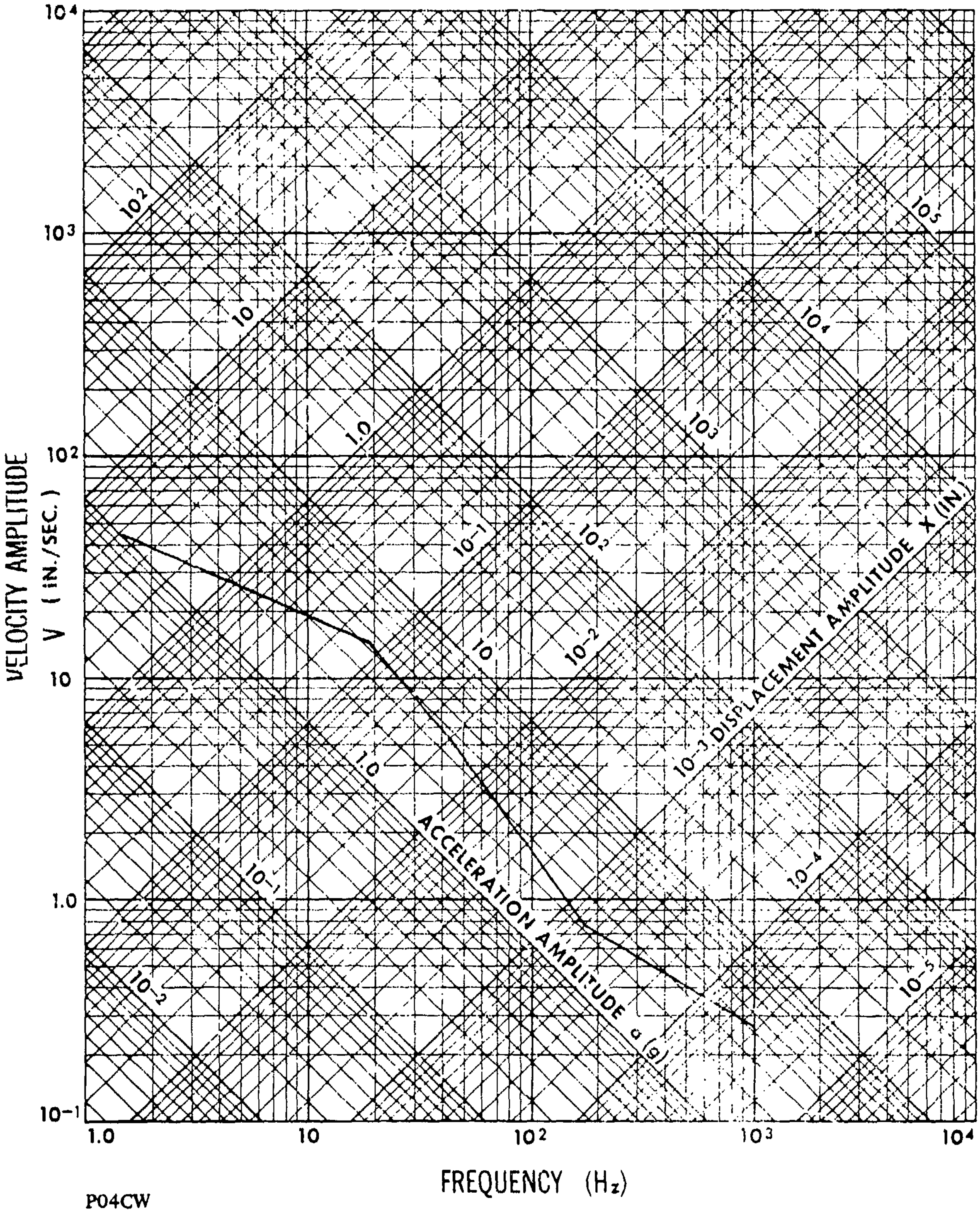
ITEM: Water Chiller
REFERENCE: B-19
DESCRIPTION: self-contained liquid chiller unit, water-cooled condensing type; cools 96 gpm from 54 to 45°F; rated at 36 refrigeration tons
MANUFACTURER: Acme Industries, Inc., Jackson, Miss.
STATISTICS: Weight: 3,025 lb; Size: 8-1/2 ft long x 3 ft wide x 5 ft high
AXIS IDENTIFICATION: longitudinal axis is along the longest dimension
NOTE: Test at the levels plotted caused a few structural failures that were corrected by welding. Several relays tripped but following the test, equipment still operated satisfactorily.

NAVFAC / NCEL

SHOCK DATA ANALYSIS



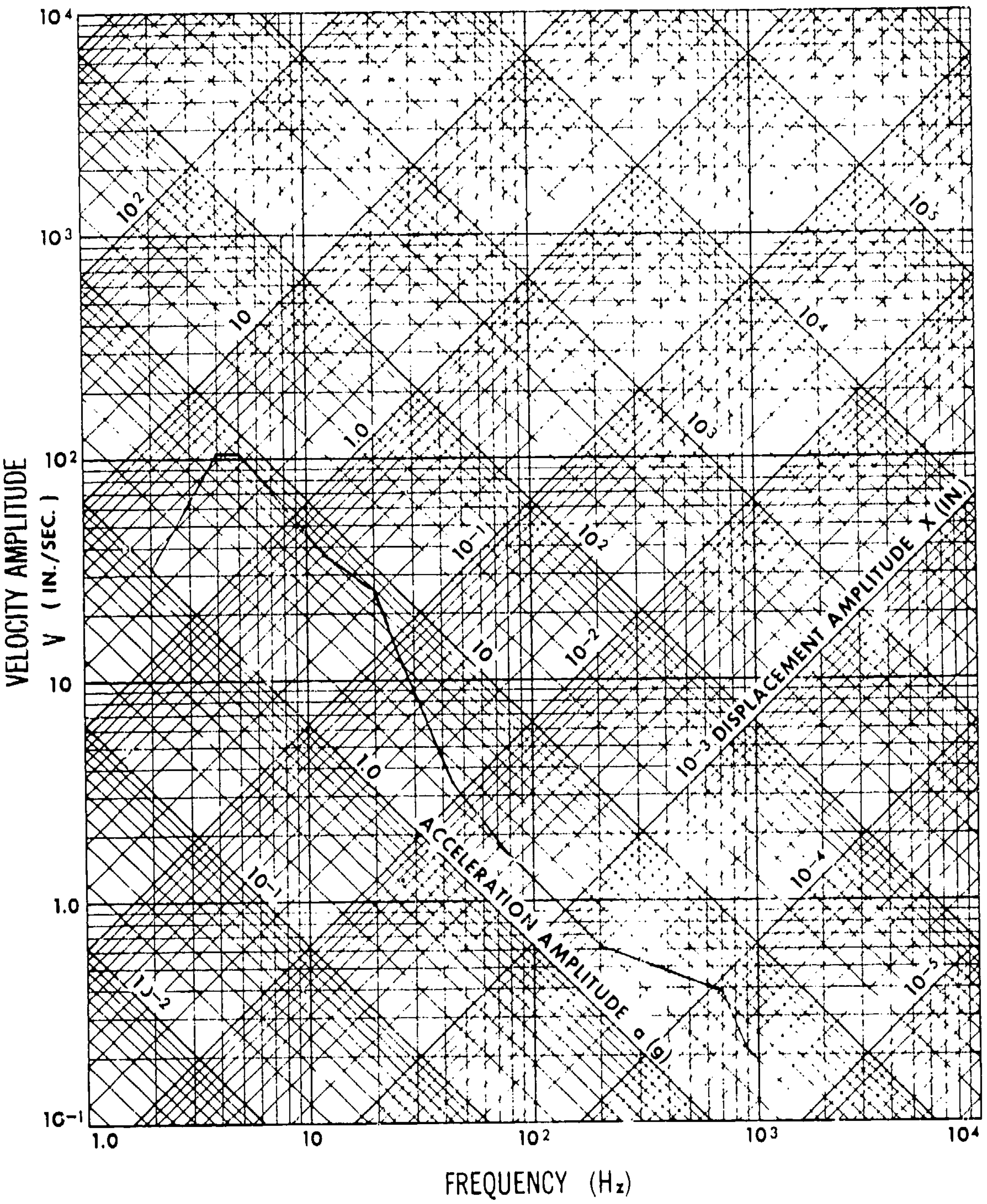

P04CW
Longitudinal




P04CW
Lateral

**NAVFAC / NCEL**

**SHOCK DATA ANALYSIS**

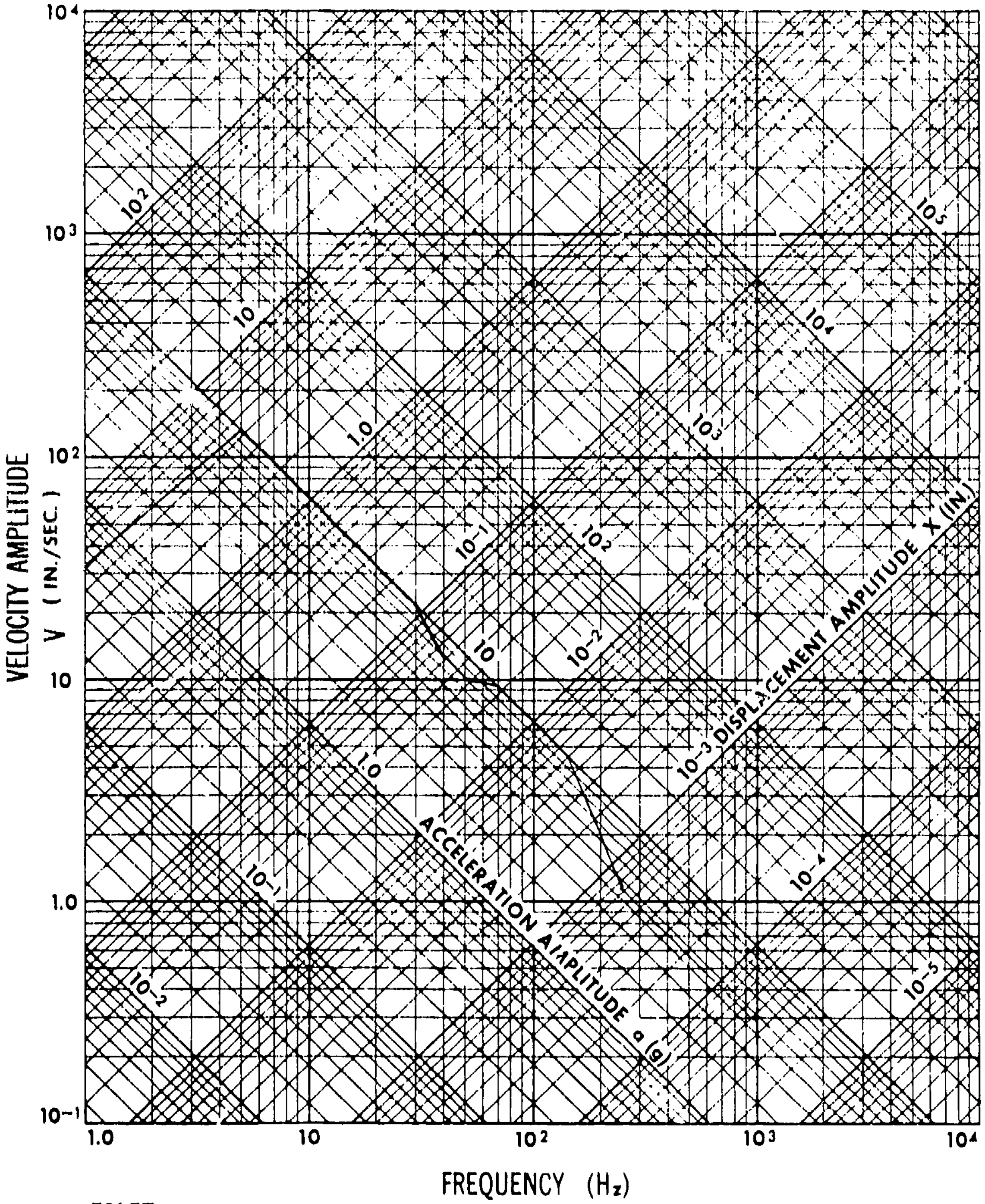



P04CW
Vertical

ITEM: Gas Turbine Generator Assembly
REFERENCE: B-21
DESCRIPTION: 400 kW, 277/480 V, 3 phase, 60 Hz, 1,800 rpm, includes gas turbine, 24:1 gear reducer, generator
MANUFACTURER: Airesearch Co. and Electric Machine Co.
P/N: Model 831-500 (turbine), Model GBS 831500 (gear reducer)
STATISTICS: Weight: 6,000 lb; Size: 11 ft 9 in. long x 4 ft wide x 4 ft high
AXIS IDENTIFICATION: x - transverse; y - longitudinal (along shaft); z - vertical

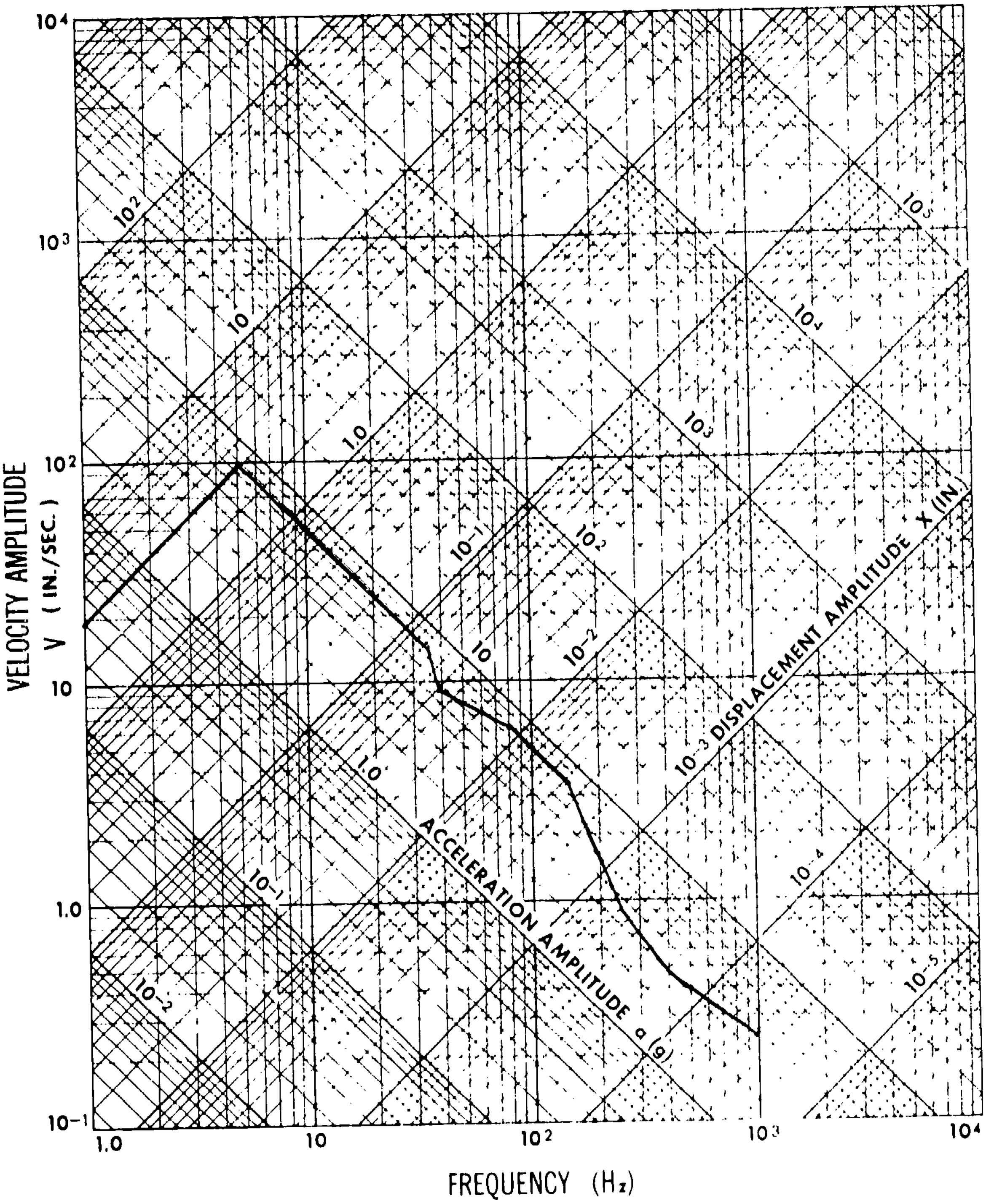
NAVFAC / NCEL

SHOCK DATA ANALYSIS




E01GT
X-Axis

## NAVFAC / NCEL
## SHOCK DATA ANALYSIS

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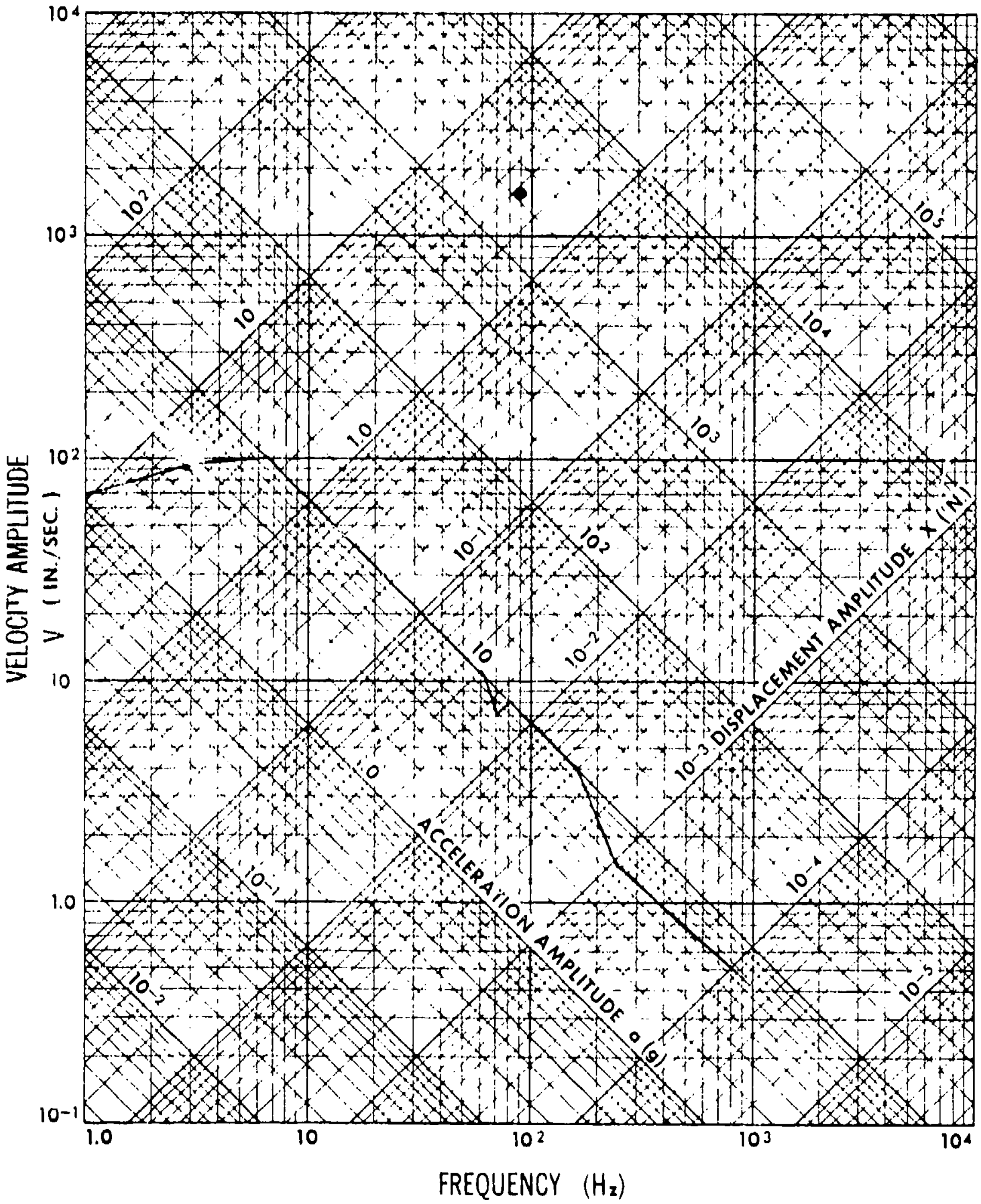
SHOCK DATA ANALYSIS

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B-1. U.S. Army Corps of Engineers. HNDTR-75-22-ED-SR: Shock test program, dynamic analysis - Air compressor (P01CR), for SAFEGUARD TSE systems and equipment. Huntsville, Ala., May 1975.

B-2. U.S. Army Corps of Engineers. HNDTR-73-12-ED-R: Shock test program, dynamic analysis diesel engine generator, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Dec 1973.

B-3. U.S. Army Corps of Engineers. HNDSP-74-345-ED-R: Shock test program, heat sensing device assembly, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Nov 1974.

B-4. U.S. Army Corps of Engineers. HNDSP-74-342-ED-R: Shock test program, pressure control valve (P83VE), for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Nov 1974.

B-5. U.S. Army Corps of Engineers. HNDSP-74-340-ED-R: Shock test program, compressor control oil shutdown switch, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Nov 1974.

B-6. U.S. Army Corps of Engineers. HNDSP-74-323-ED-R: Shock test program, generator static exciter/regulator, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Sep 1974.

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B-9. U.S. Army Corps of Engineers. HNDSP-74-316-ED-R: Shock test program, instrument air dryer, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Sep 1974.

B-10. U.S. Army Corps of Engineers. HNDSP-74-309-ED-R: Shock test program, air compressor control panel and drive motor, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., May 1974.

B-11. U.S. Army Corps of Engineers. HNDSP-74-308-ED-R: Shock test program, 660-ton chiller components, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., May 1974.

B-12. U.S. Army Corps of Engineers. HNDSP-73-97-ED-R: Shock test program, peripheral turbine pumps, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Dec 1973.

B-13. U.S. Army Corps of Engineers. HNDSP-72-77-ED-R: Shock test program, fluorescent light fixtures, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Jul 1974.

B-14. U.S. Army Corps of Engineers. HNDSP-72-77-ED-R: Shock test program, fluorescent light fixtures, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Dec 1973.

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B-18. U.S. Army Corps of Engineers. HNDSP-73-85-ED-R: Shock test program, heat exchanger, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Apr 1973.

B-19. U.S. Army Corps of Engineers. HNDSP-73-95-ED-R: Shock test program, water chiller, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Apr 1973.

B-20. U.S. Army Corps of Engineers. HNDSP-71-58-ED-R: Shock test program, air conditioner test, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., May 1972.

B-21. U.S. Army Corps of Engineers. HNDSP-74-329-ED-R: Shock test program, gas turbo-generator assembly (E01GT), for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Dec 1974.

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| Air Handling Unit | 1 |
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| Breaker, Generator Neutral | 27 |
| Cabinet, Switchgear | 25 |
| CBR Filters | 17 |
| Chiller, 660 Ton | 35 |
| Circuit Breaker | 5 |
| Circuit Breaker | 6 |
| Circuit Breakers, Unit Substation | 9 |
| Circuit Breakers, Unit Substation | 10 |
| Coils, Heating and Cooling | 18 |
| Components, Electrical | 31 |
| Control Centers, Electric Motor | 28 |
| Control Centers, Electric Motor | 32 |
| Control Components | 20 |
| Control Components, Diesel Engine | 29 |
| Control Panel, Generator | 26 |
| Cooling Coils | 18 |
| Diesel Engine Components, Monitor and Control Components | 29 |
| Duct Mounted Equipment, Monitor and Control | 24 |
| Electric Motor Control Centers | 28 |
| Electric Motor Control Centers | 32 |
| Electrical Components | 31 |
| Filters, Air Conditioning and CBR | 17 |
| Generator | 11, 12 |
| Generator Control Panel | 26 |
| Generator, Motor Set | 36 |
| Generator Neutral Breaker | 27 |
| Heat Exchanger | 33 |
| Heating and Cooling Coils | 18 |
| Monitor and Control Components | 20 |
| Monitor and Control, Diesel Engine Components | 29 |
| Monitor and Control, Duct Mounted Equipment | 24 |
| Motor Control Center, Unit Substation | 7 |
| Motor Generator Set | 11, 12 |
| Motor Generator Set | 36 |
| Panelboards, Electrical, Test | 14 |
| Panelboards, Electrical, Test | 16 |
| Piping Appurtenances, Pump | 34 |
| Piping Segments | 2 |
| Piping Segment | 3 |
| Piping Segment | 4 |
| Piping Segments | 23 |
| Pump Piping Appurtenances | 34 |
| Purification Units, Water | 15 |

| Catalog Item | Reference No. |
|---|---|
| Relay | 30 |
| Switch, Unit Substation | 5 |
| Switchgear Cabinet | 25 |
| Switchgear, Metal Clad | 21 |
| Switchgear, Metal Clad | 22 |
| Transformer | 5 |
| Transformer, Dry | 19 |
| Transformer, Unit Substation | 8 |
| Voltage Regulator | 5 |
| Voltage Regulator, Unit Substation | 6 |
| Water Purification Units | 15 |

1. U.S. Army Corps of Engineers. HNDSP-74-325-ED-R: Shock test program, air handling unit (H06AU), for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Dec 1974.

2. U.S. Army Corps of Engineers. HNDSP-74-326-ED-R: Shock test program, three piping segments (P02PC), for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Dec 1974.

3. U.S. Army Corps of Engineers. HNDSP-74-327-ED-R: Shock test program, piping segment (P39PC), for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Dec 1974.

4. U.S. Army Corps of Engineers. HNDSP-74-328-ED-R: Shock test program, piping segment (P30PC), for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Dec 1974.

5. U.S. Army Corps of Engineers. HNDSP-74-330-ED-R: Shock test program, unit substation switch - Transformer - Voltage regulator - Circuit breaker (E04SS), for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Dec 1974.

6. U.S. Army Corps of Engineers. HNDSP-74-332-ED-R: Shock test program, unit substation voltage regulator - Circuit breaker (E05SS)-B, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Dec 1974.

7. U.S. Army Corps of Engineers. HNDSP-74-333-ED-R: Shock test program, unit substation motor control center (E12SS), for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Dec 1974.

8. U.S. Army Corps of Engineers. HNDSP-74-334-ED-R: Shock test program, unit substation transformer (E16SS), for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Dec 1974.

9. U.S. Army Corps of Engineers. HNDSP-74-335-ED-R: Shock test program, unit substation circuit breakers (E27SS), for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Dec 1974.

10. U.S. Army Corps of Engineers. HNDSP-74-336-ED-R: Shock test program, unit substation circuit breakers (E29SS), for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Dec 1974.

11. U.S. Army Corps of Engineers. HNDSP-74-337-ED-R: Shock test program, motor generator set (E03GM), for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Dec 1974.

12. U.S. Army Corps of Engineers. HNDSP-74-338-ED-R: Shock test program, motor generator set (E12GM), for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Dec 1974.

13. U.S. Army Corps of Engineers. HNDSP-72-69-ED-R: Shock test program, station battery system, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Aug 1972.

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16. U.S. Army Corps of Engineers. HNDSP-72-64-ED-R: Shock test program, electrical panelboards test, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Sep 1972.

17. U.S. Army Corps of Engineers. HNDSP-73-86-ED-R: Shock test program, air conditioning and CBR filters, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Apr 1973.

18. U.S. Army Corps of Engineers. HNDSP-73-89-ED-R: Shock test program, heating and cooling coils, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., May 1973.

19. U.S. Army Corps of Engineers. HNDSP-73-300-ED-R: Shock test program, dry transformers, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Dec 1973.

20. U.S. Army Corps of Engineers. HNDSP-73-302-ED-R: Shock test program, monitoring and control components, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Dec 1973.

21. U.S. Army Corps of Engineers. HNDSP-74-305-ED-R: Shock test program, metal-clad switchgear, for SAFEGUARD TSE systems and equipment, Vol 1, Book 1 - Basic report and 5HK75 test data. Huntsville, Ala., Dec 1973.

22. U.S. Army Corps of Engineers. HNDSP-73-305-ED-R: Shock test program, metal-clad switchgear, for SAFEGUARD TSE systems and equipment, Vol 1, Book 2 - 5HK350 test data. Huntsville, Ala., Dec 1973.

23. U.S. Army Corps of Engineers. HNDSP-74-306-ED-R: Shock test program, piping segments, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Mar 1974.

24. U.S. Army Corps of Engineers. HNDSP-74-307-ED-R: Shock test program, monitor and control - Duct mounted equipment, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Apr 1974.

25. U.S. Army Corps of Engineers. HNDSP-73-91-ED-R: Shock test program, switchgear cabinet - Transfer function tests, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Apr 1973.

26. U.S. Army Corps of Engineers. HNDSP-74-310-ED-R: Shock test program, generator control panel, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Jun 1974.

27. U.S. Army Corps of Engineers. HNDSP-74-312-ED-R: Shock test program, generator neutral breaker, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Jul 1974.

28. U.S. Army Corps of Engineers. HNDSP-74-315-ED-R: Shock test program, electric motor control centers (E52MC) (E87MC), for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Jul 1974.

29. U.S. Army Corps of Engineers. HNDSP-74-324-ED-R: Shock test program, diesel engine components and M&C components, for SAFEGUARD TSE systems and equipment, Vol 1, Test report. Huntsville, Ala., Dec 1974.

30. U.S. Army Corps of Engineers. HNDSP-71-57-ED-R: Shock test program, relay - Fragility test, for SAFEGUARD TSE systems and equipment, Vol 1, Book 2 - 5HK350 test data. Huntsville, Ala., May 1972.

31. U.S. Army Corps of Engineers. HNDSP-75-349-ED-R: Shock test program, fragility tests of twenty-two electrical components, for SAFEGUARD TSE systems and equipment, Vol 1 and 2. Huntsville, Ala., Jul 1975.

32. U.S. Army Corps of Engineers. HNDSP-73-159-ED-R: Hardness program - Non-EMP, test report, for SAFEGUARD TSE ground facilities shock test program, fragility testing electric motor control centers (ITCs E06MC, E52MC, E87MC, and E89MC), Vol 1 and 2. Huntsville, Ala., Apr 1974.

33. U.S Army Corps of Engineers. HNDSP-73-158-ED-R: Hardness program - Non-EMP, heat exchanger dynamic response analysis, for SAFEGUARD TSE ground facilities. Huntsville, Ala., Mar 1973.

34. U.S. Army Corps of Engineers. HNDSP-73-160-ED-R: Hardness program - Non-EMP, pump piping appurtenance, shock response analysis, for SAFEGUARD TSE ground facilities. Huntsville, Ala., Mar 1973.

35. U.S. Army Corps of Engineers. HNDTR-77-30-ED-SR: Shock test program, dynamic analysis of a 660-ton chiller, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Apr 1977.

36. U.S. Army Corps of Engineers. HNDSP-74-344-ED-R: Shock test program, dynamic analysis - Motor generator set - ITC-E12GM, for SAFEGUARD TSE systems and equipment. Huntsville, Ala., Dec 1974.

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NAVSTA CO Roosevelt Roads P.R. Puerto Rico; CO, Brooklyn NY; Code 4, 12 Marine Corps Dist, Treasure Is., San Francisco CA; Dir Engr Div, PWD, Mayport FL; Dir Mech Engr 37WC93 Norfolk, VA; Engr. Dir., Rota Spain; Long Beach, CA; Maint. Cont. Div., Guantanamo Bay Cuba; Maint. Div. Dir/Code 531, Rodman Panama Canal; PWD (LTJG.P.M. Motolenich), Puerto Rico; PWD - Engr Dept, Adak, AK; PWD - Engr Div, Midway Is.; PWO Pearl Harbor, HI; PWO, Keflavik Iceland; PWO, Mayport FL; ROICC, Rota Spain; SCE, Guam; SCE, San Diego CA; SCE, Subic Bay, R.P.; Utilities Engr Off. Rota Spain; Wpns Offr, Roosevelt Roads, PR
NAVSUBASE ENS S. Dove, Groton, CT; SCE, Pearl Harbor HI
NAVSUPPACT CO, Naples, Italy; PWO Naples Italy; PWO, Seattle, WA; Plan Engr Div., Naples Italy
NAVSUPPFAC PWD - Maint. Control Div, Thurmont, MD
NAVSURFWPNCEN PWO, White Oak, Silver Spring, MD
NAVTECHTRACEN SCE, Pensacola FL
NAVTELCOMMCOM Code 53, Washington, DC
NAVWPNCEN Code 24 (Dir Safe & Sec) China Lake, CA; Code 2636 (W. Bonner), China Lake CA; Code 3276 China Lake, CA; Code 3803 China Lake, CA; PWO (Code 266) China Lake, CA; ROICC (Code 702), China Lake CA
NAVWPNSTA (Clebak) Colts Neck, NJ; Code 09, Concord, CA; Code 092, Colts Neck NJ; Code 092, Concord CA; Code 092A (C. Fredericks) Seal Beach CA; Maint. Control Dir., Yorktown VA; Nav Wpns Hndlng Cen (Code 8052) Colts Neck NJ
NAVWPNSTA PW Office Yorktown, VA

NAVWPNSTA PWD - Maint. Control Div., Concord, CA; PWD - Supr Gen Engr, Seal Beach, CA; PWO, Charleston, SC; PWO, Seal Beach CA; Safety Br. Concord, CA; Security Offr, Colts Neck NJ; Security Offr. Concord CA
NAVWPNSUPPCEN Code 09 Crane IN
NCBU 405 OIC, San Diego, CA
NCTC Const. Elec. School, Port Hueneme, CA
NCBC Code 10 Davisville, RI; Code 15, Port Hueneme CA; Code 155, Port Hueneme CA; Code 156, Port Hueneme, CA; Code 400, Gulfport MS; Code 430 (PW Engrng) Gulfport, MS; PWO (Code 80) Port Hueneme, CA; PWO, Davisville RI; PWO, Gulfport, MS
NCBU 411 OIC, Norfolk VA
NCR 20, Code R70; 20, Commander
NMCB 74, CO; FIVE, Operations Dept; Forty, CO; THREE, Operations Off.
NORDA Code 440 (Ocean Rsch Off) Bay St. Louis MS
NRL Code 5800 Washington, DC; Code 8441 (R.A. Skop), Washington DC
NSC Code 44 (Security Officer) Oakland, CA; Code 54.1 Norfolk, VA; Security Offr, Hawaii
NSD SCE, Subic Bay, R.P.
NSWSES Code 0150 Port Hueneme, CA
NTSB Chairman, Wash DC
NUSC Code 5202 (S. Schady) New London, CT; Code EA123 (R.S. Munn), New London CT
OFFICE SECRETARY OF DEFENSE OASD (MRA&L) Dir. of Energy, Pentagon, Washington, DC
ONR Code 700F Arlington VA
PACMISRANFAC HI Area Bkg Sands, PWO Kekaha, Kauai, HI
PHIBCB 1 P&E, San Diego, CA; 1, CSWC D Wellington, San Diego, CA
PMTC EOD Mobile Unit, Point Mugu, CA; Pat. Counsel, Point Mugu CA
PWC (Lt E.S. Agonoy) Pensacola, FL; CO Norfolk, VA; CO, (Code 10), Oakland, CA; CO, Great Lakes IL; CO, Pearl Harbor HI; Code 10, Great Lakes, IL; Code 105 Oakland, CA, Code 120, Oakland CA; Code 120C, (Library) San Diego, CA; Code 128, Guam; Code 154, Great Lakes, IL, Code 200, Great Lakes IL; Code 30C, Norfolk, VA; Code 30C, San Diego, CA; Code 400, Great Lakes, IL; Commanding Officer, Subic Bay; Code 400, Pearl Harbor, HI; Code 400, San Diego, CA; Code 420, Great Lakes, IL; Code 420, Oakland, CA; Code 422, Norfolk, VA; Code 500 Norfolk, VA; Code 505A (H. Wheeler); Code 600 (Utilities Dept) Norfolk, VA; Code 600, Great Lakes, IL; Code 601, Oakland, CA; Code 610, San Diego Ca; Code 700, Great Lakes, IL; Code 700, San Diego, CA; PWO Norfolk, VA; Util Dept (R Pascua) Pearl Harbor, HI; Utilities Officer, Guam
SPCC PWO (Code 120) Mechanicsburg PA
SUPANX PWO, Williamsburg VA
TVA Solar Group, Arnold, Knoxville, TN
U.S. MERCHANT MARINE ACADEMY Kings Point, NY (Reprint Custodian)
USCG (Smith), Washington, DC; G-EOE-4 (T Dowd), Washington, DC
USDA Forest Service Reg 3 (R. Brown) Albuquerque, NM
USEUCOM (ECJ4/L-LO), Wright, Stuttgart, GE
USNA Ch. Mech. Engr. Dept Annapolis MD; PWD Engr. Div. (C. Bradford) Annapolis MD
USS FULTON WPNS Rep. Offr (W-3) New York, NY
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CORNELL UNIVERSITY Ithaca NY (Serials Dept, Engr Lib.)
DAMES & MOORE LIBRARY LOS ANGELES, CA
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HARVARD UNIV. Dept. of Architecture, Dr. Kim, Cambridge, MA
ILLINOIS STATE GEO. SURVEY Urbana IL
LEHIGH UNIVERSITY Bethlehem PA (Fritz Engr. Lab No. 13, Beedle); Bethlehem PA (Linderman Lib. No.30, Flecksteiner)
MICHIGAN TECHNOLOGICAL UNIVERSITY Houghton, MI (Haas)
MIT Cambridge MA; Cambridge MA (Rm 10-500, Tech. Reports, Engr. Lib.); Cambridge MA (Whitman)
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