THE MEASUREMENT OF THE STRESS-% COMPRESSION RELATIONSHIP AND SPECIFIC LOSS OF CUSHIONING MATERIALS UNDER VIBRATORY LOAD

Thesis for the Degree of M. S.

MICHIGAN STATE UNIVERSITY

Richard Norman Maxson

1962

ABSTRACT

THE MEASUREMENT OF
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An important aspect of package design is the development of a package with adequate shock mitigation and vibration isolation characteristics. At the present time, G vs. static stress curves are used in the design of shock mitigation systems and the transmissability factors of package cushioning materials are beginning to be used as design information in vibration isolation problems.

The determination of the dynamic modulus (or stress - % compression relationship) offers promise of the solution of both the shock mitigation and vibration isolation problems, in that it provides data which can be used in the solution of the differential equations governing the package suspension system.

In order to determine the dynamic modulus of a typical cushioning material, an apparatus consisting of an electrodynamic shaker for providing vibratory loading of a specimen of the material and means for measuring the dynamic force on and deformation of the cushioning material was constructed. A strain gage load cell was used for measuring the instantaneous force on the specimen and a linear variable differential transformer was used to measure the instantaneous deformation of the specimen. A dual-beam oscilloscope was used for displaying the deformation and force in an X-Y plot.

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Specimens of polyurethane foam, of a nominal two pound density, were tested at approximately 5, 11, 22, 50 and 100 cycles per second. The data were recorded by photographing the oscilloscope traces and the resulting curves were transformed to stress - % compression axes.

Curves obtained at each of the five above-mentioned frequencies were plotted on an enlarged scale and specific loss measurements were made by planimetric methods.

The experimental data showed that the dynamic modulus was highly dependent upon test frequency. A smaller dependence of specific loss on frequency was also observed.

The testing program showed that this technique was capable of yielding useable data.

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A THESIS

Submitted to
Michigan State University
in partial fulfillment of the requirements
for the degree of

MASTER OF SCIENCE

Department of Forest Products

1962

G 27319
9/28/62

ACKNOWLEDGMENTS

Grateful acknowledgment is made to Dr. James W. Goff for many helpful discussions and suggestions during the course of this work. I wish also to express my thanks to the Packaging Research Laboratory of the Glass Container Manufacturers Institute for the use of several pieces of apparatus without which the successful completion of this project would have been impossible.

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INTRODUCTION

The Package Cushioning Problem

The designer of a package for a fragile item faces the problem of choosing a suitable cushioning material, of the proper size and shape, in order for the package to adequately protect the item from the hazards of the handling and transportation environment. Two of the most important hazards are those of shocks, such as might be caused by dropping or throwing the package, and vibration forces which are imposed on the package by the transporting vehicle.

The mitigation of shock is accomplished by the use of cushioning material which absorbs the energy of impact. The mechanics of this situation have been described in detail by Mindlin (1). The designer's problem is to choose the proper cushioning material for the particular item, that is its size, weight and fragility, keeping in mind the size, weight, cost and availability of the cushioning material. The solution to this problem requires that the designer know the shock mitigation capabilities of materials as determined under dynamic conditions.

The package designer must also be aware of the fact that all transportation vehicles apply forces of a vibratory nature to the packages being carried in their cargo spaces. The frequency and amplitude of these oscillatory motions depends greatly upon the mode of transportation being considered--being, for example, of low frequency and high amplitude for rail transport and of high frequency and relatively low amplitude for aircraft. The package designer must be aware of two factors in relation to vibratory motions: first, he must insure that the cushioning system provides adequate isolation of critical parts which may have resonant frequencies within the range that would be encountered in the transportation environment. Secondly, he must provide a cushioning system that, together with the mass of the item being packaged, does not have a natural frequency within the range of the transportation vibrations; or if such a situation is unavoidable--which it often is--the package cushioning system must include sufficient damping to keep within acceptable limits the motion of the packaged item and, hence, the forces on it.

Present Practice in Package Design

In considering the shock mitigation problem, the present practice followed in good package design involves the use of "G *versus* static stress" curves. These curves, which were proposed by Stern (2), are plots of peak acceleration (usually expressed in multiples of the earth's gravitational field) obtained by making impact tests of cushioning material *versus* the weight of the impacting platen divided by the area of the specimen--hence the term "static stress."

If G *vs.* static stress curves for a number of materials, in a number of thicknesses, obtained from a number of drop heights are available, the package designer can make a rational choice of which material to use and the proper dimensions of it. This procedure, for the design of packages based on G *vs.* static stress curves, forms the basis of a present military specification for the procurement of cushioning material (3). Needless to say, the obtaining of such data is a laborious and expensive process.

The vibration transmissability of cushioning materials is a property which has only fairly recently been considered. Henney and Leslie (4) and Wilson (5) have published test procedures and data concerning this property. In essence, these test procedures apply a sinusoidal vibratory acceleration of measured magnitude to a weighted specimen of cushioning material. The resulting acceleration of the weight is monitored and the ratio of the resulting to the applied acceleration is known as "transmissability". The transmissability <u>vs.</u> frequency curves show the typical resonance peak as would be expected for a mass-spring system subjected to an oscillatory displacement. However, due to the non-linearity of the spring-rate of most cushioning materials, the resonance point varies with the amplitude of the applied motion and, in principle, to make a thorough investigation of this property a large number of amplitudes and applied weights must be investigated. This process is also extremely laborious unless considerable effort is applied to make the recording of data at least semi-automatic.

THE STRESS - % COMPRESSION RELATIONSHIP

The relationship between the stress on a cushion and the amount of its compression, measured under dynamic conditions, which will be termed the "dynamic modulus," contains all the information necessary to the solution of both the shock mitigation and the vibration transmissability problems. This procedure is given a thorough treatment by Mindlin (and others) but, unfortunately for the practical situation, most of this work is based upon idealized cushions having moduli which can be represented by reasonably simple mathematical expressions.

The solution of the shock mitigation problem, in general, lies in the solution of the differential equation for a mass, m, on a spring:

$$m\ddot{x} + d\dot{x} + kx = 0$$

with the boundary conditions:

$$x = 0 \quad \text{at } t = 0$$

$$\dot{x} = v_0 \text{ at } t = 0$$

where x is the displacement of the mass and d and k are constants related to the amount of damping and the stiffness of the spring, respectively.

It can be shown from the solution of this relatively simple case, that the area under the dynamic modulus curve represents the energy absorption capabilities of a cushioning material and that the area enclosed within the loop of the curve represents the energy dissipated by the cushion, and thus is a measure of its damping characteristics.

However, in the actual situation, where the cushioning is supplied by a bulk material, the solution of the equation is somewhat more difficult. In the first place, k is not a constant, being a function of x and perhaps also of its derivatives. Secondly, it is likely that the damping of real materials is not strictly of the velocity dependent type, in which d is a constant. The actual functional dependence of k and d must be determined by experiment. Given experimental data regarding these two terms, the solution of the differential equation would probably be obtained most easily by either numerical or computer methods. A particularly interesting method of solution of this problem by means of an analog computer has been described by Venning and Gutteridge (6).

Similarly, the solution to the vibration transmissability curve lies in the solution of the equation

$$m\ddot{x} + d\dot{x} + kx = \sin \omega t$$

with appropriate boundary conditions. The above-mentioned difficulties also apply to the solution of this equation in the actual situation.

Little information has been available regarding the dynamic modulus of any of the commonly used cushioning materials. A group working at the University of New Mexico have published work (7, 8, 9) of somewhat this nature using impact loading conditions. In addition, Payne (10) has published work using a sinusoidal strain of relatively low frequency on elastomeric materials.

The measurement of the dynamic modulus of a typical cushioning material forms the basis of this present work.

APPARATUS

It was desired to measure the dynamic modulus of cushioning materials under forces of a vibratory nature. Thus it was necessary to have some means of applying vibratory motion to the specimens and a method of measuring and displaying both the displacement and force during the motion.

Electrodynamic Shaker

The means chosen to apply the force to the specimen was an electrodynamic shaker. A Ling Model LPM-25 was used. This shaker was capable of producing twenty-five pounds of force over a frequency range from 5 - 5000 cycles per second. The total available double amplitude of the shaker was one-half inch. The shaker was driven by a 100 watt amplifier (Ling Model 100A) which was, in turn, excited by an audio oscillator (Hewlett-Packard Model 200 CD).

Due to the fact that the shaker armature must be flexibly mounted, an auxiliary means of supplying an initial load to specimens was necessary; otherwise, the armature, in static equilibrium with the spring force supplied by its mounting and the (relatively) large cushion force, was near its rear stop. Thus, on application of the exciting voltage, the armature could strike this stop and damage the shaker. It was found that a voltage, supplied by dry cells, placed in series with the shaker and amplifier would adequately supply this preload. The internal resistance of the dry cells was not great enough to cause a serious impedance mis-match.

The armature was fitted with a loading platen machined from magnesium (to keep the weight to a minimum). The platen was about

2-5/8" in diameter and the loading surface was lapped flat with fine emery paper placed on a ground steel surface.

Load Cell

The force exerted on the specimen was measured with a strain gage load-cell (Baldwin Model V-1S of 500 pound capacity). The gages in the cell were connected in a conventional Wheatstone bridge circuit. The bridge was excited by an 11 volt D.C. source derived from a Video Instruments Model SRB 200 Strain Gage Module. The output of the bridge was amplified by a D.C. amplifier (Video Instruments Model 71A). The output of this amplifier was connected to the input of an oscilloscope (Tektronix Model 502) through a three-section, R-C filter.

The load cell was fitted with a platen, machined from aluminum, about 2-5/8" in diameter. The surface of this platen was also lapped to eliminate tool marks and make it as flat as possible.

The load cell was calibrated by clamping it in a vertical position and placing weights of known magnitude on the platen. The oscilloscope deflections corresponding to these vertical forces were plotted vs. the force and a least-squares fit was made. The slope of the line thus obtained was compared with the deflection of the oscilloscope trace which occurred when a resistor of known value was placed in parallel with one arm of the bridge. It was found that a 1 megohm (precision) resistor was equivalent to 23.4 lbs. The use of a parallel calibration resistor offers a very good way of effecting calibration since it is unaffected by variations in bridge supply voltage or gain of the amplifiers in the system.

Differential Transformer

The deformation of the cushion specimen under load was sensed by a linear variable differential transformer (11). The linear variable differential transformer (abbreviated LVDT) used was a Schaevitz Model 200 SS-L. The primary of the LVDT was excited by a sinusoidal voltage supplied by a Heath AO-1 audio oscillator at a 0.75 volt r.m.s. level.

The core of the LVDT was coupled to the loading platen through the armature of the shaker. A 7" length of thin-walled stainless steel tubing fitted at each end with aluminum inserts which were threaded to screw into the core at one end and the shaker armature at the other. (The use of stainless steel and aluminum was necessary to avoid seriously affecting the magnetic fields of either the shaker or the LVDT.)

A piece of plexiglass tubing was cemented inside the LVDT. This tube was sized to fit snugly inside the transformer body and to make a free sliding fit with the transformer core. The tube was used to eliminate the possibility of radial vibrations of the core within the transformer since the LVDT is somewhat sensitive to motion in this direction.

The LVDT unit, as supplied by the manufacturer was designed for an excitation frequency of 20,000 cps, however it was found that a larger linear range could be obtained when the excitation frequency was reduced to 10,000 cps. Also, this particular unit was specified to be linear, within 1%, over a range of plus and minus 0.200" from the null position. Since the shaker was capable of producing a greater stroke than this, the possibility of extending the operating range of the LVDT was investigated. It was found that the unit was linear within about 2% over a range of plus and minus 0.250"; hence, since this error did not seem to be too great in view of the other possible errors in the system, the unit was used.

Ordinarily, an LVDT is used either on one side of the null voltage position or the other (if voltage vs. deflection is to be measured) or used in a null balance system. For working over the entire range of the transducer it was necessary to use a phase sensitive detector. Rather than connecting the two secondary windings in "series opposing" as is usually done, the secondaries were connected in "series aiding" and this junction was grounded. (See Figure 1.) The other lead from each of the secondary windings was then fed to the grid of a voltage amplifier stage. The output of each of these stages was rectified using a half wave rectification configuration. The output from the rectifiers was then added and filtered with a United Transformer Co. LMI-1500 low-pass filter to remove the 10,000 cps component. The output of the filter was connected to the oscilloscope.*

Since a filter was necessary in the output of the amplifier-demodulator a phase shift, measured to be about 12° at 100 cps, appeared in the output signal. In order to compensate for this phase shift, a filter was also used in the output of the load cell (as was mentioned above). It was found that a three-section, R-C

* The circuitry for the phase sensitive detector was adapted from the Daytronic Corporation's Model 400 differential transformer amplifier.

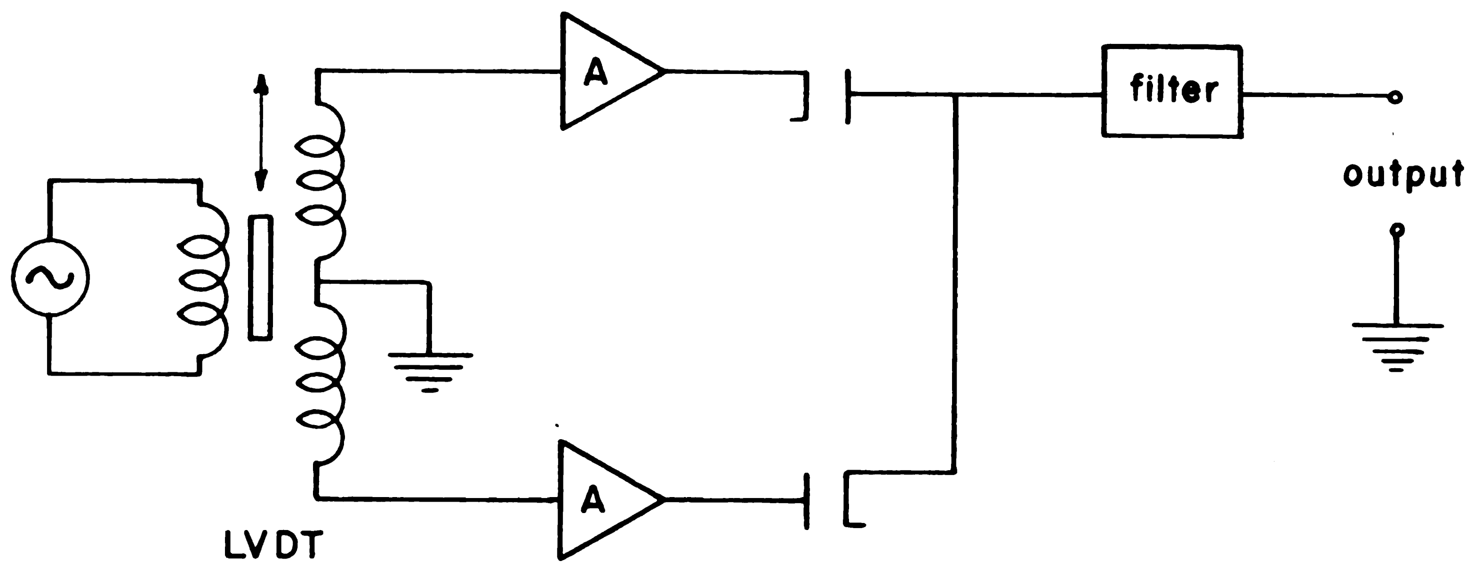

Figure 1. Block diagram of amplifier-demodulator for LVDT

low-pass filter, each section of which was designed to have a cut-off frequency of about 3,400 cps, gave a phase shift equal to that of the filter used in the demodulator. The filter in the output of the load-cell amplifier also served to reduce the noise level in that circuit.

Oscilloscope

Both the deformation signal and force signal were fed into a Tektronix Type 502 oscilloscope. D.C. amplifiers were used on both channels. This oscilloscope, being of the dual beam type, was capable of simultaneously presenting both the force and deformation signals as a function of time, or one signal could be used on the X-axis and the other on the Y-axis. The procedure followed was to make preliminary checks using a time axis presentation and then to switch the deformation signal to the X-axis. Thus a plot of force *vs.* deformation could be obtained. This trace was photographed with a Land-process camera for data analysis.

Complete Apparatus

A photograph of the loading apparatus including the shaker, platens, LVDT and load cell is shown in Figure 2. The shaker was clamped between angle iron standards. The load cell was mounted on a large angle iron which was further reinforced by triangular pieces welded in place. The whole assembly was mounted on a piece of 10" channel iron about 30" long. It was found that this assembly was quite rigid, although there appeared to be a slight resonance in the neighborhood of 45-48 cps as judged by the fact that the audible output of the system increased near this point.

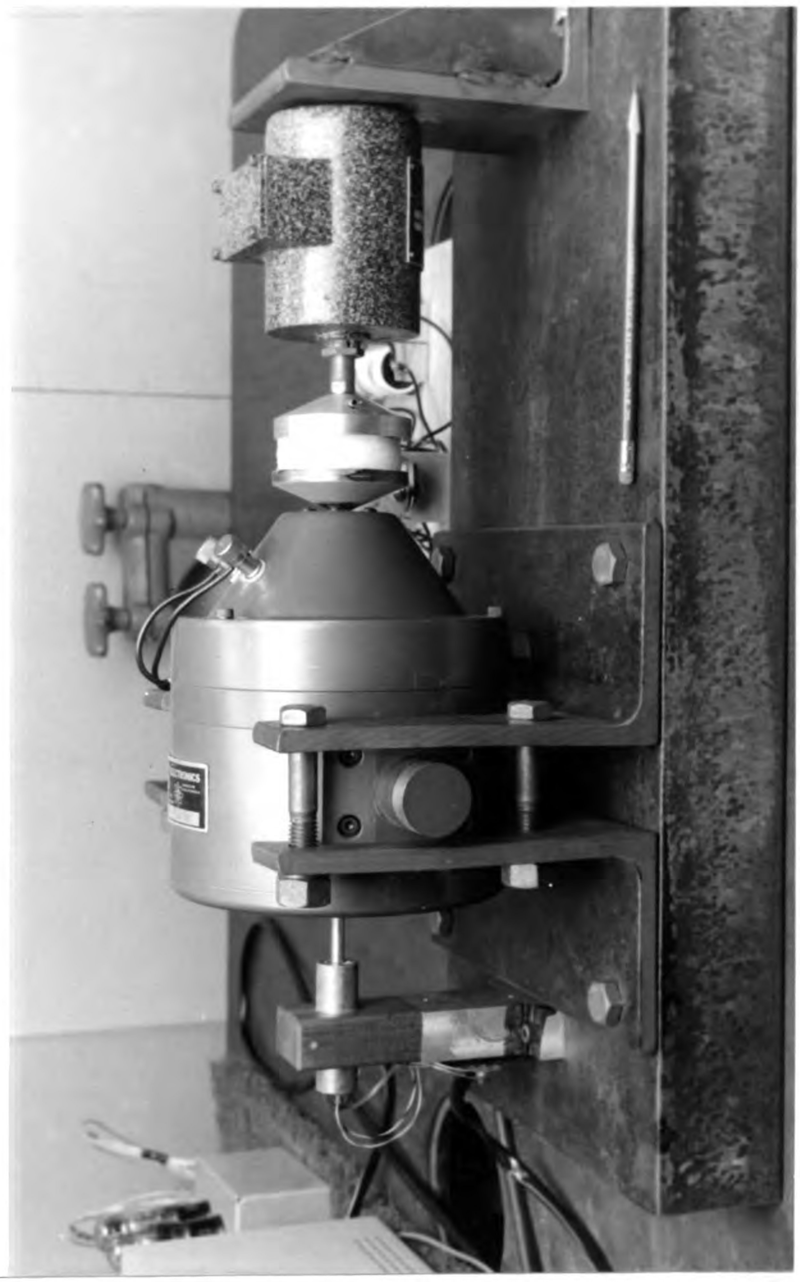

Figure 2. Photograph of loading apparatus

Figure 3 shows a schematic block diagram of the complete apparatus.

The filters in both the deformation and force measuring circuits limited the frequency response of the measuring system. It was found that the frequency was flat, within 5%, up to 500 cps. This was entirely adequate since the highest test frequency was 100 cps.

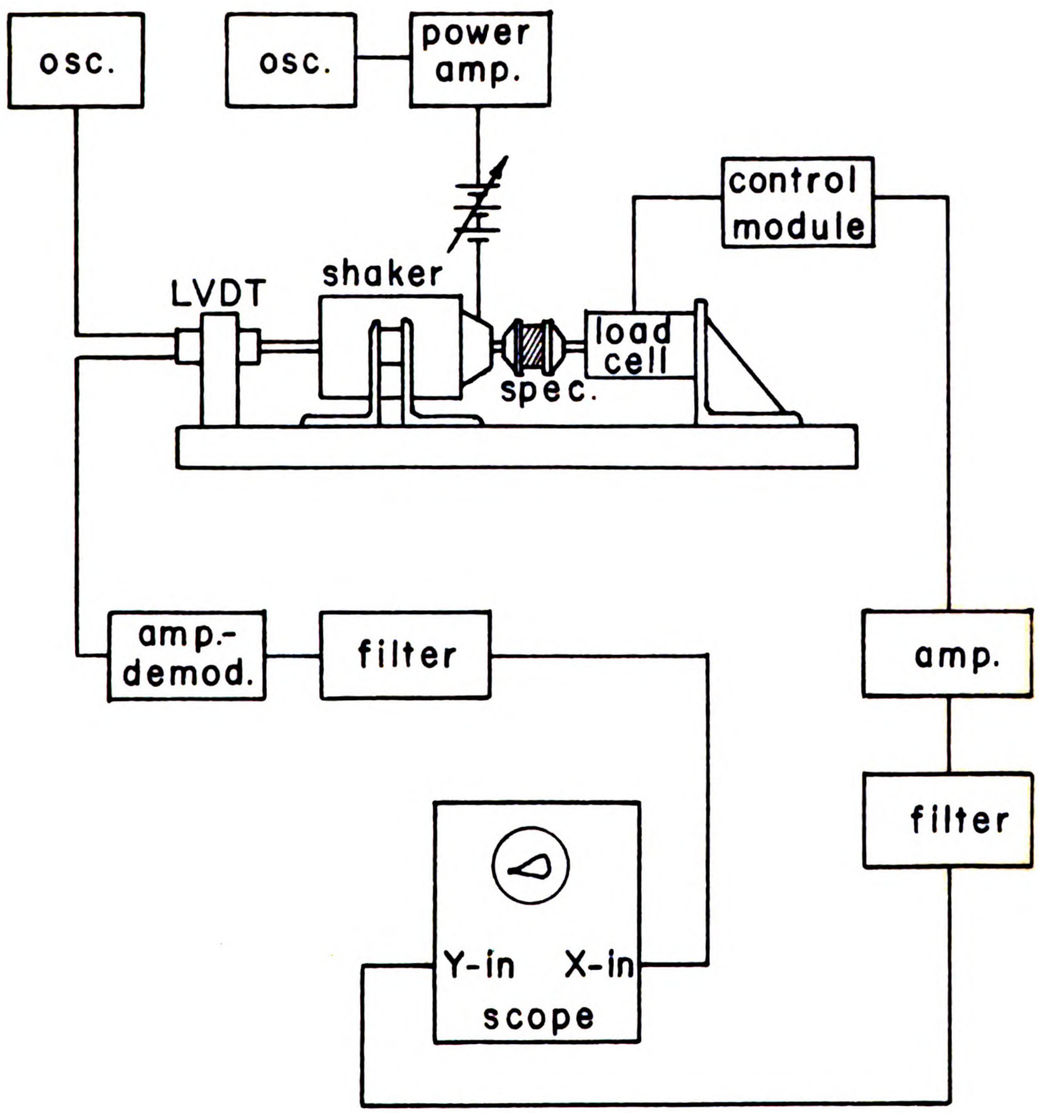

Figure 3. Block diagram of apparatus

SPECIMENS AND SPECIMEN PREPARATION

Specimen Material and Size

The specimen material, polyurethane foam, was chosen for several reasons. First it is one of the widely used materials (12). Secondly, it is a material of high damping characteristics, thus making the specific loss determinations more readily obtainable. Also, urethane foam has a highly non-linear stress-% compression curve when tested statically (13) and it was desired to ascertain whether this type of behavior was also present under dynamic loading conditions.*

The size of the specimens to be used was quite largely determined by the apparatus available. Since it was desired to strain the specimens to about 70% strain, the double amplitude limit of the shaker dictated a specimen thickness of about 0.7". The specimen area was limited by the maximum output force of the shaker. Preliminary, static, stress-% compression data indicated that a specimen area of about five square inches was the allowable maximum. It appeared wise to stay near this maximum area in order to obtain as high a signal-to-noise ratio as possible in the load cell circuitry.

Specimen Cutting

Specimens in the form of right circular cylinders were prepared.

* It is worthwhile to note that the specimens used in this work were cut from specimens furnished to the Glass Container Manufacturers Institute by the Feltman Research and Engineering Laboratory of Picatinny Arsenal for the determination of G <u>vs.</u> static stress curves. Thus, these curves have been obtained and may be useful for comparison purposes.

The foam was cut to thickness from nominal one-inch sheets using a band saw. The circumferential cuts were made with a hole saw, modified by removing the center pilot drill and replacing it with a rod which was so positioned that it did not interfere with the specimen material during cutting. In order to facilitate the cutting process, the specimens were pre-chilled in a dry ice chamber. For cutting the specimens, the hole saw was chucked in a drill press operating at about 2000 rpm.

Specimen Dimensions and Density

Measurements of specimen diameter were made with a scale divided in 0.01". Two readings, at approximately right angles, were taken on each face of the specimen. These four readings were averaged and this average value was used in computing the specimen area.

Specimen thickness was measured with a 0.106 lb. (0.025 psi) load placed on the specimen (14). Readings to the nearest 0.001" were taken at the center of the specimen using a dial micrometer.

Specimen weight was measured using an analytical balance and recording the weight to the nearest 0.1 milligram. Density of the specimens was calculated to give an indication of the homogeneity of the sample lot.

Thickness, area and density of the specimens are given in Table I.

Table I - Physical characteristics of specimens

Specimen No.	Thickness (in.)	Area (sq. in.)	Density (lb./cu. ft.)
1	0.682	4.23	2.04
2	0.712	4.30	2.04
3	0.732	4.45	2.00
4	0.685	4.30	1.99
5	0.710	4.52	1.94

TEST PROCEDURE

A specimen was placed between the platens and attached to the fixed platen with a small piece of double-faced pressure-sensitive tape. The oscillator driving the shaker was set at the desired frequency and the power amplifier gain control advanced. The specimen was subjected to about one minute of vibration prior to making a measurement. When the desired time had elapsed, a photograph of the oscilloscope trace was made.

After making the exposure of the oscilloscope trace the specimen was removed. There then being no force acting on the load cell, an exposure of the oscilloscope beam, which appeared as a dot, (representing zero load) was made. A third exposure was made with the 1 megohm calibration resistor in parallel with one arm of the bridge. This gave a point located 23.4 lbs. from the zero-load point.

A fourth and fifth exposure were made with a 0.251" and 0.500" thick aluminum bar placed between the platens. These two points then gave reference points from which % compression could be derived.

The frequencies of operation were chosen to give points approximately equidistantly spaced on a logrithmic scale from 5 to 100 cps.

A new specimen was used at each frequency of operation.

DATA REDUCTION METHODS

The first step in the data reduction consisted of tracing the photograph using a micro-projector. By this means, the photographs were enlarged about 8 times, thus improving the accuracy of subsequent measurements and calculations. Figure 4 is a sketch of a hypothetical force-deformation trace as it would appear on a photograph (or tracing).

Stress Determination

The point f_1 shown in Figure 4 was the zero-force point while f_2 was the point representing 23.4 lb. force. The point f_1 allowed the determination of the zero-stress axis and a measurement of the distance between f_1 and f_2 allowed the calibration of the force axis. The units of the vertical axis were converted to stress units (psi) by dividing by the area of the specimen.

% Compression Determination

The points d_1 and d_2 were obtained with the 0.500" and 0.251" spacers placed, respectively, between the platens. The distance between these two points, which was thus equivalent to 0.249", allowed the calibration of the % compression axis.

It was found that if % compression was computed using the thickness measured statically and the distance d_1-d_2, the point "O" would have had a substantial positive % compression. Quite obviously, this was not physically possible and it was concluded that the one-minute pre-vibration of the specimen had caused a deformation which was "permanent," at least on the time scale involved

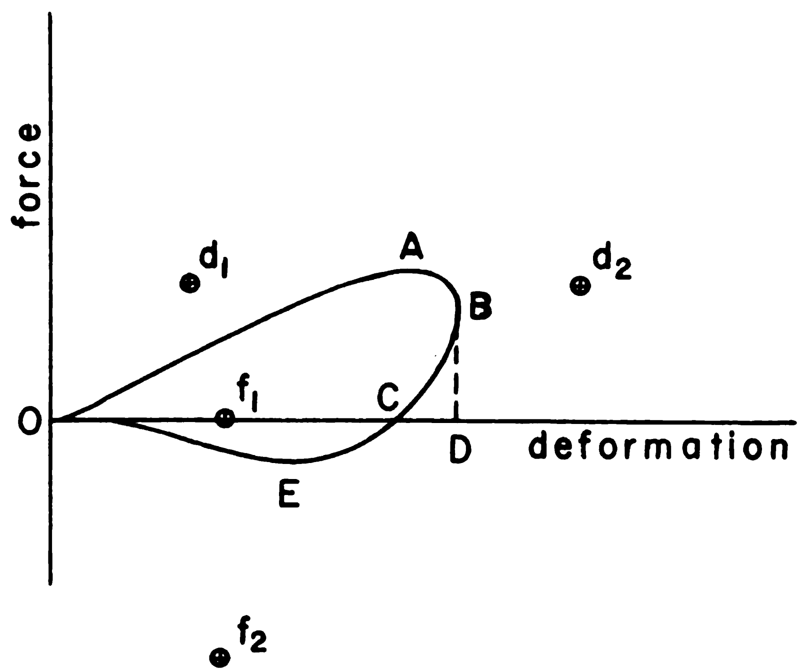

Figure 4. Hypothetical force-deformation curve as it would appear on photograph

here. Therefore the point "O", at which positive stress first appeared was taken as zero % compression. Furthermore, it was possible to construct the % compression axis using the zero point together with the known fact that the 100% compression point was 0.251" beyond the deformation represented by point d_2*.

Specific Loss Determination

The specific loss determination was made by measuring, with a planimeter, the total area under the curve and the area enclosed within the curve. The specific loss was then obtained by dividing the enclosed area by the total area. That is: (See Figure 4.)

$$\text{Specific Loss } (\%) = \frac{\text{Area OABCO}}{\text{Area OABDO}} \times 100$$

* It was not possible to follow exactly this procedure for the 100 cps data due to the fact that no point was recorded which was not above zero force, thus making a direct determination of the position of the vertical axis impossible. However by computing, for the other four tests, the thickness of the specimen during vibration (that is, the thickness at the zero-stress point) and comparing these values with those measured statically, it was found that the loss in thickness was about 18% for the specimens tested at 22.4 and 50 cps. (It was found to be about 5% for the 5 cps test and 12% for the 10.6 cps test.) Therefore, it was assumed that the thickness loss was 18% for the 100 cps test and the vertical axis was placed accordingly.

EXPERIMENTAL RESULTS

Stress - % Compression Curves

The stress vs. % compression curves obtained in this project are shown in Figures 5 through 9. Figure 5 gives the curve obtained at 5.0 cycles per second, Figures 6, 7, 8 and 9 give the curves obtained at 10.6, 22.4, 50.0 and 100.0 cycles per second, respectively. Specimen 1 was tested at 5.0 cps, specimen 2 at 10.6 cps, specimen 4 at 22.4 cps, specimen 3 at 50.0 cps and specimen 5 at 100.0 cps.

Figure 9 deserves special mention. Due to the force limitations of the shaker, it was not possible to obtain a complete curve at the 100 cps frequency. It was possible, however, by adjusting the pre-load battery voltage to obtain three small portions of the curve. These three portions, along with an estimated envelope, shown as a dotted line, were the best which could be obtained with this apparatus.

Specific Loss Data

The data for specific loss for each of the five test frequencies are given in Table II. Due to the fact that complete curves were unobtainable for the 100 cps test frequency, the specific loss for each of the three segments of the curve are given, together with an average of these three values.

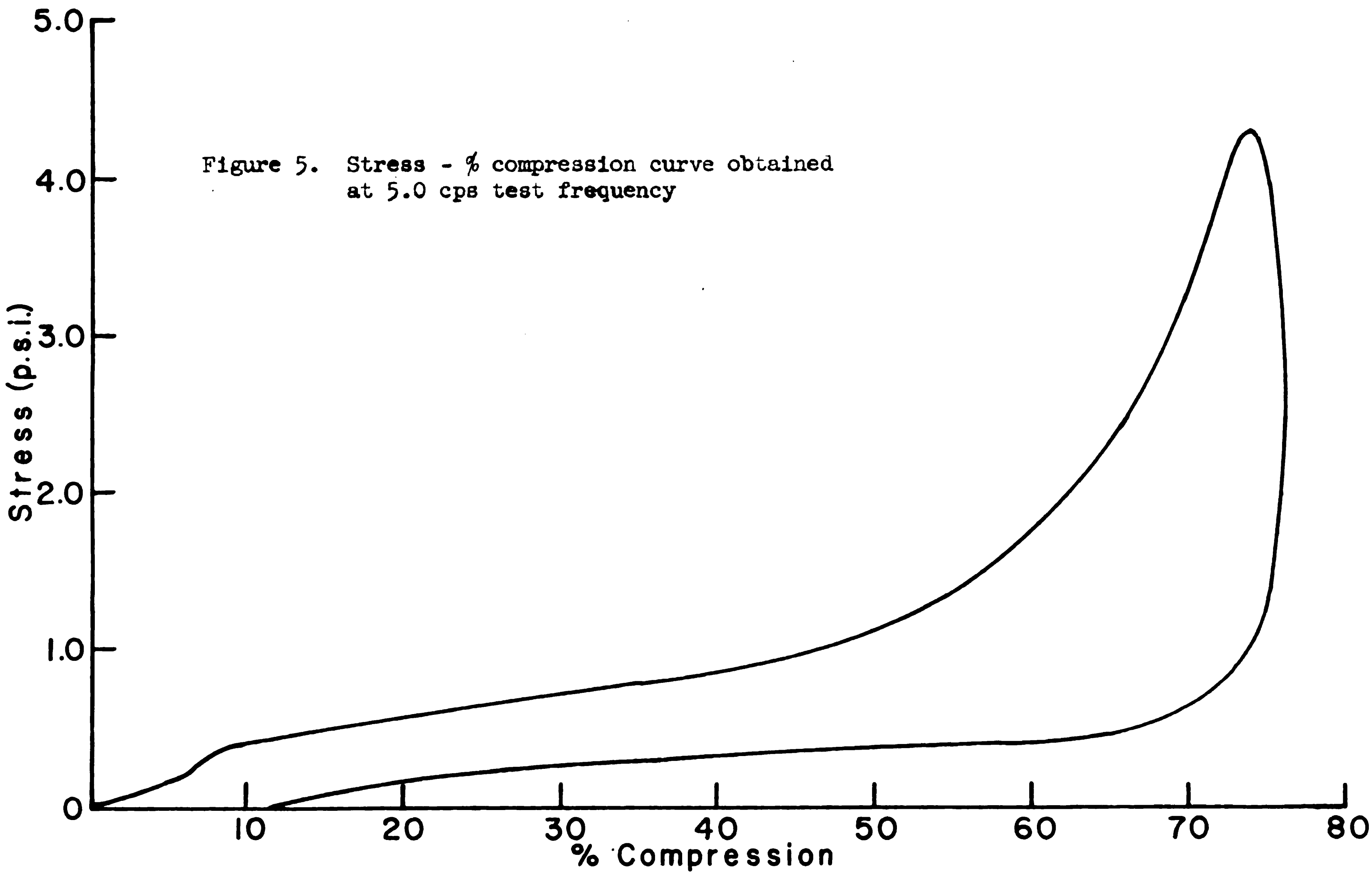

Figure 5. Stress - % compression curve obtained at 5.0 cps test frequency

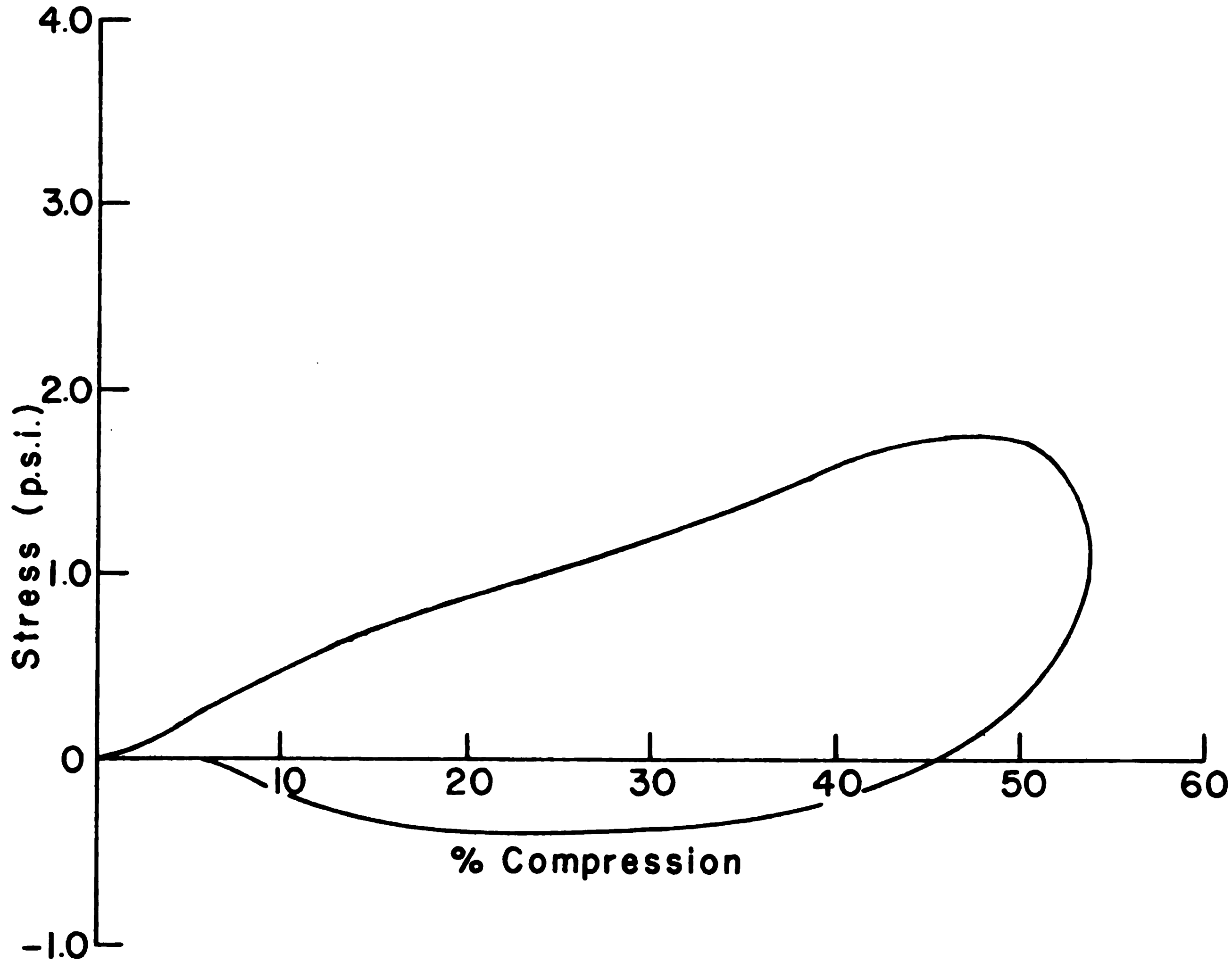

Figure 6. Stress - % compression curve obtained at 10.6 cps test frequency

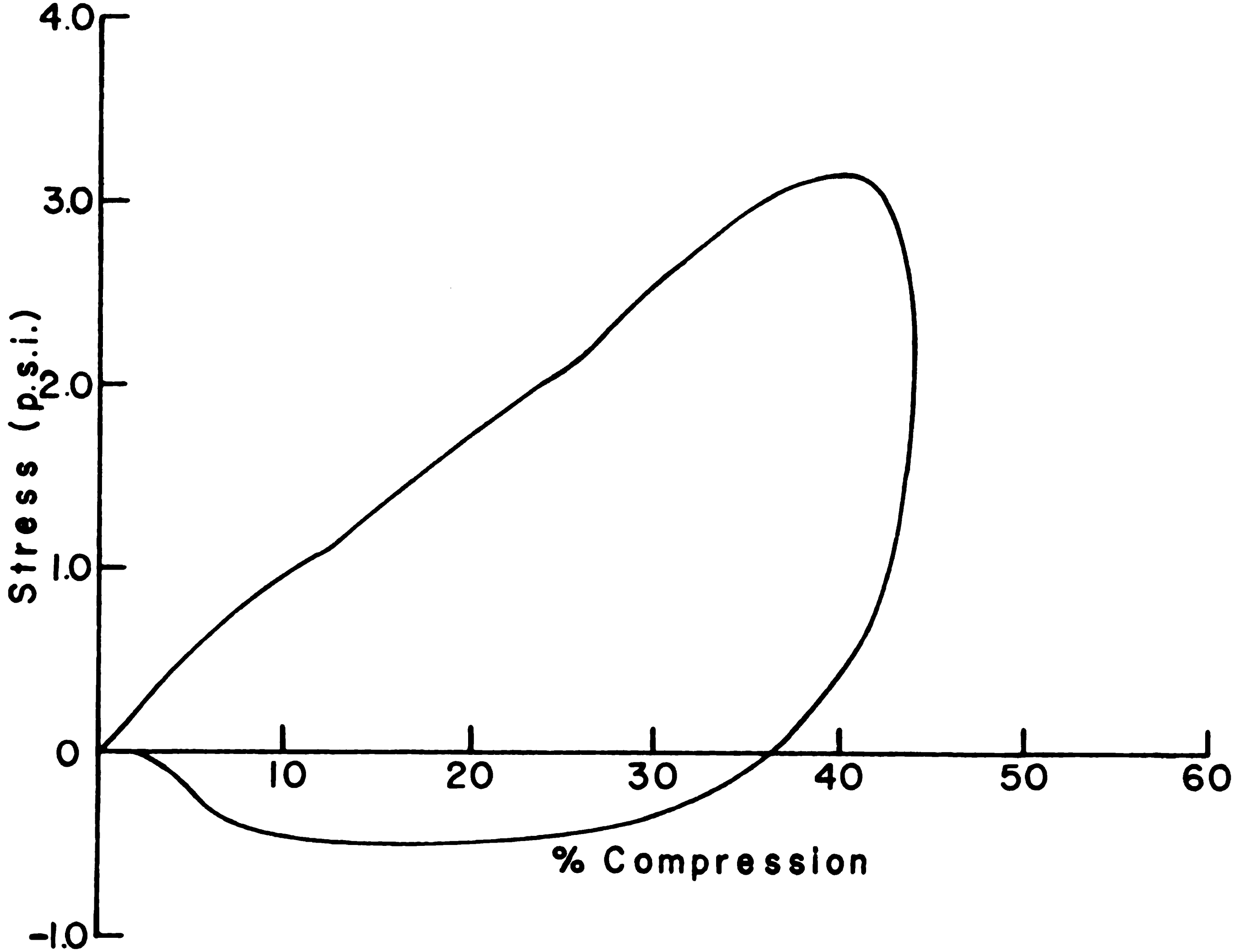

Figure 7. Stress - % compression curve obtained at 22.4 cps test frequency

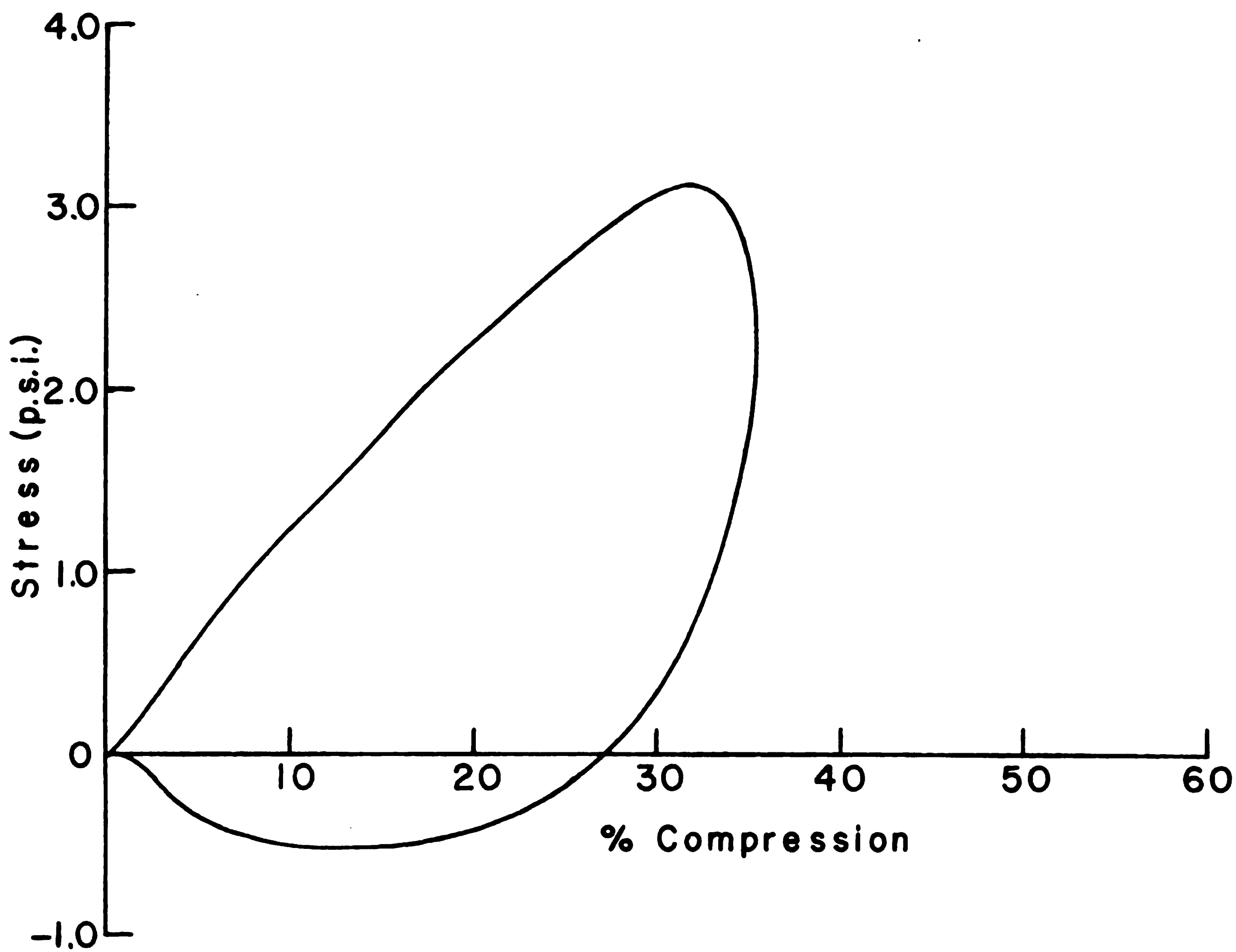

Figure 8. Stress - % compression curve obtained at 50.0 cps test frequency

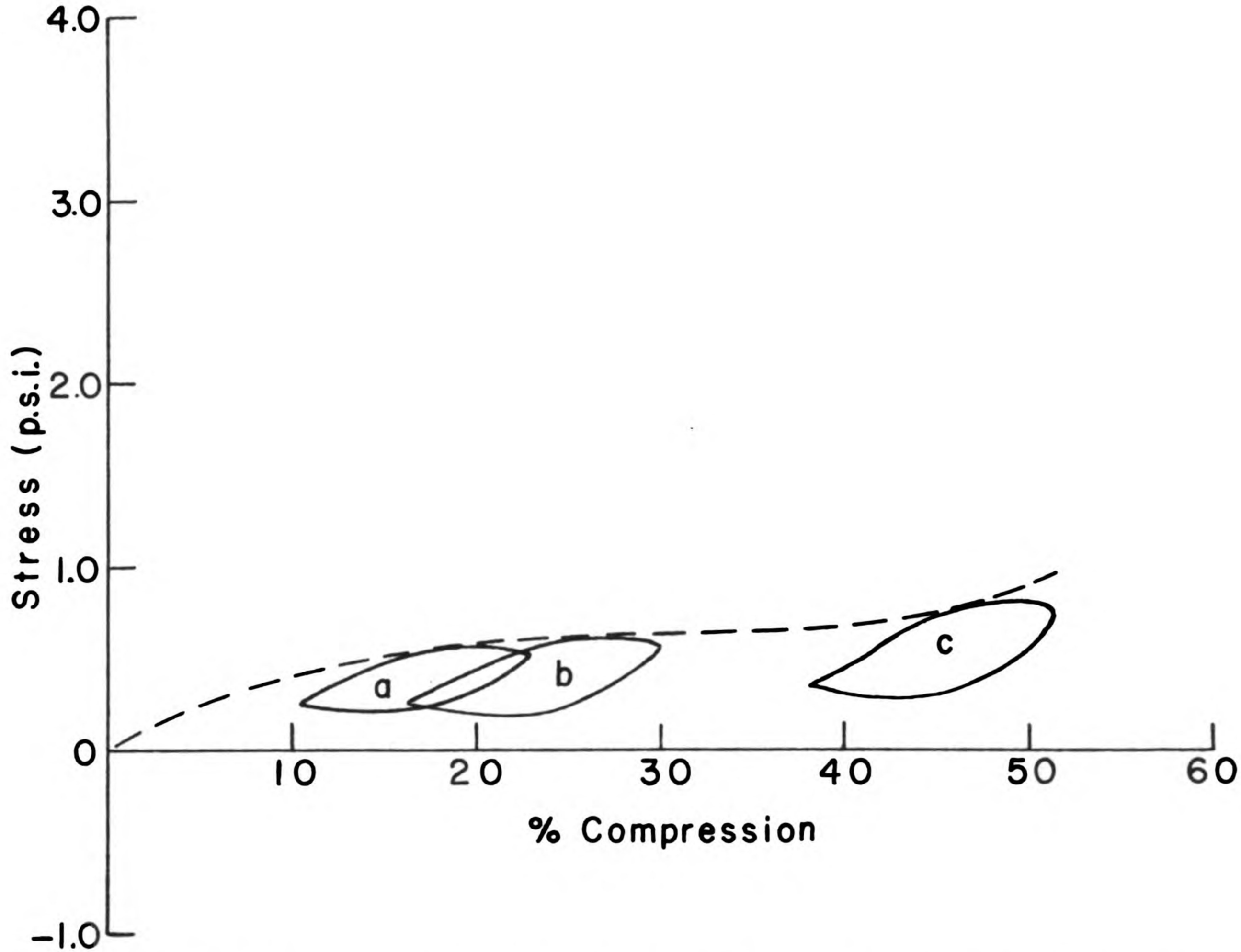

Figure 9. Stress - % compression curve obtained at 100.0 cps test frequency

Table II - Specific loss measurements of urethane foam

Frequency (cps)	Specific Loss (%)
5.0	75.7
10.6	95.2
22.4	94.7
50.0	91.4
100.0 (a)*	80
100.0 (b)*	85
100.0 (c)*	85
100.0 - average	83

* Letters in parentheses denote the portion of the curve from which the tabulated value was derived. (See Figure 9.)

DISCUSSION OF DATA

The stress - % compression curves given in Figures 5 through 9 show several interesting points. First, the 5 cps curve (Figure 5) shows a stress - % compression relationship quite similar to that obtained statically in that it exhibits a relatively steep rise in stress for small percentages of compression, followed by a long, relatively flat portion where the cushion is absorbing energy without appreciable increase in stress. As the cushion begins to bottom, the stress rises rapidly, as will occur with all types of bulk cushioning materials.

When the frequency is increased to 10 cps, however, the behavior is quite different. (See Figure 6.) The slope is much larger and the long, relatively flat portion is no longer present. This trend is continued and accentuated in the 22 and 50 cps curves (Figures 7 and 8). The lack of availability of complete data at the 100 cps test frequency prohibits making detailed evaluation of the cushioning material's behavior at this frequency; however it is readily apparent that the slope is much smaller and thus the cushion is capable of absorbing less energy at this frequency for a given deformation. (See Figure 9.)

The specific loss data given in Table II show that the specific loss is quite high at all frequencies--thus implying a large amount of internal damping. The specific loss appears to have a maximum value in the 10-50 cps range, falling somewhat at frequencies both higher and lower than this.

Two rather unexpected phenomena appeared in the data. Referring to Figure 4 it will be seen that a region of negative stress exists (region CEO). This negative stress, as recorded by a tension on the load cell was entirely unexpected. It appears plausible that this phenomenon is caused by the fact that air is expelled from the cellular structure of the cushion during the compression stroke of the platen; then as the platen begins to retract, the partial vacuum within the specimen causes the specimen to adhere to the platens and thus create a tension on the load cell which appears on the oscillograms as a "negative" stress.

The second anomaly consists of the fact that even though the stress reaches a maximum at point A (see figure 4) and thereafter decreases, a positive deformation continues to the point B. This behavior appears to be due to the fact that the material creeps under the applied load, even on this fast time scale.

CONCLUSIONS

It can be concluded that, in view of the results obtained in this project, this technique of evaluating some of the significant properties of cushioning materials is practical. Further, as explained above, it is felt that this method is capable of yielding package design data which is more fundamentally a property of the material than some of the more simulative test methods now being used.

Although the primary purpose of this project was to explore the possibilities of this method of cushioning material evaluation, the data indicate that significant changes in the properties of polyurethane foam occur when the speed of loading is increased. It is apparent that the dynamic modulus (that is, the spring rate, $\underline{k}$, discussed on pages 4 and 5) is not only a function of the compression, but also of rate of compression. It appears also, due to the fact that a maximum appeared in the specific loss data, that the damping of this material is dependent not only on velocity, but on other factors as well.

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