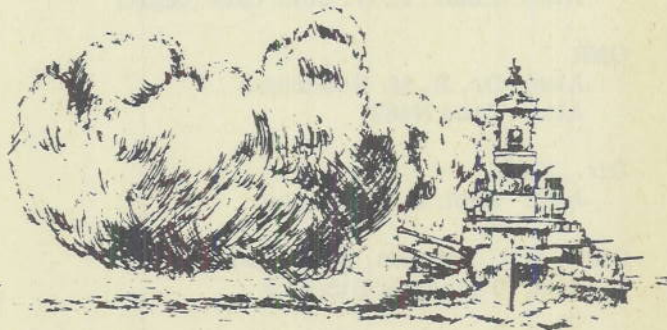


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# **SHOCK AND VIBRATION BULLETIN**

**NO. 8**



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**NAVAL RESEARCH LABORATORY**

CAPTAIN H. A. SCHADE, USN, DIRECTOR

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## **FOREWORD**

The ninth Symposium attempted to review and appraise certain instruments which are employed in the study of shock and vibration. So far, no single instrument has been devised which is a universal shock indicator or which can serve as an all-purpose vibration meter. Just as the ship, the plane, and the tank differ in many of their functional characteristics, so do also some of the instruments which are used to investigate specific shock and vibration problems in those vehicles. Only through the proper use of suitable instruments are we able to provide quantitative information so urgently needed by the engineer, the designer, and the constructor. Therefore, in choosing a measuring device for a particular job, it might be well to ask the following questions:

- (a) What information is needed from this study?
- (b) What quantities must be measured to satisfy the need?
- (c) What is the simplest and most practical method of obtaining the required measurements?
- (d) Will this or that instrument best serve the purpose and provide the accuracy expected?

It is hoped that some of the questions will be answered by critically examining and objectively discussing the various instruments which are described in these pages.

## ATTENDANCE

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Norman E. Lee  
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Squier Signal Laboratory, SCEL

Joseph J. Oliveri  
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Watson Laboratories, AMC

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Los Alamos Scientific Laboratory

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John A. Southwick

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Albert E. McPherson  
H. A. Pratt  
L. B. Tuckerman  
J. H. Van Trump

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Wing Comdr. J. H. Neal

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H. M. Trent  
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## PROPERTIES AND USES OF RESISTANCE STRAIN GAGES

by

David E. Weiss, NAMC, NAES

The theory and construction of the variable resistance strain gage are discussed. Examples from associated equipments used in aircraft research and testing are presented. The advantages and disadvantages of the strain gage are enumerated. Special techniques and applications are illustrated.

### INTRODUCTION

Dynamic investigations of aircraft structures have been undertaken at NAES with one or more of the following objectives:

- a). Test aircraft structures for conformance to procurement specifications.
- b). Extend operating range of existing structures and to obtain design data for new structures.
- c). Trouble-shoot; i.e., determine causes of service failures.

One of the more important tools in these investigations has been the variable resistance strain gage. Because of its small size and light weight and consequent small effect upon the system to which it is applied, it very nearly approaches the ideal instrument. Another important advantage is its versatility. It has been successfully used to measure force, acceleration, velocity, displacement, temperature, pressure. These are but a few of the applications to which it has been put. Among its disadvantages is the uncertainty that sometimes arises concerning the bonding of the gage to the test specimen. This should happen rarely but when it does, it is a source of consternation. Another disadvantage is the fact that the range of the gage is limited for some applications. Under static or infrequent loading, it is useable up to approximately 5,000 microinches per inch, but under repeated loads the gage will fail at appreciably lower strains, as low as 500 microinches per inch for some types of gages.

The following paragraphs will describe the gage in general and will briefly discuss the associated equipment as well as some applications in the field of aircraft testing.

### THEORY OF THE WIRE RESISTANCE GAGE

The wire resistance gage makes use of the principle that the electrical resistance of a filament of wire is a function of the strain to which the wire is subjected. Under tension, the length of the wire will increase and its cross-sectional area will decrease, both factors tending to increase the electrical resistance of the wire. If the wire is properly supported, compression will produce the opposite effects. However, closer examination shows that this simple picture is far from complete. The resistance of

a cylinder of wire is given by the expression:

$$R = e \frac{L}{A} \quad (1)$$

where  $R$  = total resistance of wire  
 $e$  = specific sensitivity of wire  
 $L$  = length  
 $A = kD^2$ , the cross-sectional area  
 $\mu$  = Poisson's ratio

Taking the logarithm of both sides of (1) and then taking differentials, we obtain:

$$\frac{\delta R}{R} = \frac{\delta e}{e} + \frac{\delta L}{L} - \frac{\delta A}{A} \quad (2)$$

But

$$\frac{\delta A}{A} = 2 \frac{\delta D}{D} = -2\mu \frac{\delta L}{L} \quad (3)$$

Therefore (2) becomes:

$$\frac{\delta R}{R} = (1 + 2\mu) \frac{\delta L}{L} + \frac{\delta e}{e} \quad (4)$$

The gage sensitivity factor,  $S$ , is defined as the ratio of the unit resistance change to the unit elongation. Thus:

$$S = \frac{\delta R/R}{\delta L/L} = 1 + 2\mu + \frac{\delta e/e}{\delta L/L} \quad (5)$$

The fact that  $S$  has values ranging from less than 1 to 3.5, depending upon the wire material, indicates that  $e$  is undoubtedly a function of strain. From the experimental observation that  $S$  is constant over a wide range of strain and from (5) it is deduced that  $\delta e/e$  is probably a linear function of strain. Attention is called to the fact that no data are available on the value of  $\mu$  for small diameter specimens.

Another important characteristic of the wire gage is its response to temperature changes. The resistance of most metallic wires is quite responsive to temperature and means of compensating for this phenomenon are necessary. This problem is not solved by using a wire material of zero temperature coefficient of resistance or by using a bimetallic gage, for it must be remembered that the gage is attached to a metallic surface and that the gage resistance is therefore affected by the differential expansion of the gage and the test material. The simplest and most practical means of eliminating thermal effects is through the use of a dummy gage mounted on the same material and kept at the same temperature as the active gage. The active and dummy gages are then wired in adjoining arms of the bridge circuit, balancing out the thermal effects electrically.

Table 1 lists the important characteristics of four wire materials, the last three of which are widely used in the fabrication of strain gages:

TABLE 1

## Properties of Wire Materials

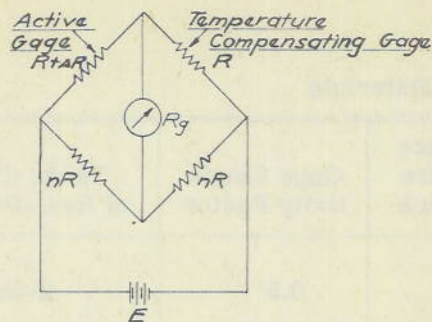
| Material    | Composition<br>%                             | Resistance<br>.001 Wire<br>Ohms/Inch | Gage Sensi-<br>tivity Factor | Temp. Coef.<br>of Res.-Per °C |
|-------------|----------------------------------------------|--------------------------------------|------------------------------|-------------------------------|
| Manganin    | 84 Cu<br>12 Mn<br>4 Ni                       | 22                                   | 0.5                          | $2-5 \times 10^{-6}$          |
| Advance     | 55 Cu<br>45 Ni                               | 25                                   | 2.0-2.1                      | $2-10 \times 10^{-6}$         |
| Nichrome    | 80 Ni<br>20 Cu                               | 50                                   | 2.1-2.3                      | $170 \times 10^{-6}$          |
| Iso-Elastic | 36 Ni, 8 Cr<br>C, 4Mn, Si, Mo, Va<br>Bal. Fe | 40                                   | 3.5                          | $300 \times 10^{-6}$          |

## CONSTRUCTION OF GAGES

The two main methods of fabricating gages are: (1) by drawing the wire into a zig-zag by means of two linear sets of meshing hooks, and (2) by winding the wire around a flat plate or a circular cylinder which is subsequently flattened. The gage is then attached by means of a suitable cement to a thin piece of rice paper and leads are welded or soldered to the gage ends. Usually the exposed side of the gage is then covered by another layer of paper. Each method of fabrication has its own advantages. With the zig-zag gage, the wire lies in one plane and is separated from the test member by only one thickness of paper. However, because of the length of transverse wire in the end loops, the gage sensitivity factor is reduced slightly. Further, greater difficulty is encountered in making this factor consistent for a batch of gages. With the wound type of gage, consistency of gage sensitivity factor is more easily attained, but the gage is thicker and the wire lies in two planes separated by one or two thicknesses of cement impregnated paper.

## STRAIN GAGE CIRCUITS

The basic electrical arrangement of strain gages for both static and dynamic work is the conventional bridge circuit. The active and dummy gages are placed in adjoining arms, the other two arms being fixed resistances shunted by a potentiometer for initial balancing. The change in resistance of the active gage and hence the strain to which it is subjected may then be determined either by measuring the unbalanced voltage or current or by measuring the amount by which the resistance of one of the other arms must be changed to restore balance. For static work both systems are used, the latter being preferable because of its independence of bridge voltage and because of the availability of multipoint self-balancing recording potentiometers which permit the automatic scanning of more than 100 gage points during each load increment.



$$I_g = \frac{E \Delta R / R}{2(2R_g + R + nR)}$$

$$\text{FOR } R_g \rightarrow \infty$$

$$E_o = \frac{E \Delta R}{4 R}$$

Fig. 1 - D-C strain gage bridge

In dynamic test work the short duration and rapid variation of the strains do not permit use of the null balancing systems and the method usually employed is to record the bridge unbalance. The bridge circuits used in aircraft test work are classified as d-c or a-c systems depending upon the nature of the electrical power supplied to the bridge. The d-c circuit is the simpler and is usually the more accurate system. As shown in Figure 1, the essential elements are the bridge network, two or four arms of which are strain gages, a battery and a sensitive galvanometer. Expressions for the galvanometer current and for the open circuit output voltage are also given. Both of these quantities are proportional to the unit resistance change  $\Delta R/R$  and hence, via the gage sensitivity factor, to the strain. For very low frequency strains such as occur during dives, a well-balanced microammeter having an internal resistance less than 100 ohms and a range of 200 microamperes is a suitable and inexpensive detector. In actual

practice, a bank of 10 to 15 such meters mounted within the fuselage may be photographed by a single camera, thus providing a simultaneous record of a relatively large number of strains. In landing tests, catapult operations and in laboratory drop tests of aircraft, the important component frequencies are well below 100 cps. For the range from 0 to 100 cps, it is still possible to use the simple d-c system. Electromechanical galvanometers are available with sensitivities as high as 35 microamperes per inch deflection of the oscillogram trace and natural frequencies as high as 160 cps. However, under many conditions of shock, the galvanometers themselves are affected by the accelerations and give spurious readings. These highly sensitive galvanometers are more suitable for laboratory tests where it is not necessary to accelerate the oscillograph. Where the oscillograph cannot be isolated from the shock conditions under investigation, it becomes necessary to use higher frequency, less sensitive galvanometers which require preceding amplifiers. Amplifying systems are also necessary for all work involving frequencies above 100 cps. Coil galvanometers are commercially available with natural frequencies up to approximately three thousand cycles per second. For a flat frequency response to higher than 1500 cps, it becomes necessary to use cathode ray oscillographs.

Since d-c amplifiers are quite troublesome, particularly in field use, it is usual in aircraft work to excite the bridge with a-c by means of an audio-frequency oscillator and to amplify the output by means of an a-c amplifier. This arrangement is shown in Figure 2. In this type of circuit, known as a "carrier" system, the frequency requirements placed on the amplifier are very mild. For example, if it is desired to measure all frequencies up to 200 cps, a carrier frequency of 2000 cps will be used, but it is only necessary that the amplifier have a flat frequency characteristic in the band of  $2000 \pm 200$  cps. Other important characteristics of amplifiers in flight test work are small bulk, freedom from microphonic components, and stability. A twelve-channel carrier system operating at 2000 cps is shown in Figure 3. Each of the two power supplies houses a 2000 cps oscillator which supplies 10 volts to each of six

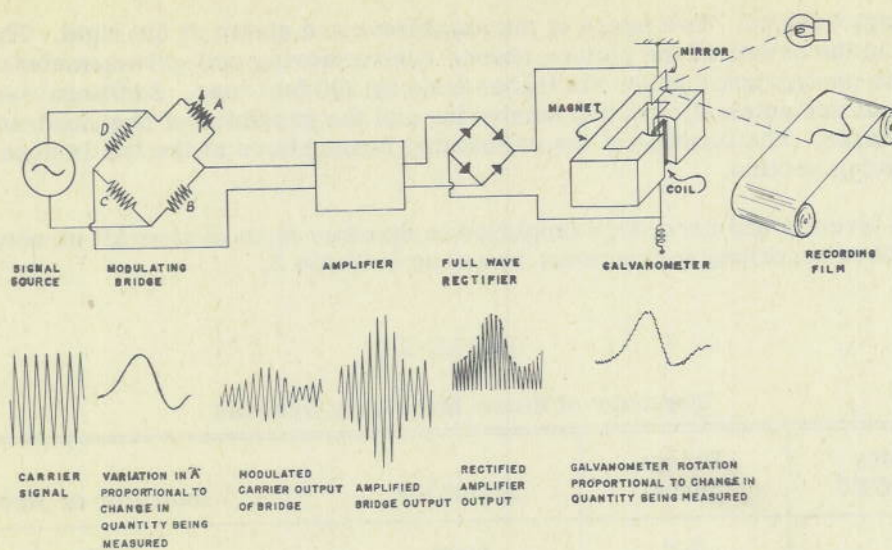


Fig. 2 - Basic principles of the carrier system

Insert as Fig. 3 p. 17

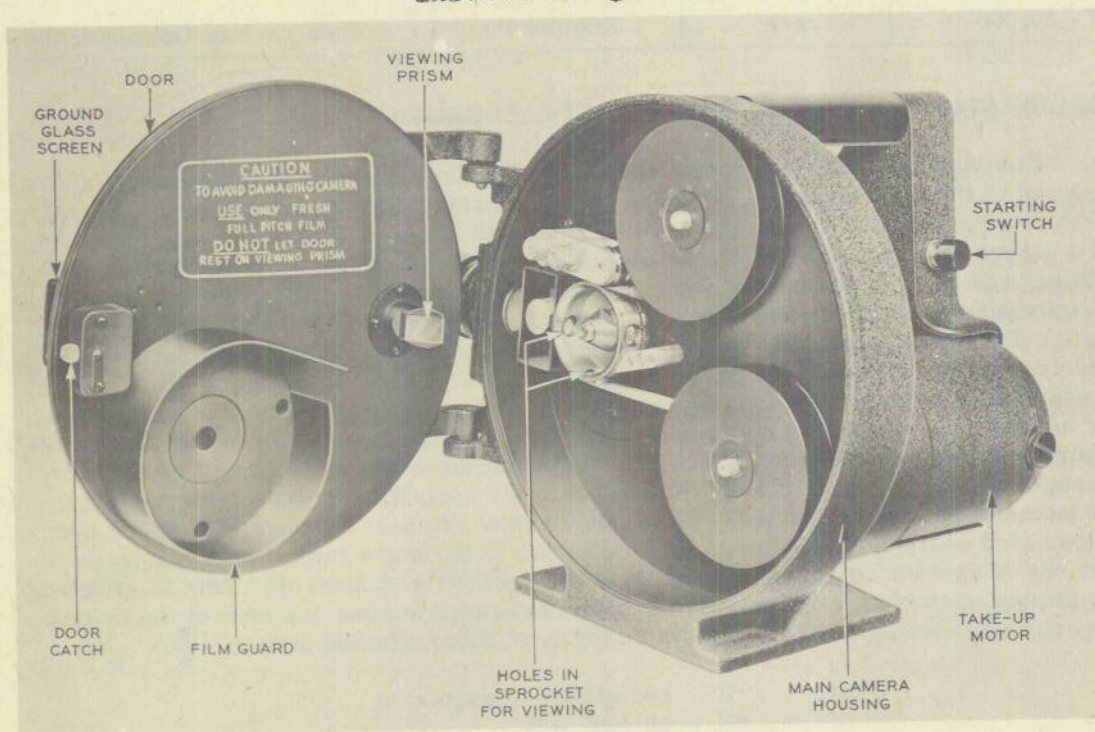


Fig. 3 - Wm. Miller carrier equipment with NAES automatic calibrators

strain gage bridges. Two banks of six amplifiers are shown at the right. The oscillograph in the center of the picture houses twelve moving coil galvanometers which record on photographic paper six inches wide by 200 feet long. Additional features of this unit are automatic record numbering and the provision of one-hundredth second timing marks. The function of the calibrating boxes shown at the left is described in the following section.

The circuits and recorders employed in dynamic tests of aircraft structures are summarized according to frequency response in Table 2:

TABLE 2  
Summary of Strain Measuring Systems

| Frequency Range - CPS | Bridge Input | Amplifier | Indicator or Recorder                        |
|-----------------------|--------------|-----------|----------------------------------------------|
| 0 - 1/2               | d-c          | none      | Microammeter                                 |
| 0 - 100               | d-c          | none      | Moving Coil Galvanometers With Optical Lever |
| 0 - 1,500             | a-c          | Required  | Moving Coil Galvanometers With Optical Lever |
| 0 - 20,000            | a-c          | Required  | Cathode Ray Oscilloscope                     |

#### CALIBRATION OF WIRE RESISTANCE STRAIN GAGES

The calibration of wire strain gages is complicated by the fact that, once cemented to the test specimen, the gage cannot be removed without damage. The calibration procedure to be followed will depend upon the objective of the measurement. If a direct determination of the average strain over the gage site is desired, and if the gage is in a convenient location, it is possible to effect calibration by comparison of galvanometer deflections with the readings of a superimposed standard gage such as the Tuckerman optical strain gage. This procedure is a direct calibration and involves no prior knowledge of the gage sensitivity factor. If, as more often occurs, a calibration in situ cannot be accomplished, a reasonable procedure is to measure, or accept the manufacturer's measurement of, the gage sensitivity factor of a small number of gages out of each batch. Since this factor is the ratio of unit resistance change to strain, it remains only to calibrate the recording or indicating equipment in terms of deflection per unit resistance change. In fact, the ease with which the associated equipment may be calibrated is one of the more important advantages in the use of resistance gages, particularly in flight tests of aircraft. This is effected by simply shunting a precisely known high resistance across one arm of the bridge thereby introducing a unit change in resistance of the affected arm of  $\frac{\Delta R}{R} = \frac{R}{R + R_S}$ ,

where  $R$  = resistance of gage arm  
 $\Delta R$  = change in  $R$   
 $R_S$  = value of shunt resistor

In most dynamic investigations undertaken in recent years at NAES, the determination of the exact strain at the gage site has not been important. In tests such as arrested landings the knowledge of the strain at a relatively small number of points is not very helpful in evaluating a structure as complicated as an airplane. At present, 24 is the practical upper limit to the number of dynamic strains which can be recorded simultaneously in fighter-type aircraft. For this reason the attempt is usually made to measure forces and moments rather than strain. For example, it may be desired to obtain data on wheel loads during arrested landings or stick forces and elevator hinge moments occurring during a pull-out. In such typical examples, the gages are placed on existing or inserted structures so as to respond only to the desired load. Calibrations are then effected by the application of known loads. At the time of load calibration as well as immediately before and after each test condition, shunt resistance calibrations are made. This procedure calibrates all components of the measuring system except the gage and thus provides the data for correcting for changes in oscillator level, amplifier gain and galvanometer sensitivity. Over a period of three months, which is the approximate duration of many tests, over-all changes in the sensitivity of these three items may be as much as 10 percent even with the best available electronic equipment. Therefore without a resistance calibration procedure, frequent load calibrations would be required. This would multiply by several times the length of time and number of man-hours required for completion of a test program. Automatic resistance calibrators have been made an integral part of almost all strain recording equipment in use at NAES. Immediately before and after a test condition, the pilot presses a button which after initiating action of the oscillograph, automatically shunts one arm of each bridge with a prechosen resistor. A typical resistance calibration record is shown in Figure 4. This particular record was obtained from a pilotless aircraft and recorded on the ground by means of telemetering equipment developed at NAES. This calibration forms the basis for determining the accuracy of the transmitted data.

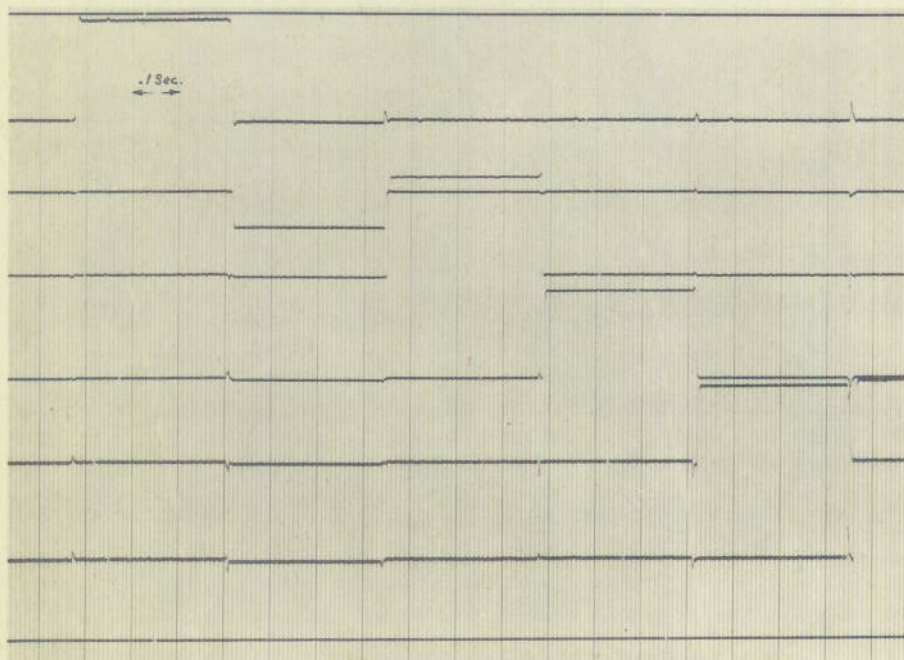


Fig. 4 - Automatic resistance calibration

## APPLICATIONS TO AIRCRAFT TESTING

In the demonstration landings of a new single-seat seaplane, it was desired to measure the forces and moments applied to the fuselage. In the airplane under consideration the loads are applied to the fuselage through a W-type arrangement of four struts. An analysis of the geometry and end fixity conditions showed that it was possible to determine the three forces and three moments applied to the fuselage by measuring the axial loads on all four struts and side bending moments on two struts. Figure 5 is a photograph of the seaplane in a loading apparatus. Figure 6 shows the disposition of the strain gages on the two most forward struts. Axial loads are measured on each of these members by placing gages parallel to the longitudinal axis and symmetrical with respect to the neutral bending axis. A like number of gages are placed perpendicular to these gages as may be seen on the diagonal strut and on the vertical strut, center gage set. All the axial gages are wired in one arm of the bridge; all the transverse gages in an adjacent arm. This provides added sensitivity as well as temperature compensation. The top and bottom gage sets on the vertical strut measure side bending moments at their respective stations. The gages

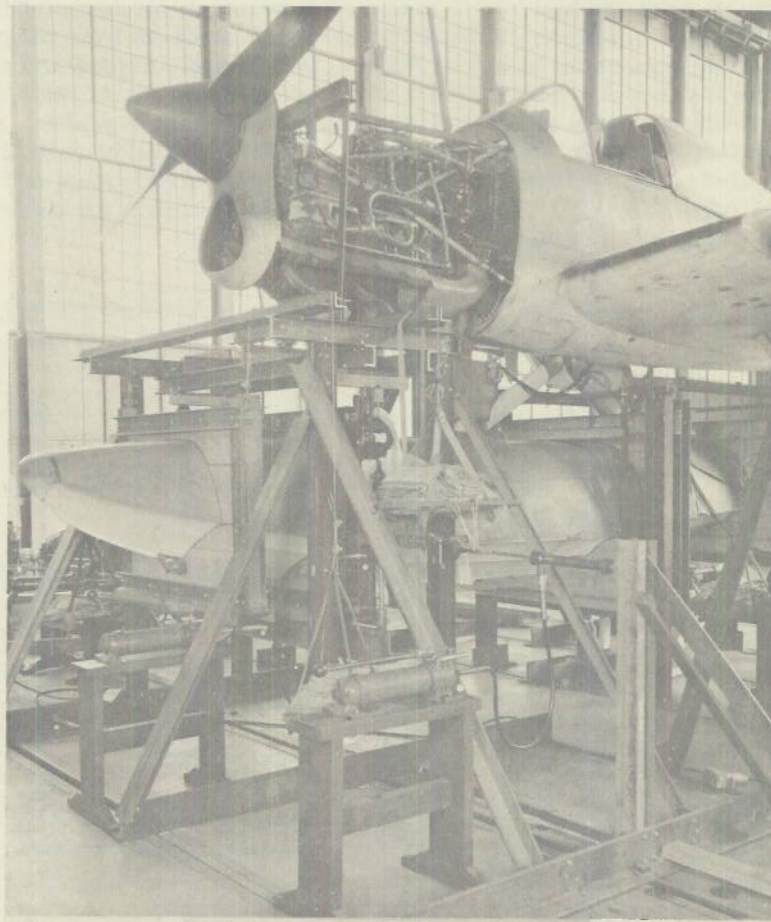


Fig. 5 - Load calibration of Model XOSE-1 airplane

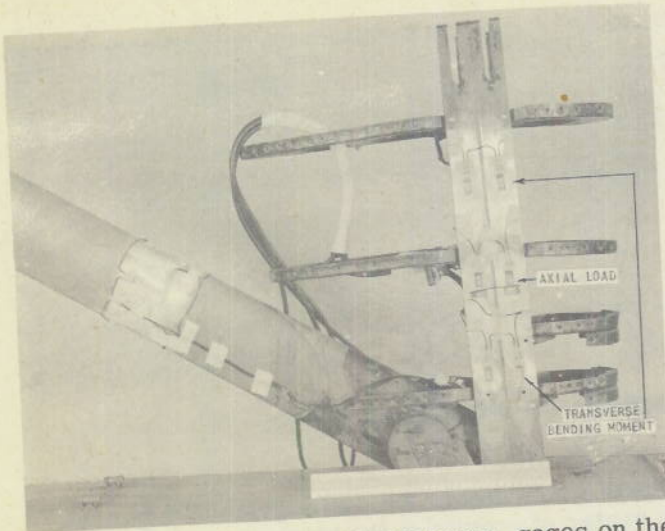


Fig. 6 - Gage installation -  
forward struts

in view are placed in one bridge arm, gages on the opposite side of the strut are in an adjacent arm. This arrangement is not responsive to axial loads or to temperature changes. Protection of gages against moisture and salt water is an important item in tests of seaplanes, particularly for external gage sites. The gages were attached to the structure with Duco cement, permitted to dry for 24 hours and then covered with heated petrosene wax. The struts were then bandaged with 2-inch aircraft tape and painted with aircraft dope. After this dried, the entire gage area was covered with several coats of neoprene cement, each coat having been subjected to the heat of infrared lamps for several hours. Finally the struts were covered with metallic shields as protection against the hydraulic action of the water. (Figure 7.) No malfunctioning of the gages

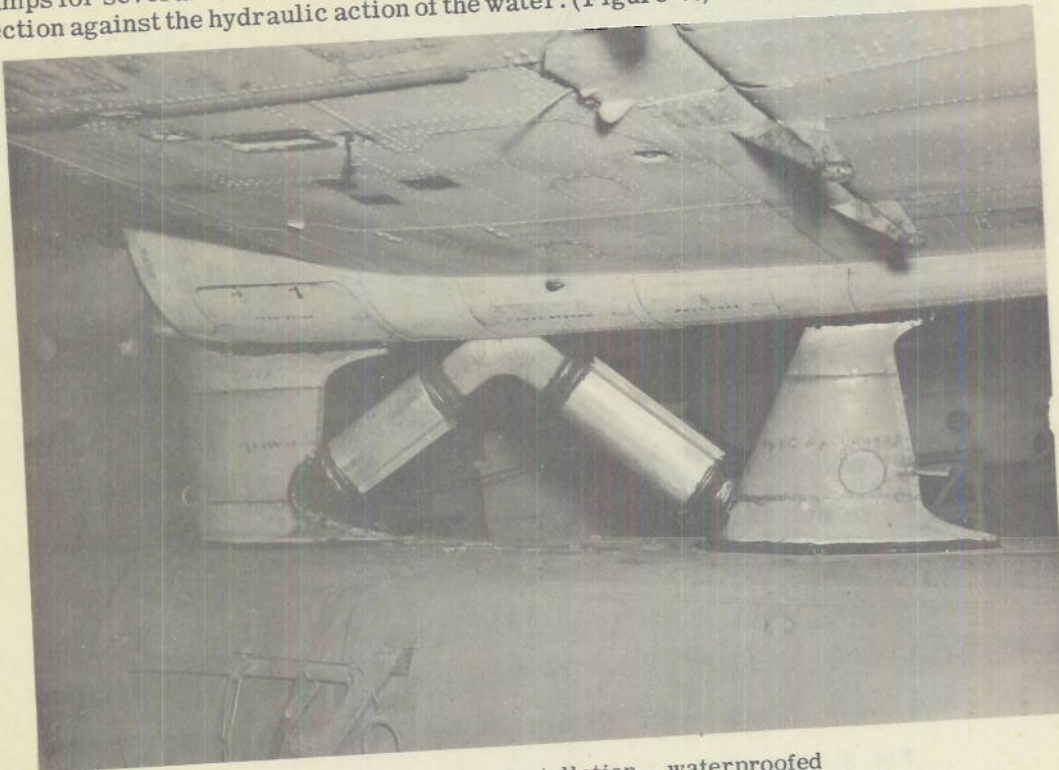


Fig. 7 - Gage installation - waterproofed

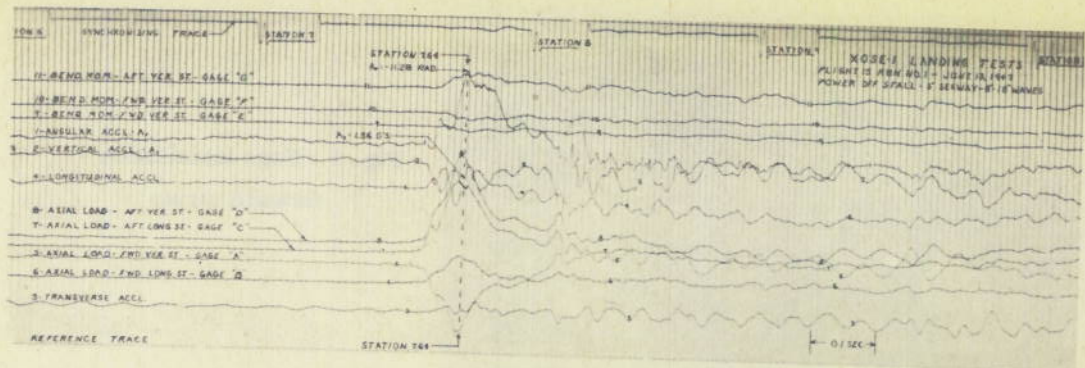


Fig. 8 - Typical oscillogram - XOSE-1 water tests

has occurred in more than six months of water tests. An oscillogram of a typical landing showing accelerations of the center of gravity of the airplane as well as strut loads is shown in Figure 8.

In an investigation of the external loads applied to aircraft during catapult launchings, strain gages were attached to the landing gear of an F4U-1 airplane and to links which were inserted in the catapult tow bridle (Figure 9). Calibrations, made by applying known loads in the laboratory, enabled the determination of wheel reactions and catapult towing forces. An oscillogram of a launching is shown in Figure 10.

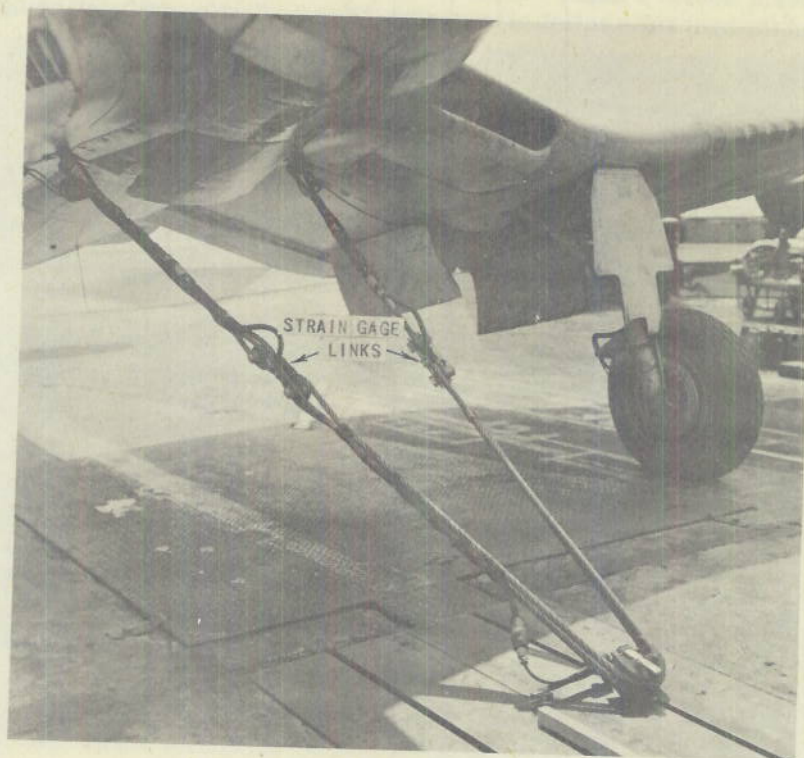


Fig. 9 - Catapult tow bridle - load measuring installation

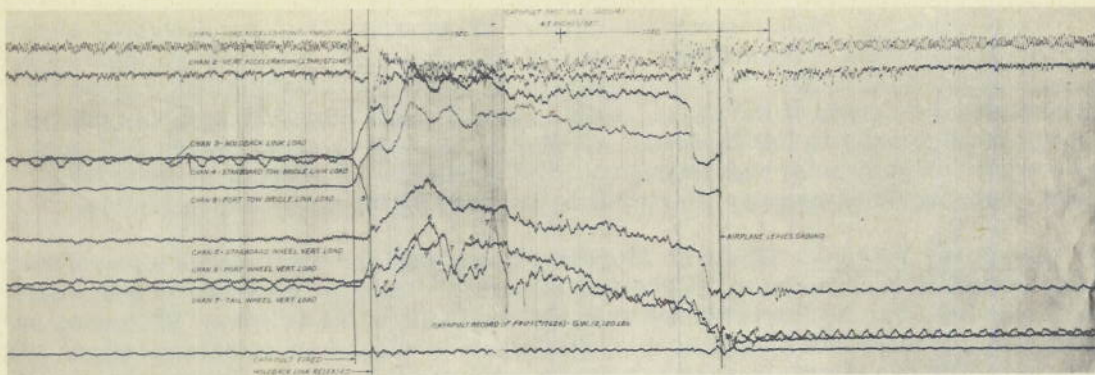


Fig. 10 - Oscillogram of catapult launching - wheel and tow bridle loads

At the NAES variable resistance gages have been applied to many specialized measuring devices, a few of which are briefly described below:

**Dynamic Reaction Platform.** This device (Figure 11) was designed to measure impact loads occurring during drop tests of landing gear. It consists of an equilateral platform supported by three tubular pedestals to which gages are affixed to respond only to axial loads. The calibration of this device is extremely linear and is constant over long periods of time.

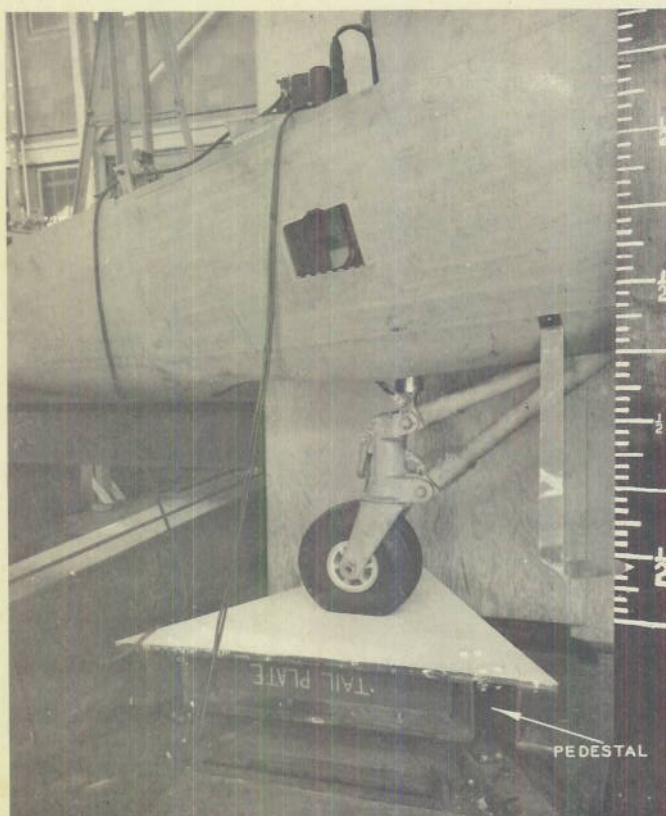


Fig. 11 - Dynamic load platform

Accelerometer. This instrument (Figure 12) consists of an end-weighted double cantilever supported at the center. A total of four gages are mounted symmetrically on both sides of the beam. This arrangement balances out the effects of transverse forces along the length of the beam. With a natural frequency of 20 cps, this unit responds satisfactorily to low frequency accelerations such as occur during pull-outs but discriminates against high frequency local accelerations induced by the power plant. Damping to 65 percent of critical is provided by the use of silicon fluids.

Air-Speed Pick-up. This unit (Figure 13) designed for oscillographic recording of air speed, is responsive to differential pressure. The high pressure is introduced within the bellows, the low pressure around the outside of the bellows. Expansion or contraction of the bellows deflects two cantilevers to which four resistance gages are attached. The two discs are used to provide mechanical balance. The response of this device is almost entirely free from hysteresis and from vibration and temperature effects. A change in temperature of  $56^{\circ}\text{C}$  results in no measurable zero shift and in a 2 percent reduction in sensitivity. Its range is from zero to 700 miles per hour (0 to approximately 22 inches of mercury). This unit was manufactured by Baldwin-Southwark to N.A.E.S. specifications.

Pressure Pick-up. Figure 14 is a device for measuring short duration pressure transients such as occur in aircraft hydraulic systems and powder-fired catapults. Resistance gages are used to measure circumferential strains. The gages at the closed end provide temperature compensation. Units having ranges up to 10,000 pounds per square inch have been constructed.

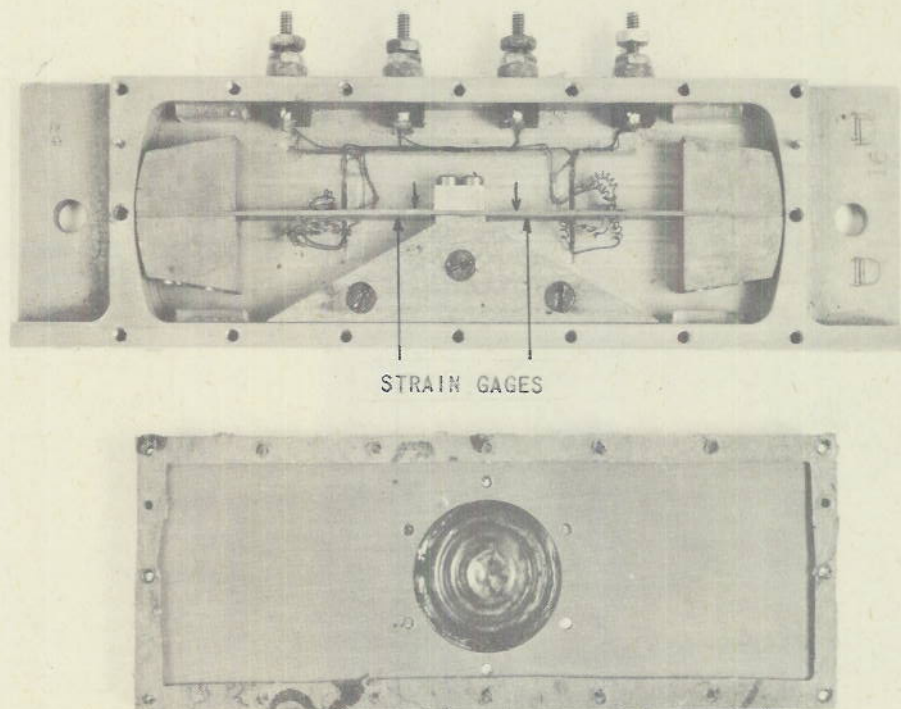


Fig. 12 - Bonded strain gage accelerometer

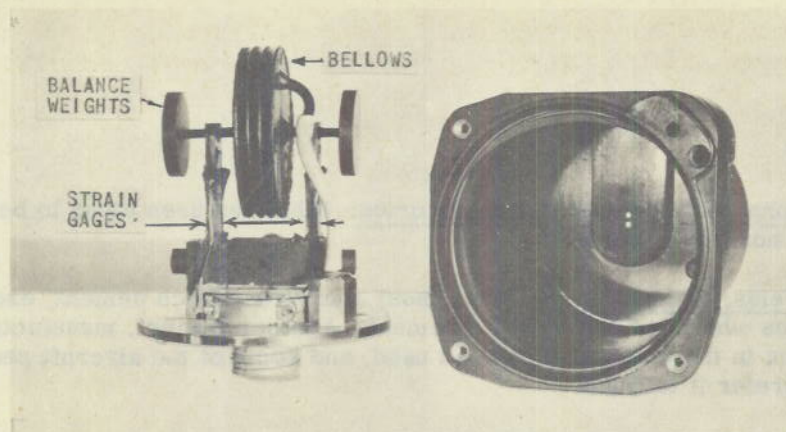


Fig. 13 - Air speed pickup

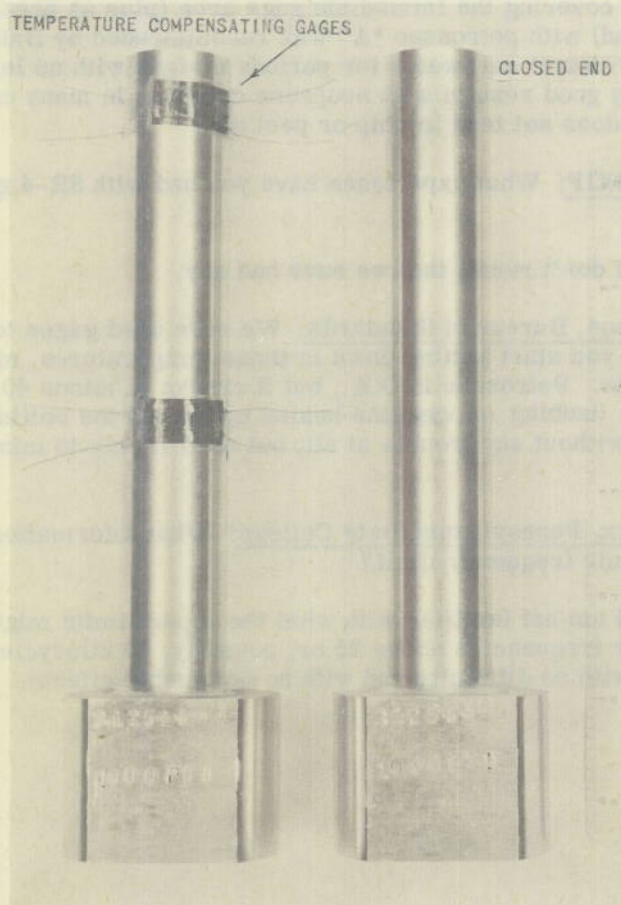


Fig. 14 - Pressure pickup

## DISCUSSION

G. R. Gohn, Bell Telephone Laboratories: What has been found to be the most satisfactory bonding material?

D. E. Weiss, NAMC, NAES: We almost always use Duco cement, except on bakelite gages where we use bakelite cement. Another cement, manufactured by a radio concern in the midwest, has been used, and some of the aircraft people on the west coast prefer it to Duco.

W. H. Watkins, Puget Sound Naval Shipyard: Have you had any experience with waterproofing gages for underwater work?

D. E. Weiss: We have run tests on seaplane landings. I mention this because it comes close to underwater work (specimens stay under water for a length of time). We have made these investigations in the laboratory and may be idealizing conditions, but have found that covering the immediate gage area (plus an area of about 1 inch or 2 inches beyond) with petrosene "A" wax recommended by Baldwin Southwark will permit submersion to salt water for periods of days with no leakage. We have experienced equally good results with neoprene cement. In many cases, the latter is preferred, as it does not tend to chip or peel off.

A. Leventhal, NGF: What experience have you had with SR-4 gages at low temperatures?

D. E. Weiss: I don't recall that we have had any.

A. E. McPherson, Bureau of Standards: We have used gages to 40 degrees below. As soon as you start getting down in those temperatures, moisture is condensed out of the air. Petrosene is O.K., but it cracks at minus 40 degrees. We obtained results by daubing on vaseline heated up beyond the boiling point of water. We could put it on without any trouble at all, but used it only to minus 40 degrees, however.

Eugene Von Arx, Pennsylvania State College: What information have you concerning the ultrasonic frequency limit?

D. E. Weiss: I am not familiar with what the upper limits might be. We have rarely used carrier frequencies above 25 or, possibly, 30 kilocycles but have used them to that limit with no difficulty and with no noticeable effects.

## MEASURING SHOCK WITH HIGH SPEED MOTION PICTURES

by

J. T. Muller, BTL

This paper briefly describes and illustrates the operation of the high speed motion picture camera, the preparation of the test object, and precautions to be considered when films are taken. The difficulties encountered in obtaining numerical data as well as the advantages and disadvantages of the high speed motion picture film for shock work are enumerated.

### INTRODUCTION

This presentation is a general outline of the technique involved when high speed motion pictures are used to record the motions associated with high impact shock.

High impact shock is the name given to the action which hammer blows produce in a suitable machine to test the mechanical sturdiness of certain types of apparatus. A more precise definition is, that shock is the physical manifestation of the transfer of mechanical energy at extremely short intervals of time. The order of magnitude of the time interval is milliseconds and quite frequently microseconds. A transient motion is always associated with such a transfer of energy, and since the time displacement curve is a parameter easy to visualize and to work with, it has been chosen as the final indication to be recorded.

Although the analysis of shock phenomena is usually confined to one direction only, which is a perfectly plausible method to obtain an insight into the behavior of the apparatus under test, it is never experienced when a shock is produced on our high impact machines. On the contrary, the equipment may bounce around in all directions and go through many modes of motions. Granted there is a major direction, it is nevertheless important to know about the minor ones.

### DESCRIPTION

An easy way to perceive these complex motions is to observe them visually, and in order to accomplish this we have to resort to motion pictures which can reproduce the action in slow motion. Long before high impact machines, the Bell Laboratories had developed a high speed motion picture camera (known as the Fastex) which seemed to be ideally fitted to do the job of recording shock displacements. Exterior and interior views of the Fastex are shown in Figures 1 - 5.

In Figure 6 we have a drawing showing the use of a prism to obtain an image on the film; in other words, the camera belongs to the rotating prism type.

Either 8 or 16mm film can be used. The average maximum exposure is 4000 frames per second for 16mm and 8000 frames per second for 8mm. For shock work we rarely, if ever, use the 8mm. We have found that 16mm at 4000 frames is quite satisfactory.

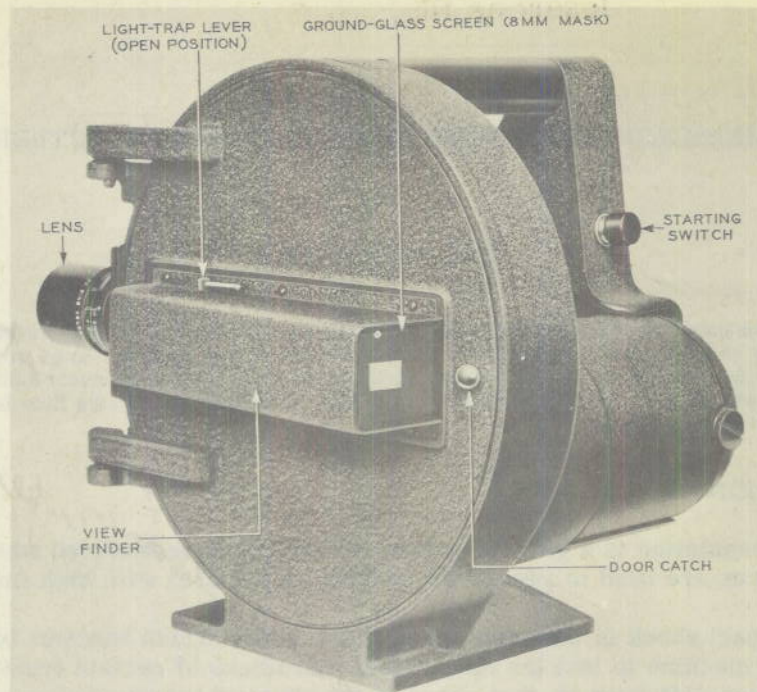


Fig. 1 - Exterior view from the cover side

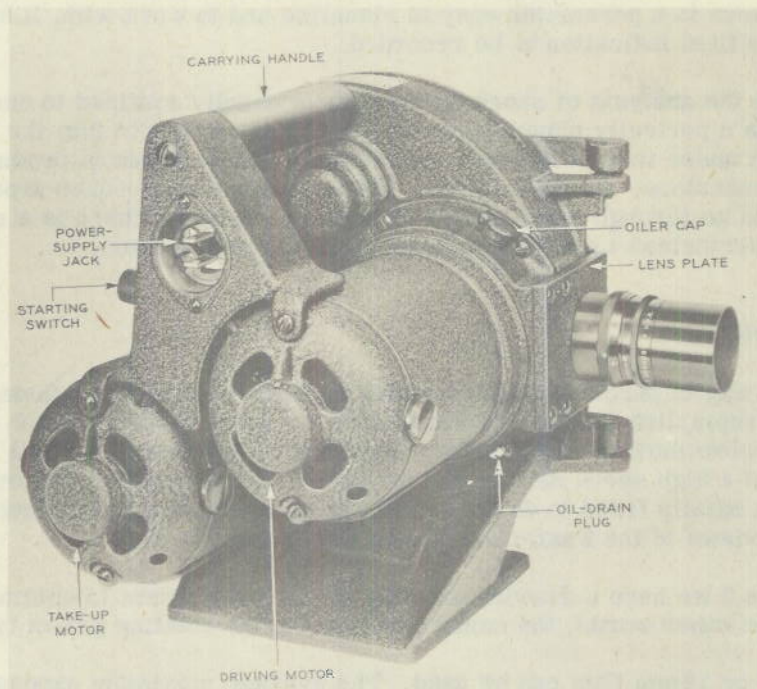


Fig. 2 - Exterior view from the motor side, showing the drive and take-up motor

Insert as Fig. 3 p. 5



Fig. 3 - Interior view with the cover open

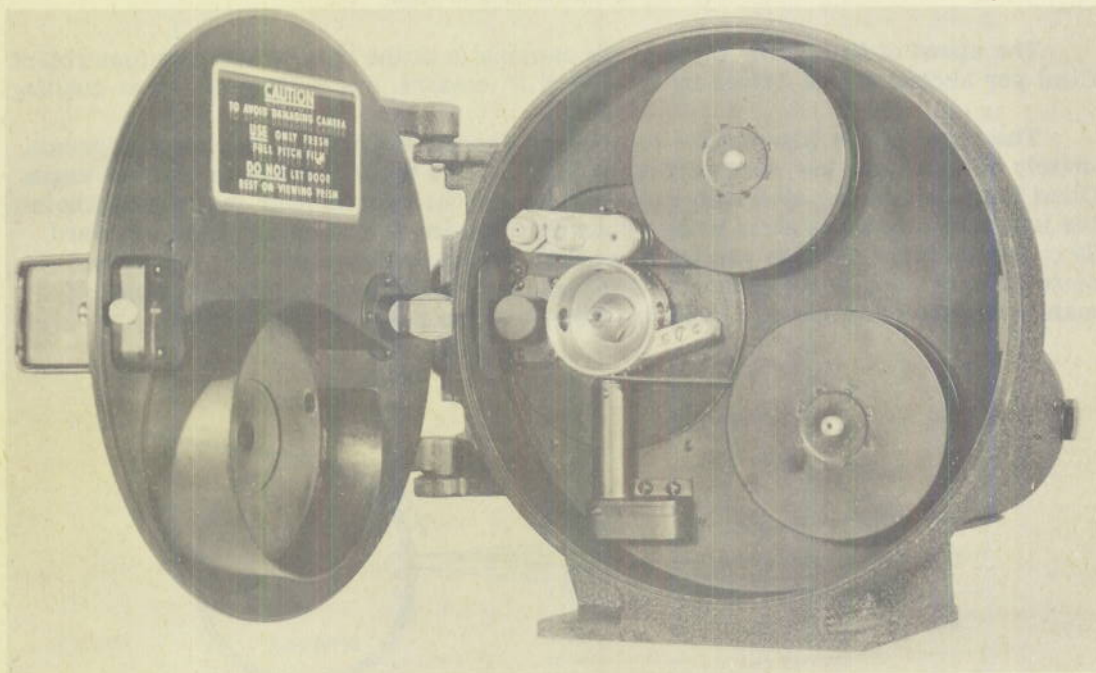


Fig. 4 - Interior view with the timing light in place

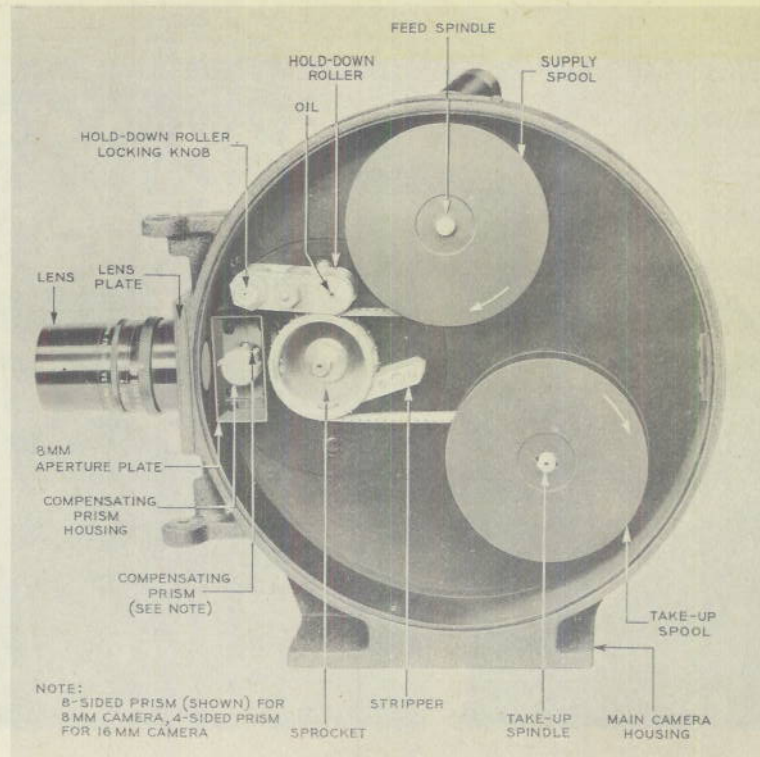


Fig. 5 - Detailed inside view

The speed of film going through the camera is at the rate of 100 feet (one roll of film) per second and is driven by two  $1\frac{1}{4}$  H.P. motors, one pulling the other pushing.

The speed of the film varies somewhat, starting at 2500 and ending at approximately 5000 frames per second (Figure 7). To time the film there is a small argon filled lamp, producing alternate white streaks on the margin near the sprocketholes, the length of this white strip being accurately related to one cycle of the standard 60 cycle current. To give you an idea of the exposure time involved, it may be stated that of the  $\frac{1}{4}$  millisecond or 250 microseconds between frames, approximately 80 microseconds are used for exposure, the rest for film transit.

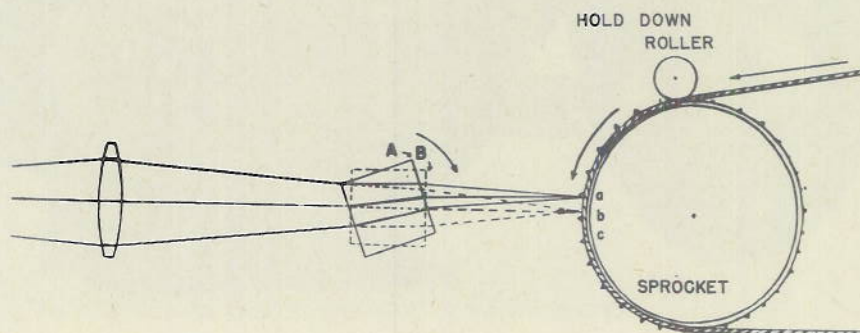


Figure 6

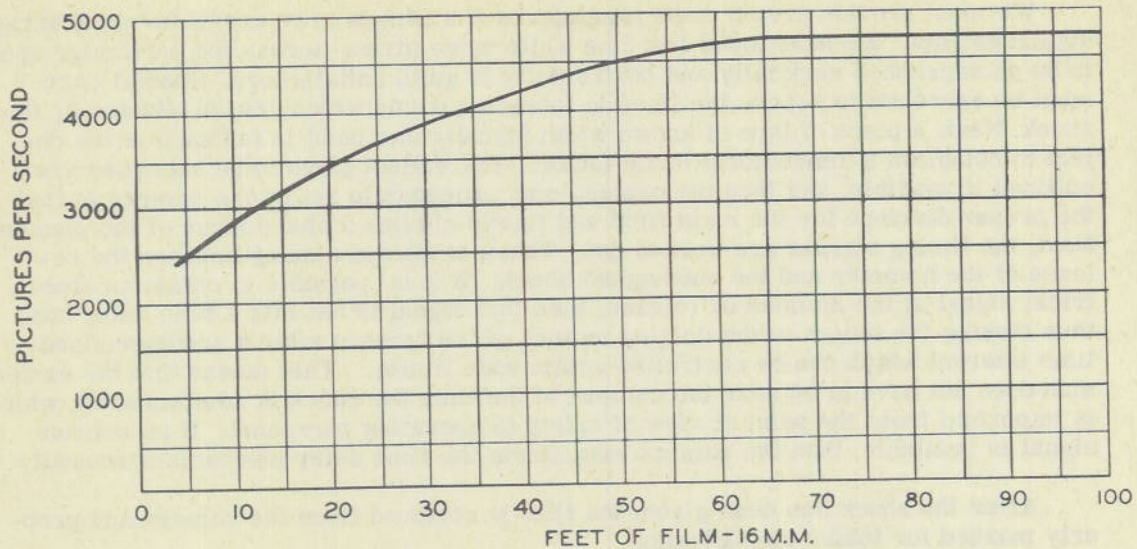


Fig. 7 Speed Characteristics at 130 Volts ac

One of the secrets of good photography is proper illumination of the object, and high speed motion pictures are no exception. It stands to reason that since the exposure time is so extremely short, the intensity of the light source must be great. Special projection lamps are used which are capable of delivering approximately 10,000 to 100,000 foot candles. About three of these spotlights are used and the general camera setup, with lights in place, is shown in Figure 8.

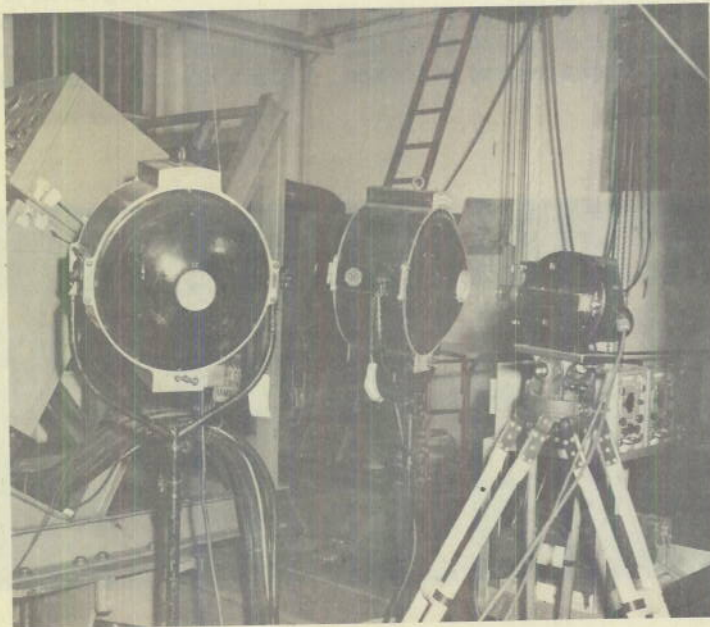


Figure 8

We must provide proper reference marks if a film is to be useful for measuring displacements. We have found that fine white twine strung across the particular spot to be photographed vertically and horizontally is quite satisfactory. Special care must be exercised to secure the lines to fastening points which are not affected by the shock. Next, a piece of tape of known width (usually one inch) is fastened to the object to establish a dimensional scale factor. The salient parts to be recorded are outlined if possible, and then the camera man proceeds to set up the camera to find the proper distance for the right field and finally obtains a sharp focus of the picture. Next, the timing signals are worked out. There is always a delay between the release of the hammer and the subsequent shock. If it is possible to obtain an electrical signal at the moment of release, then this signal is fed into a time delay box, thus closing the switch to the driving motors of the camera after a predetermined time interval which can be controlled within wide limits. This means that the camera man does not have to be near the camera at the time the shock is administered, which is important from the point of view of safety to operating personnel. If no release signal is available, then the camera man starts the time delay mechanism manually.

After the shock has been given, the film is removed from the camera and properly marked for future identification.

After the films have been developed they are first examined for (1) any unusual occurrences, or (2) to demonstrate some mysterious happenings which could not be explained satisfactorily at the time of the shock. Next comes the procuring of specific data, which in our case means the measuring of the displacement of a particular part with respect to the reference lines. To examine the film frame by frame we use a commercially available projector called a Recordak, which will enlarge the picture approximately ten times. We first mark off that portion of the film which is to be measured and establish the time interval between two successive frames by means of the marking on the side of the film. (Figure 9). We simply establish the length of one cycle and count the number of frames. The action we are interested in is usually not very long, so that this frame interval determination does not have to be repeated (Figure 10). The frames are also numbered from a chosen starting point, marking every tenth frame. Next, the tape is measured to establish the dimensional scale factor which will enable us to find the actual displacement in inches. Then a tabulation is made, measuring

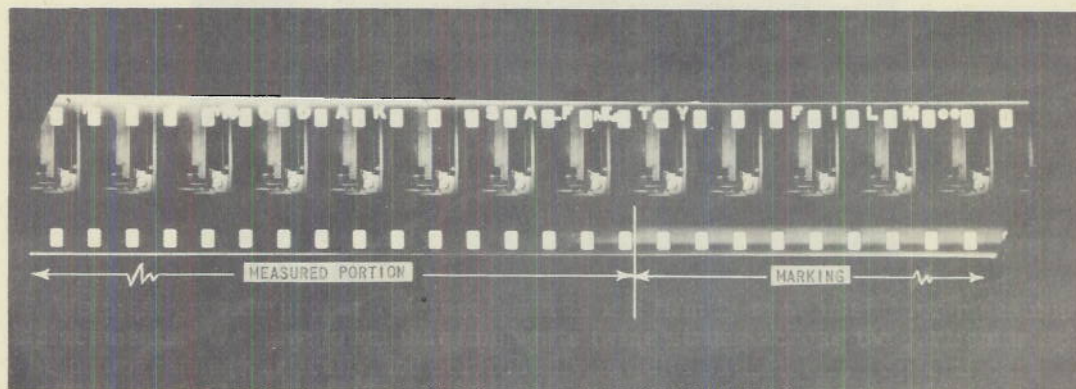


Figure 9

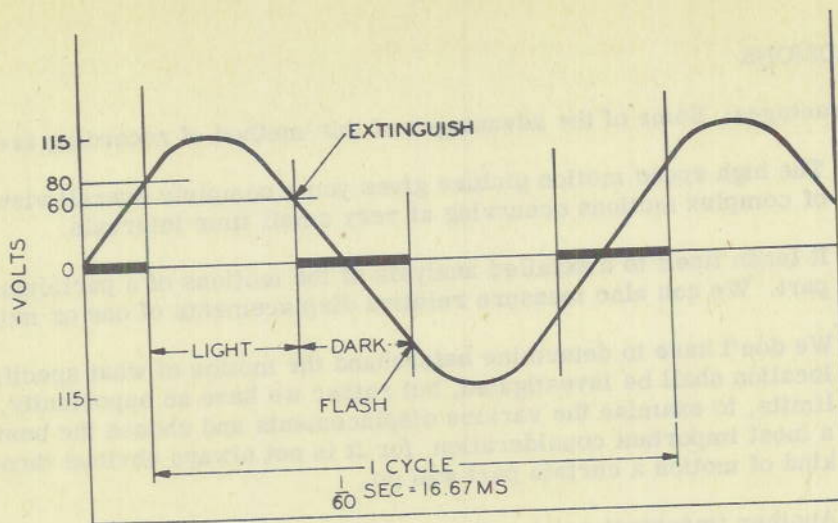


Figure 10

the displacement of one or more parts. Here is where we have run into most of our trouble. It is somewhat hard on the eyes and a tiresome and monotonous job. The tabulated data is then plotted, and the final results are displacement time curves which are then processed further to obtain supplementary information (Figure 11).

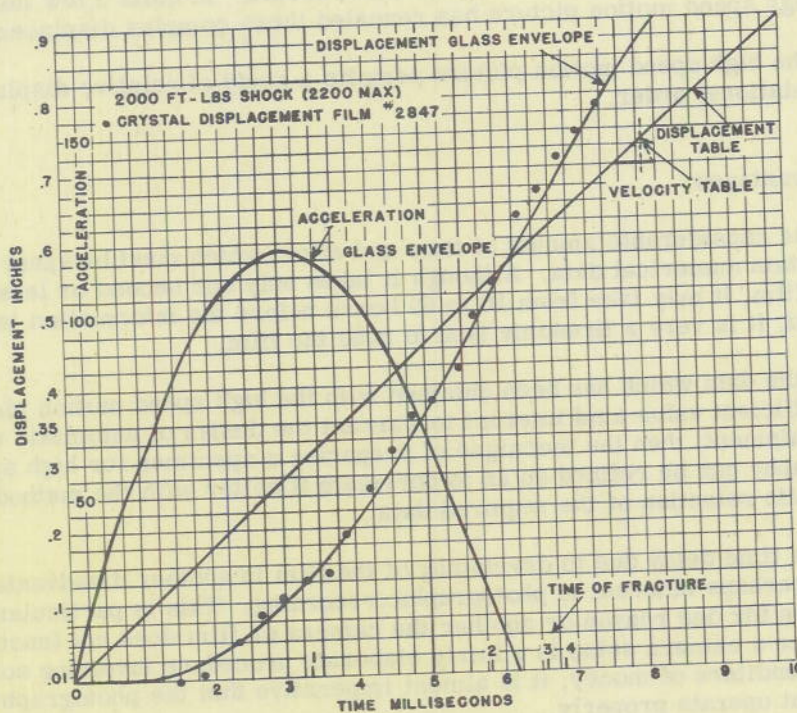


Figure 11

## CONCLUSIONS

Advantages: Some of the advantages of this method of recording are:

1. The high speed motion picture gives you a complete overall view of a group of complex motions occurring at very small time intervals.
2. It lends itself to a detailed analysis of the motions of a particular point or part. We can also measure relative displacements of one or more points.
3. We don't have to determine beforehand the motion of what specific part or location shall be investigated, but rather we have an opportunity, within limits, to examine the various displacements and choose the best. This is a most important consideration, for it is not always obvious through what kind of motion a certain part will go.
4. Another important point and one which is not often realized is the superposition of displacements. By this I mean that a part may have a displacement relative to the fundamental displacement because this part is not rigidly connected to the main body. Most of the radar equipments are highly complex mechanisms, and a cabinet is not a rigid body. It is, therefore, extremely difficult to determine the overall fundamental displacement of such a cabinet by merely taking the displacement measurement of one location. It is this sort of thing which will make "grass" grow on an acceleration record and ripples on a displacement record. In quite a few instances the high speed motion picture has revealed these complex displacements.
5. The high speed motion picture permits a study of relative displacement of a rotational order.

Disadvantages:

1. The considerable amount of time and effort which must be spent in order to obtain numerical data. Although it takes only one second or less to produce a film, it may take from 10 to 50 hours before the information is processed; and, it is very a tiresome task to read the film.

If the data which has been obtained with the high speed motion picture is of sufficient value and interest to warrant the design of automatic reading equipment, then the technique of preparing a specimen for high speed photography can be refined so as to correlate it better with the method of automatic selection of the required data.

2. The time delay due to developing of the film is another disadvantage which is perhaps inherent to photographic recording. This is particularly true when for one reason or another the camera or film does not function properly. Since a camera setup is a fairly elaborate procedure involving some time and expenditure of money, it is almost imperative that the photographic equipment operate properly.

## A BRIEF SURVEY OF THE BRITISH-TYPE VELOCITY METER

by

George Chertock, TMB

The material which is presented on the British-type velocity meter--its construction, its use, and some of its characteristics--is not new. It is based on both British and American sources, but since the use of the instrument is so widespread, a discussion of its features would be of value to all.

This meter is a copy of a British instrument which was first designed in 1940 or 1941. The prototype has been the basic shock measuring instrument used by the English. It was used throughout the Cameron tests and apparently it is still being used on their current full-scale shock tests. In this country it has been used on several ship tests, including the Crossroads test.

Figure 1 is a photograph of the meter with the case removed. The meter consists essentially of two parts: first an electromagnet which is attached by springs to the frame and which is constrained, by rollers, to move in a direction normal to the meter base, and second, a pickup coil which is rigidly fastened to the base and

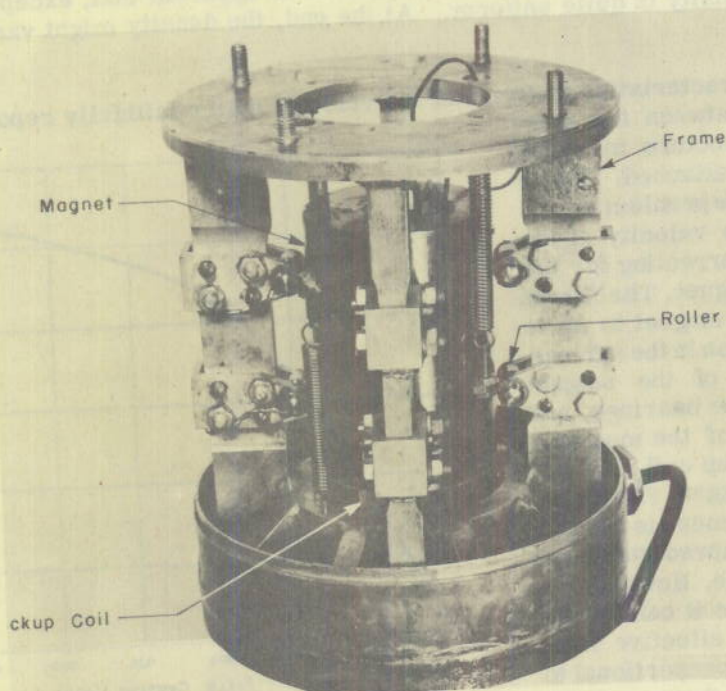


Fig. 1 - British-type velocity meter

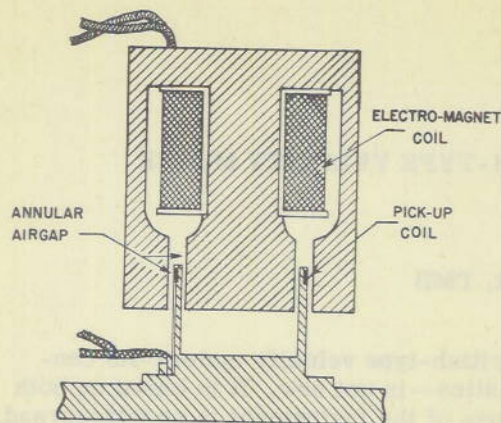


Fig. 2 - Schematic section of velocity meter

string galvanometer without the use of an amplifier, but sometimes a cathode-ray oscillograph is used. Figure 3 shows the experimentally measured flux in arbitrary units as a function of the field current. It is obvious that the curve is fairly flat at 500 ma, so that a ten percent change in the field current, for example, will change the flux by only three percent.

The only cause for error in a measurement of the relative velocity in this meter would be a variation in flux density along the two inch travel length. Figure 4 shows an experimental plot of the total flux vs the distance along the gap in the magnet. The flux density would be the slope of this graph. It is apparent that, except for the end of the gap, the density is quite uniform. At the end, the density might vary as much as ten percent.

These characteristics insure that the velocity meter faithfully reports the relative velocity between the magnet and the structure to which the meter is attached. There still remains the problem of obtaining the true velocity of the structure by correcting for the motion of the magnet. The forces which cause the magnet to move are (1) the tension in the springs, (2) the weight of the magnet, (3) friction at the bearings, and (4) the reaction of the magnetic field on the pickup coil. As will be shown, the biggest shortcoming of this instrument is the fact that friction is unpredictable and difficult to analyze. However, in many applications it can be assumed that the effective frictional forces are proportional to the velocity. In that case, the

which fits inside an air gap in the electro-magnet. Figure 2 is a sectional view which shows the magnet and the pickup coil more clearly.

The length of the cylindrical gap in the magnet is two inches. So long as the flux density remains constant over the length of the gap and so long as the pickup coil stays within the gap, the meter output should vary directly as the relative velocity of the magnet and pickup coil. Ordinarily, the field current is 500 ma and is taken directly from a 12-volt storage battery. With this current, the sensitivity is about 0.37 volts for a relative velocity of one foot per second. The output is usually recorded directly by a

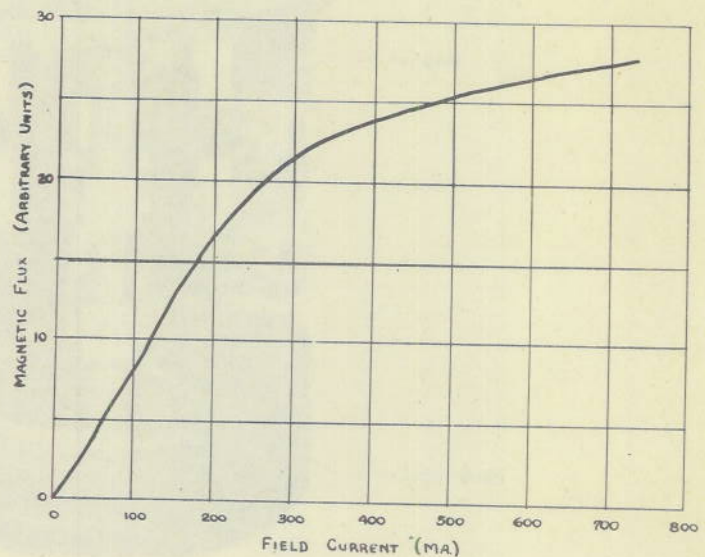


Fig. 3 - Variation of magnetic flux with field current

equation of motion of the magnet can be written as

$$\ddot{x} + 2\omega \dot{x} + \omega_0^2 x = -\ddot{s}$$

where  $x$  is the relative displacement of the magnet and structure

$s$  is the true displacement of the structure

$\omega_0$  is the natural circular frequency of the magnet on its springs and is about 3 or 4 cps multiplied by  $2\pi$ .

$\omega_1$  is a damping coefficient which is about 1 to 5 percent of the value required for critical damping.

If this equation is integrated from the beginning of the shock to some arbitrary time,  $t$ , before the magnet hits the stop at either end of its travel.

Then

$$\dot{x}(t) + 2\omega_1 x(t) + \omega_0^2 \int_0^t x(t)dt = -\dot{s}$$

The first term on the left can be obtained directly from the record; it is simply the relative velocity. The second term is a constant times the relative displacement which can be obtained by integrating the record with an integrator. The third term can be obtained by a double integration of the record. By adding the three terms, the true velocity of the structure can be obtained, which is the term on the right side of the equation.

In practice, the magnitude of the two correction terms as compared with the first term will depend on how the velocity-time curve breaks down in a Fourier analysis. If the spectrum is rich in high-frequency components (higher than, say, 20 cps) then the corrections should be negligible. For lower frequencies, the corrections might be appreciable.

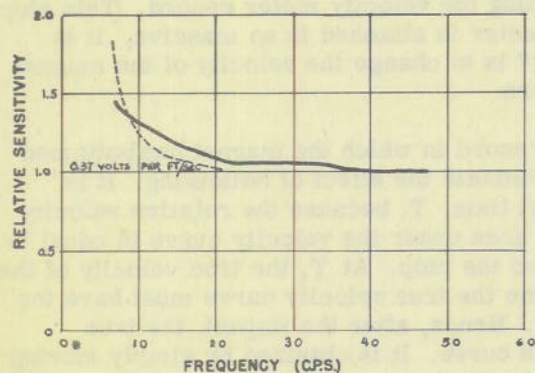


Fig. 5 - Frequency response curve for velocity meter

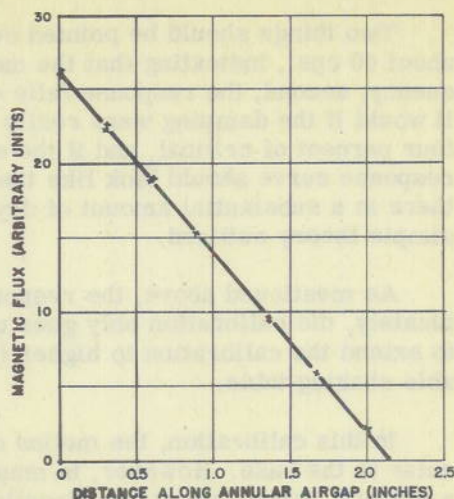


Fig. 4 - Variation of magnetic flux with distance along airgap

Figure 5 shows an experimentally determined frequency-response curve for this instrument. The meter was mounted on a shake table which was vibrated sinusoidally by an electrically driven eccentric. The peak velocity of the table was determined by measuring the frequency and the peak displacement. The voltage output of the meter was measured on a calibrated oscillograph. The resulting calibration is shown in the figure.

Two things should be pointed out: first, the response seems to level off at about 40 cps., indicating that the motion of the magnet is negligible above that frequency; second, the response falls off from its resonance peak more slowly, than it would if the damping were really proportional to the velocity. If the damping were four percent of critical, and if the natural frequency were 4.3 cps, then the frequency-response curve should look like the dotted curve shown in Figure 5. Apparently there is a substantial amount of dry friction which is not taken into account in the simple theory outlined.

As mentioned above, the response seems to level off at about 40 cps. Unfortunately, the calibration only goes up to about 50 cps. It would have been desirable to extend the calibration to higher frequencies, but 50 cps was the limit of the available shaking table.

In this calibration, the motion of the meter was entirely in a direction perpendicular to the base. However, in many cases the velocity meter is used to measure a velocity component in one direction when the structure has an appreciable velocity component in a transverse direction. In such a case it is doubtful if a faithful record will result. The trouble is that the sidewise component of the motion will introduce an excessive and unpredictable friction between the roller and the magnet.

Some British tests, conducted about three years ago, showed how serious an error the friction could introduce. They bolted the velocity meter to the table of a shock machine and made high-speed motion pictures of the motion of the meter case and of the magnet. Then they repeated the tests with the meter inclined at 20 degrees to the shock table. In all cases they found that the relative displacement between magnet and case, as obtained by integrating the velocity-time curve, agreed very well with the relative displacement measured on the film; but the motion of the magnet itself was appreciable. During the first 20 or 30 milliseconds, the magnet moved about 20 percent as much as did the shock table, i.e. when the meter was clamped normal to the table. But in the tests where the meter was inclined at 20 degrees to the shock table, the motion of the magnet was about 30 percent of that of the shock table.

The next point to be discussed is the effect of the limited travel length of only two inches. We are apt to overlook the fact that even though the magnet has "bottomed" (i.e., it has hit and rebounded from the stop at the base of the meter) it may still be possible to continue interpreting the velocity meter record. This stop is so rigid and the structure to which the meter is attached is so massive, it is likely that the sole effect of the "bottoming" is to change the velocity of the magnet without affecting the velocity of the structure.

Figure 6 illustrates a velocity meter record in which the magnet has bottomed and shows how to correct this record to eliminate the effect of bottoming. It is known that the magnet hit the bottom stop at time,  $T$ , because the relative velocity becomes zero at this time and because the area under the velocity curve is equal to the original distance between the magnet and the stop. At  $T$ , the true velocity of the structure must be continuous; after this time the true velocity curve must have the same shape as the curve actually recorded. Hence, after the impact, the true velocity is given by the dotted portion of the curve. It is obtained by simply moving up the recorded curve to eliminate the discontinuity.

The same technique could be used, if the stop was sufficiently rigid, to extend the velocity-meter record beyond the point where the magnet hit the top of the meter.

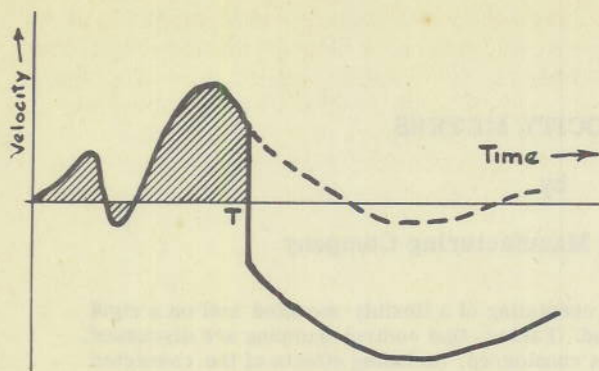


Fig. 6 - Velocity meter record showing correction for bottoming

Ordinarily this upper stop is not very rigid, however, and the time of contact is so long that the velocity of the structure could change appreciably while the magnet and stop are in contact. Furthermore, when the magnet reaches this top stop, the pickup coil begins to move in a region in which the flux density is no longer uniform; hence, the meter output is no longer proportional to the relative velocity.

There is one further point which should be mentioned: the weight of the meter and its effect on the velocity of the structure. The effective weight of the meter is 40 lbs. The magnet itself weighs another 10 lbs.

The base is 8 inches in diameter. If we attach the meter to plating, which runs about 10 lbs per sq foot., it is rather obvious that the motion of the plating is not going to be the same with the meter in place as it would otherwise have been. The British recommend that the mass of the meter should not exceed three percent of the mass of the local structure to which it is attached. This means that the weight of the meter will have a negligible effect if placed on a ship's keel or on heavy machinery. It may have some effect if placed on stiffeners.

The Taylor Model Basin is now testing two modifications of the velocity meter. They have the same pickup coil and the same magnet, but weigh only half as much as the original. The difference is in the base and columns. At the present time records taken with the modified instruments have not been analyzed. It is hoped that, in decreasing the weight, low natural frequencies have not been introduced.

It is apparent that this type of velocity meter is not the ideal answer to the problem of measuring shock motions. It would be desirable to decrease its weight, extend the travel and minimize the friction. Until these improvements can be made, the British-type meter remains an essential instrument.

## MB VELOCITY METERS

by

Karl Unholtz, MB Manufacturing Company

A basic type of velocity pickup, consisting of a flexibly mounted coil on a rigid arm pivoted in a bearing, is described. Factors that control damping are discussed. Application to shock measurements is considered, including effects of the connected load on the damping of the unit.

A pickup as used in shock measurements is a relatively small instrument which usually generates a change in output voltage as an indication of a change in its condition of relative rest or motion. Depending upon the design characteristics of the pickup, the output voltage may be proportional to the displacement, velocity, or acceleration of the motion. In general, one of these three types is better adapted than the others to obtain the measurements required, and the specific design of the particular type depends largely upon the character of the motion to be measured, i.e., shock or steady state vibratory motion or combinations of both.

The general principles of operation of the velocity type of pickups recently constructed for NRL by the MB Company have been based on a lightweight coil of wire flexibly supported in an air gap of a magnetic circuit. Relative motion between the mounted coil and magnet structure causes a voltage to be generated which is proportional to the instantaneous value of velocity of relative motion. If the coil were ideally mounted, i.e., supported by a spring of zero restoring force, and if the damping forces could be eliminated, then the flexibly mounted coil could remain fixed in space while the magnet structure was forced to move, thus making the relative motion between the coil and magnet and the absolute motion of the magnet structure one and the same quantity.

Unfortunately, practical considerations preclude using supporting springs having zero restoring force, and zero damping forces are not obtainable with the rugged moving coil support construction required.

A well-designed pickup can be described mathematically, using a single displacement coordinate (single-degree-of-freedom system). The classical theory for this case shows the variation in phase as well as magnitude between the relative and absolute motions referred to above as a function of both the spring constant and the damping forces associated with the coil suspension.

Additional characteristics which can be enumerated as ideal for incorporation in a pickup are as follows:

1. Minimum size. The advantages of small size, allowing placement in any portion of a structure, are obvious.
2. Zero weight. Any added weight to a structure under test distorts the response, particularly in the lighter weight structures.

3. High voltage sensitivity. Direct recording, without electrical magnification, eliminates unknown characteristics of associated equipment as well as the inconvenience of their use.
4. Low impedance. Eliminates stray electrostatic and electromagnetic disturbances which can completely mask the true signal. Low impedance coupled with high voltage means power available for external circuits without amplification.
5. Precise electrical reproduction of instantaneous value of velocity of motion. Unit could be applied to any measurement, steady state or shock.
6. Large amplitude capacity. The moving coil should not contact stops during the time measurements are desired.
7. Freedom from directional errors. The pickup should indicate the instantaneous velocity at one point in the unit along one direction independently of components of motion in any other direction.

The construction details of the MB pivoted, rigid-arm type of pickup, having a seismically mounted coil, are shown in Figure 1. The coil is supported by a rigid arm and cross shaft mounted in ball bearings. The spiral springs give the supporting force to overcome gravity in the vertical direction, determine the natural frequency of the coil system, and make the electrical connections to the coil. In addition, one end of the spring is adjustable to allow for positioning the moving coil in either the center or at one end of the available coil travel.

Most of the weight of the moving coil assembly is in the coil itself, the rest of the arm and shaft being made of aluminum tubing. The coil suspension may be electromagnetically damped, if desired. The magnetic circuit is also indicated in Figure 1. The flux in the air gap is perpendicular to the direction of coil wire and coil motion. The flux density variation from the mean, from one end of coil travel to the other, can be held in production to plus or minus one and one-half percent.

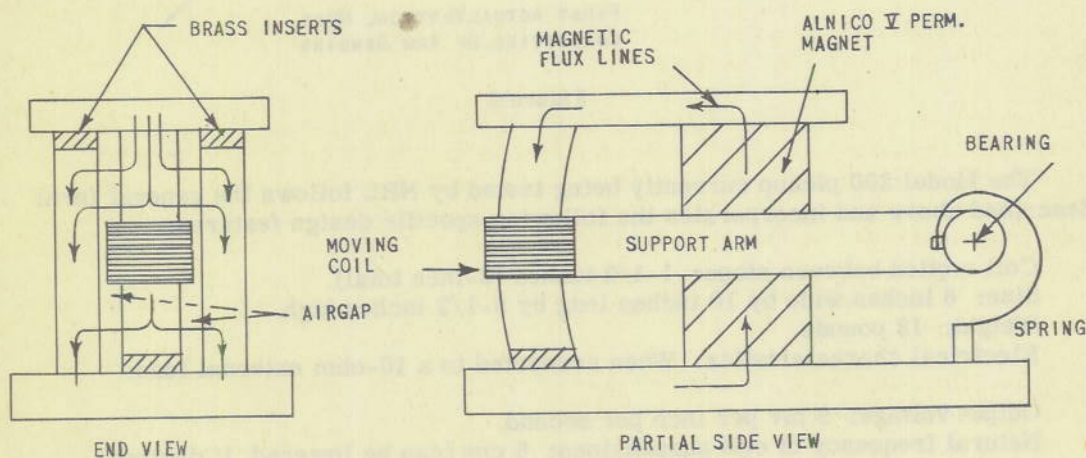


Figure 1

An advantage of the construction shown in Figure 1 is that resonant vibration modes of the rigid arm assembly have small effect on the output voltage. For example, the lowest resonant mode in the arm is lateral, bending in a direction parallel to the magnetic flux and, therefore, produces little or no electrical disturbance. Due to the ball bearing construction and the low moment of inertia of the cross shaft, the first resonant mode in the vertical direction, corresponding to the lateral mode described, is nonexistent. The first resonant mode of consequence in the vertical direction is shown in Figure 2. Due to the fact that the nodal point in this mode is near the coil center, the electrical disturbance associated with this type of motion of the arm is small. Due to the mass distribution and stiffness of the arm, this mode and the higher modes are of relatively high frequencies.

In regard to the direction characteristics of the construction (Figure 1) it may be said that the unit indicates velocity of motion at a point near the center of gravity of the coil itself, if the coil is heavy by comparison to the supporting arm. The unit is relatively free from rotational errors but is not theoretically free from effects of linear motions in the vertical plane and directed at other than 90 degrees to the axis of the arm. These effects can be visualized by remembering that tension or compression forces only are imposed on the coil supporting arm by the bearings because of their inherently low friction. If a shock motion is imposed on the arm assembly along the direction of the arrow shown in Figure 2, compression forces in the arm result. As the bearing end of the arm moves upward, the coil is forced downward by the resulting compression force, thereby giving a voltage output slightly in excess of the correct value based on the actual vertical component of the shock displacement. Quantitative measurements made by NRL, however, have shown these effects to be small.

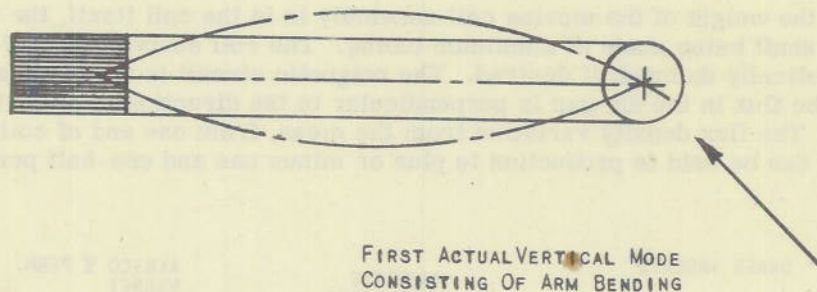


Figure 2

The Model 200 pickup currently being tested by NRL follows the general form described above and incorporates the following specific design features:

Coil motion between stops: 1-1/2 inches (3-inch total).

Size: 6 inches wide by 10 inches long by 5-1/2 inches high.

Weight: 18 pounds.

Electrical characteristics: When connected to a 10-ohm external load:

Output voltage: 9 mv per inch per second.

Natural frequency of coil suspensions: 6 cps (can be lowered, if desired).

Damping: 3 percent of critical (including bearings and effect of external resistance).

In connection with the electromagnetic damping, it can be shown that:

$$c/c_c \text{ is proportional to } \frac{(\text{air gap flux density})}{\omega_n M r}$$

where  $c$  = proportionality constant relating damping force to velocity.

$c_c$  = value of "c" for critical damping.

$\omega_n$  = natural frequency of coil suspension.

$M$  = effective mass of moving coil.

$r$  = resistance of damping circuit. This may be either coil from resistance, the coil and external resistance, or both.

In general, the coil form in the type of pickup here described is made of a continuous metallic loop which determines the value "r" above and allows  $c/c_c$  to be adjusted to the desired value. For example, for a uniform voltage sensitivity, independent of frequency variation under steady state vibratory excitation, a value of  $c/c_c = 0.65$  is desirable. By including all of the damping in the coil from electrical characteristics and feeding the output of the pickup into a high impedance load, consistent calibrations can be maintained. For measurements of shock, however, no damping is usually required. This means that the coil form must be nonconducting or, if metallic, open-circuited. If a low impedance load were to be connected directly to the pickup output, its contribution to the damping of the moving coil must be considered.

Tests are presently being conducted at NRL on a pickup incorporating important variations of the basic design features of the above-described unit, but to date the data are inconclusive.

## MULTIFREQUENCY REED GAGES, TYPES 1 AND 2

by

B. J. W. Grier, TMB

The performance characteristics of the multifrequency reed gages, as well as their theory and applications, are presented.

### INTRODUCTION

The multifrequency reed gages were designed and constructed for the measurement of shock motions on shipboard. The gages consist of a set of reeds of different natural frequencies rigidly attached as cantilevers in a heavy frame, which in turn is fastened rigidly to the structure subjected to the shock motion. The reeds have masses at their extremities to which styli are fastened. These styli scribe a record of the motion of the reed on waxed paper. For measurements, the gage is installed so that the longitudinal axis of the reeds is normal to the direction of the applied motion. One of the outstanding features of the gages is that their operation requires no power supply. They may be installed in many places where other available instruments would prove impracticable.

### DESCRIPTION

The Type 1 gage, Figures 1 and 2, has a stationary record platen across which waxed paper is stretched and secured by rollers at each end. The maximum deflections of the reeds occurring during the shock motion are recorded.

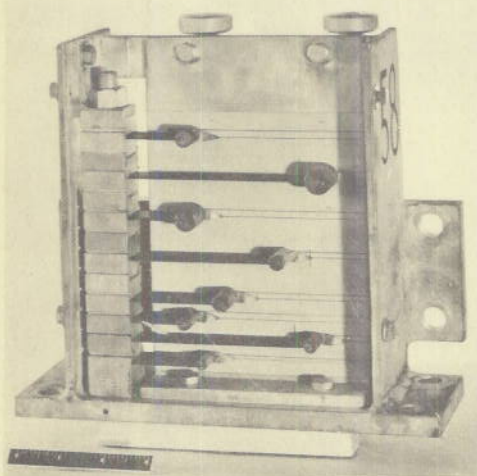


Fig. 1 - Multifrequency reed gage  
Type 1 (front view)

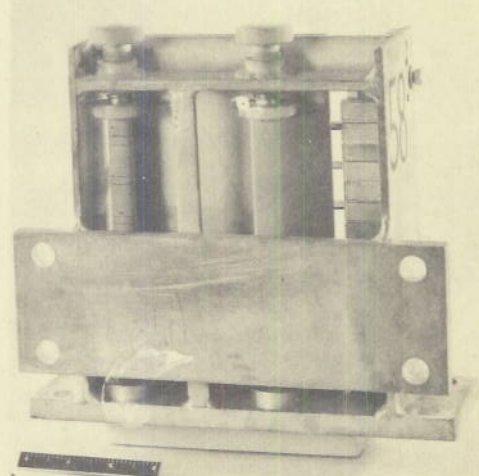


Fig. 2 - Multifrequency reed gage  
Type 1 (back view)

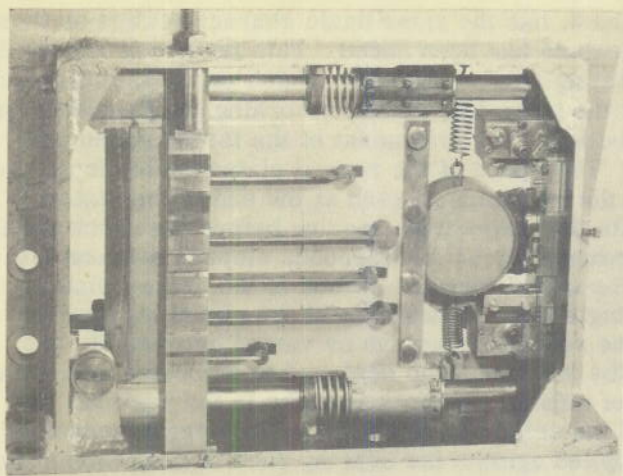


Fig. 3- Multifrequency reed gage Type 2 (front view) showing bumper assembly and inertia weight; record platen at left side prior to motion.

Fig. 4 - Multifrequency reed gage Type 2 (front view) showing record platen at right side after motion.

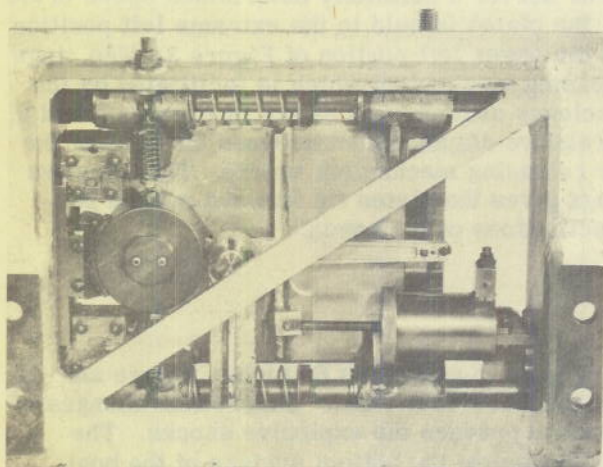
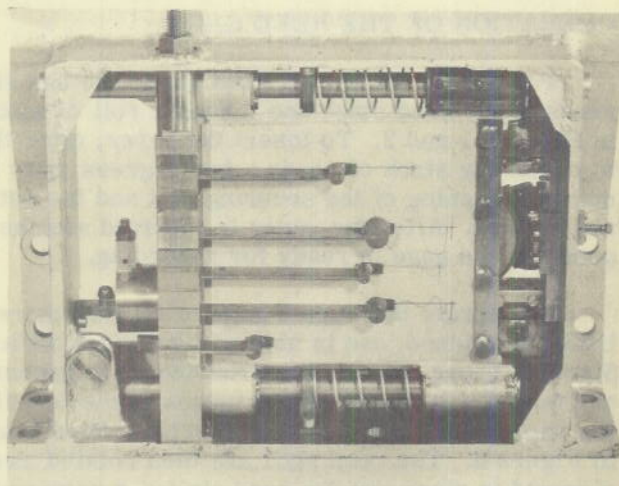


Fig. 5 - Multifrequency reed gage Type 2 (rear view) showing inertia weight and release arm assembly, air dash-pot, and bumper assembly.

The Type 2 gage, Figures 3, 4 and 5, has the same basic characteristics as those of the Type 1 gage; however, a new feature has been added. This feature is an inertia actuated releasing mechanism and sliding target assembly, which provides a means of obtaining a number of oscillations of the individual reeds, recording the frequency of motion as well as the maximum deflection. The movement of the target platen provides a record 1.86 inches in length. The speed of the record platen varies from approximately 30 inches per second at the beginning to zero at the end of the platen stroke. A restraining device prevents the platen from moving before the shock motion commences. A time interval of approximately 14 milliseconds elapses between the application of the shock motion and the initial forward motion of the record platen, consequently the reed gives only a single line deflection until the platen moves forward. The actual time of travel of the record platen can be varied between 50 and 500 milliseconds. The structure of the instrument is designed to withstand an acceleration of 200 g. The gage is capable of measuring shock motions in the vertical and horizontal directions. The weight of the seismic mass used in the trigger assembly is 2.8 pounds and the spring force applied against the bearing race of the restraining device is 25 pounds.

#### OPERATION OF THE REED GAGE

To operate the Type 1 reed gage, one has simply to install the appropriate set of reeds in the reed stack and thread a roll of waxed paper onto the two rollers as shown in Figures 1 and 2. To insert the paper, after the reeds are installed, it is necessary to rotate the stack of reeds a few degrees away from the plane of the paper. This requires loosening of the securing bolt and the set of reeds must then be lifted slightly and rotated. After the paper is inserted and the stack of reeds returned to the original position, the gage is ready for recording.

To operate the Type 2 gage is slightly more involved; however, with relatively little experience, one is able without much difficulty, to set the gage for recording. Since no paper rollers are provided, it is necessary to cut the waxed paper to size and fasten it to the platen by means of the two clamps. Before inserting the recording paper, the record platen must be shifted into the extreme left position as shown in Figure 3. The reed styli are then rotated away from the paper. The paper is clamped in place and the reed styli are returned to their recording position. The air dashpot, calibrated beforehand, should be set for a definitely determined speed of the record platen. In the present position, the platen is held in the extreme left position by a locking screw assembly, shown at the lower left portion of Figure 3. The gage is then in a position for setting the releasing mechanism which is facilitated by the use of an alignment pin on the inertia release assembly as shown in Figures 4 and 5. When the shock motion is applied, the relative displacement between the base of the gage and the seismic weight causes the releasing mechanism to trip. The effective force of the horizontally mounted springs gives the platen its forward motion permitting the recording of a number of oscillations of the reeds.

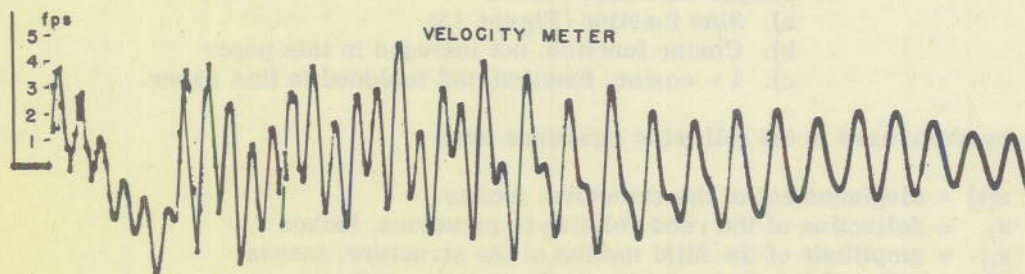
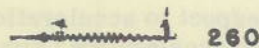
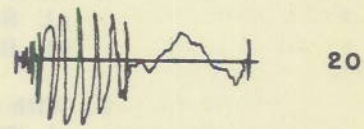
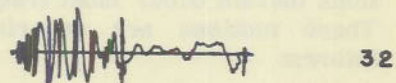
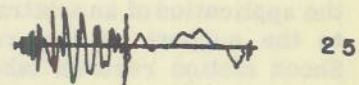
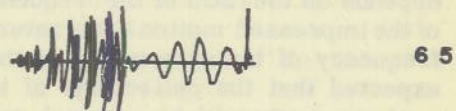
#### TEST METHOD

In the performance tests on the reed gage, a steel boat 66 inches square and 12 inches deep, was floated in the water of the TMB Test Pond. Underwater charges of tetryl weighing 22 grams each, were used to produce the explosive shocks. The charges were placed at a depth of 66 inches below the bottom surface of the boat, directly under the center of the diaphragm. The British type velocity meter was

mounted directly on the diaphragm, and the Type 2 reed gage was mounted directly on top of the velocity meter. The reed gage and velocity meter records shown in Figures 6 and 7 were taken simultaneously. These records have been analyzed for some of the dominant frequencies and were found to differ by not more than 25 percent in the estimated velocity. A more detailed analysis is underway.

Fig. 6 - Type 2 reed gage record, showing deflections of reeds under test conditions; charge of tetryl 22 grams, depth of charge 5-1/2 feet.

# REED GAGE cps

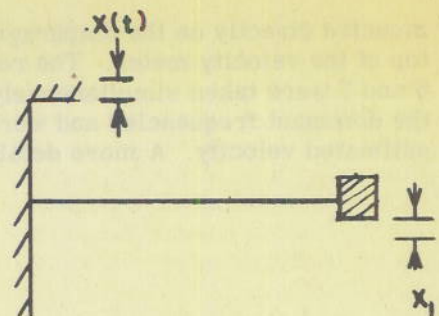


100 CPS SIGNAL

Fig. 7 - British type velocity meter record, showing curve under test conditions; charge of tetryl 22 grams, depth of charge 5-1/2 feet.

# THEORY OF THE MULTI-FREQUENCY REED GAGE

The response of a reed to various motions impressed on the gage has been calculated on the assumption that the reed behaves as a system of one degree of freedom, Figure 8. Under steady state conditions the amplitude of a system of one degree of freedom depends on the ratio of the frequency of the impressed motion to the natural frequency of the system. It is then expected that the deflections of the various reeds will be unequal upon the application of an arbitrary motion to the support of the reed gage. Shock motion records taken aboard ship reveal that certain elementary types of motion or close approximations thereof occur most frequently. These motions are described as follows:



$$\ddot{x}_1 + \frac{2p}{c} \dot{x}_1 + p^2 x_1 = -\ddot{x}(t)$$

$$\text{where } p = \sqrt{\frac{k}{m}} \text{ and } \frac{2p}{c} = \frac{c}{m}$$

Fig. 8 - One degree of freedom system, showing sketch and accompanying differential equation of motion.

1. With respect to velocity
  - a). Step velocity (Figure 9)
  - b). Rectangular pulse of velocity (Figure 10)
2. With respect to acceleration
  - a). Step acceleration (Figure 11)
  - b). Rectangular pulse of acceleration (Figure 12)
3. Simple harmonic motion
  - a). Sine function (Figure 13)
  - b). Cosine function, not included in this paper
  - c). 1 - cosine, function, not included in this paper.

The symbols used in the following equations are:

- $x(t)$  = displacement of the structure, inches
- $x_1$  = deflection of the reed relative to structure, inches
- $x_0$  = amplitude of the SHM motion of the structure, inches
- $V_i$  = value of the step velocity, inches per second
- $a$  = acceleration of the structure, inches per second<sup>2</sup>
- $f$  = frequency, CPS
- $\omega$  = natural circular frequency of the structure, radians per second
- $p$  = natural circular frequency of the undamped reed, radians per second
- $\omega_n$  = natural circular frequency of damped reed =  $p \sqrt{1 - (c/c_c)^2}$
- $T$  = period of the reed =  $1/f$ , seconds
- $c/c_c$  = damping ratio, percent of critical damping
- $\dot{x}$  = velocity of the structure, inches per second
- $\ddot{x}$  = acceleration of the structure, inches per second

## METHOD OF ANALYZING THE RECORDS

Calculations were made of the response of the reed as a single degree of freedom system, to various impressed motions. Friction or damping between the reed stylus and the waxed paper also affect the reed response. Measurements have shown generally that little damping is present among the reeds. It may range up to 12 percent of critical. From the equations obtained, it can be observed that if the damping factor  $c/c_c$  is assumed to be zero, the value of the motion seldom differs by more than 15 percent from that of the motion of the damped reed.

The time duration of a specific motion considered is a very important factor as far as shock intensity is concerned. It is believed that the time duration predominates over the shape of the shock motion. It is also believed that a large displacement and high velocity must occur in a relatively short interval of time in order to have shock effect. A high acceleration must be sustained over a relatively long interval of time in order to have shock effect.

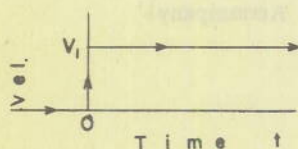
The methods used in analyzing the records of the reed gages will be described in the following examples.

## 1. Step velocity (Figure 9)

A step function is one which attains a finite value instantaneously and maintains this value indefinitely or at least for the time considered.

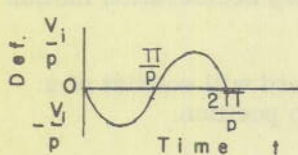
For the Type 1 gage, the deflection  $x_1 = x_m$  of each reed is multiplied by its natural circular frequency  $p = 2\pi f$ , where  $f$  is the natural frequency in CPS. If the products  $x_m p$  obtained for all the reeds are approximately equal, a step velocity motion is indicated, and its value is equal to that of the product.

For Type 2 gage, the analysis is basically the same as that of Type 1, since the first cycle of the reed's motion (which is the maximum deflection) usually occurs before the record platen commences its forward motion. The record will consist of a damped sinusoidal curve.



$$\ddot{x}_1 + \frac{2p_c}{c} \dot{x}_1 + p^2 x_1 = 0$$

(a)



$$x_1 = -\frac{V_1}{p} \sin pt \quad (\text{undamped reed})$$

$$x_1 = -\frac{V_1}{\omega_n} e^{-\frac{c}{c_c} pt} \sin \omega_n t \quad (\text{damped reed})$$

(b)

Fig. 9 - (a) Step velocity vs applied shock motion with accompanying equation. (b) Time-deflection curve of a reed's deflection with accompanying equation.

## 2. Rectangular pulse of velocity (Figure 10).

A rectangular pulse function is one which attains a finite value instantaneously from a zero position and maintains this value for a time shorter than the time under consideration and then reduces instantaneously to its zero position.

For the Type 1 gage, the product  $x_{mp}$  is obtained as in 1. If this value for the higher frequency reeds does not differ by more than a factor of two and the products for the lower frequency reeds decrease with frequency, a rectangular pulse of velocity is indicated. It is also possible to estimate the duration of this velocity pulse from the data.

For the Type 2 gage the method is the same. The curve will not be purely sinusoidal, but the maximum deflection will occur at the beginning.

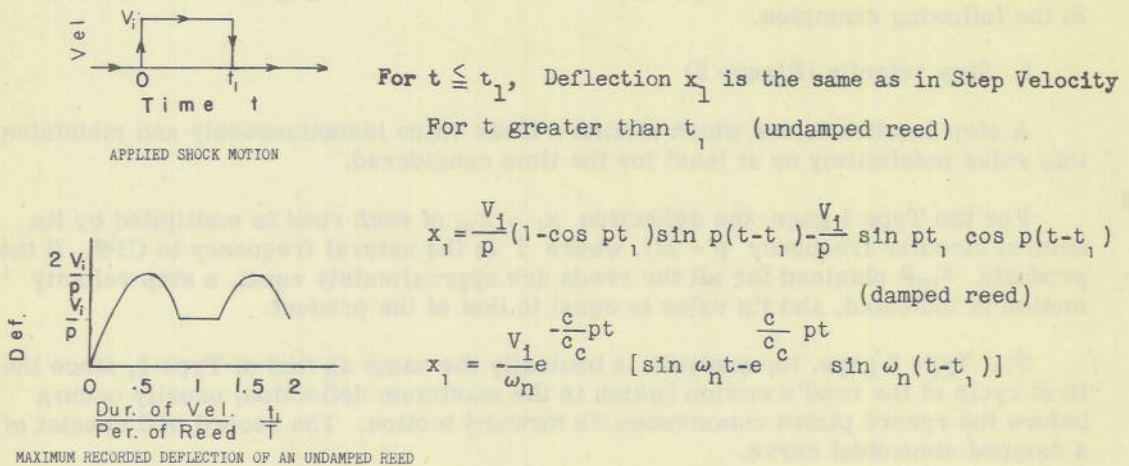
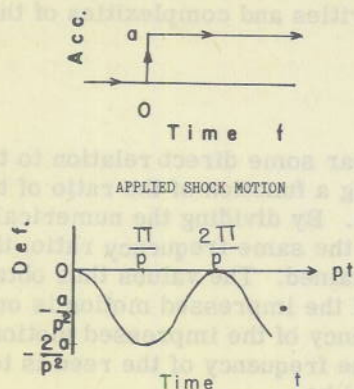


Fig. 10 - Rectangular pulse of velocity vs applied shock motion; time-deflection curve of reed's relative deflection. Accompanying equations.

## 3. Step acceleration (Figure 11).

For the Type 1 gage, the method of analyzing a record for step acceleration is similar to that for the step velocity. The product will be, in this case,  $x_{mp}^2/2$ . If these values are approximately equal for all the reeds, a step acceleration motion is indicated.

For the Type 2 gage, the method is the same. The record will consist of a damped sinusoidal curve on one side only of the reed's zero position.



$$x_1 = -\frac{a}{p^2} (1 - \cos pt) \quad (\text{undamped reed})$$

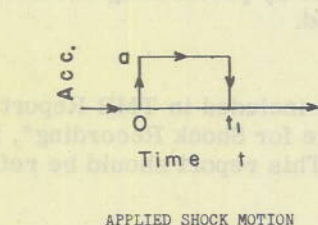
$$x_1 = -\frac{a}{p^2} \left[ 1 - e^{-\frac{c}{c_c} pt} \left( \frac{p}{\omega_n} \frac{c}{c_c} \sin \omega_n t + \cos \omega_n t \right) \right]$$

Fig. 11 - Step acceleration vs applied shock motion; time-deflection curve of a reed's relative deflection. Accompanying equations.

REED'S MOTION UNDER A STEP ACCELERATION

#### 4. Rectangular pulse of acceleration (Figure 12).

For the Type 1 gage, the expression  $a = x_m p^2/2$  (where  $a$  is the amplitude of acceleration,  $x_m$  is the maximum deflection and  $p$  is the natural circular frequency of the reed). If the calculated values of acceleration are equal for the higher frequency reeds and lower for the lower frequency reeds, decreasing with reed frequency, a rectangular pulse of acceleration is indicated.



Duration of Acc.,  $t_1$   
Period of Reed,  $T$

MAXIMUM RECORDED DEFECTION OF AN UNDAMPED REED

$$t \leq t_1$$

$$t > t_1$$

$$x_1 = -\frac{a}{p^2} (1 - \cos pt) \quad (\text{undamped reed})$$

$$x_1 = -\frac{2a}{p^2} \sin p \left( t - \frac{t_1}{2} \right) \sin \frac{p t_1}{2} \quad (\text{undamped reed})$$

$$x_1 = -\frac{a}{p^2} \left[ 1 - e^{-\frac{c}{c_c} pt} \left( \frac{p}{\omega_n} \frac{c}{c_c} \sin \omega_n t + \cos \omega_n t \right) \right]$$

$$x_1 = +\frac{a}{p^2} e^{-\frac{c}{c_c} pt} \left[ \left( \frac{c}{c_c} \frac{p}{\omega_n} (1 - \cos \omega_n t_1) \right) \right. \\ \left. - e^{-\frac{c}{c_c} pt_1} \sin \omega_n t_1 \sin \omega_n t + \left( \frac{c}{c_c} \frac{p}{\omega_n} \sin \omega_n t_1 \right) \right. \\ \left. - e^{-\frac{c}{c_c} pt_1} \cos \omega_n t_1 + 1 \right) \cos \omega_n t \right] \quad (\text{damped reed})$$

Fig. 12 - Rectangular pulse of acceleration vs applied shock motion; time-deflection curve of a reed's relative deflection. Accompanying equations.

For the Type 2 gage the method is the same. It should be pointed out here that one observes the number of oscillations for irregularities and complexities of the motions impressed.

#### 5. Simple harmonic motion (Figure 13).

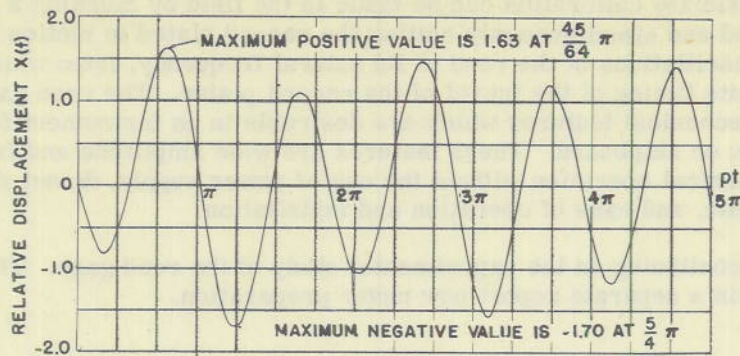
For the Type 1 gage, the reed deflections will bear some direct relation to the amplitude of the impressed motion, that relation being a function of the ratio of the frequency of the reed to that of the impressed motion. By dividing the numerical value of a reed's deflection by the unit deflection for the same frequency ratio, the value of the amplitude of the impressed motion is obtained. The values thus obtained for all the reeds will be equal, or approximately so, if the impressed motion is one of the types of the simple harmonic motion. The frequency of the impressed motion can be approximated due to the fact that the closer the frequency of the reed is to that of the impressed motion, the larger its deflection will be.

For the Type 2 gage, the records should be compared with the theoretical deflection curves of the various types of simple harmonic motion. If the recorded deflection for all the reeds have the same general shape as those of one of the theoretical curves, the type of the impressed harmonic motion is indicated as well as the value of the  $p/\omega$  ratio and of the  $c/c_c$  ratio for each curve. As the value of the reed circular frequency  $p$  is known for each reed producing the record and as the value of the ratio  $p/\omega$  is known from the theoretical curves, the value  $\omega$  can be calculated. All these values should be equal if the impressed motion is actually represented by the theoretical curves. This is not always the case; irregularities do occur in the shape of the impressed motion. Some calculations have been made on records at the TMB Underwater Test Pond. Figure 6 shows one such record. It has been found that the values obtained were within 20 percent of the values on the theoretical curves. The value of the amplitude of the impressed motion  $x_0$  is obtained by performing the same type of calculation as that for the Type 1 Reed gage record.

The shock motions enumerated in this paper are included in TMB Report 613 "A Theoretical Study of the Multifrequency Reed Gage for Shock Recording", by Davidson and Adams, which is soon to be published. This report should be referred to for a more detailed analysis.

#### CONCLUSION

It is advisable to attempt analyzing a record in the order presented in this paper, i.e., from the simple step function to the more complex SHM Motion. With the Type 2 gage, however, the shape of the record will give some indication of the type of impressed motion. It is important to select reeds with proper frequencies in order to obtain satisfactory records. Reeds whose frequencies are many times higher than the frequency of the impressed motion will deflect very slightly with consequent difficulty in measuring the correct deflection. If the record obtained on the Type 2 gage is to be interpreted on a time base a displacement-time calibration of the record platen must be first obtained. Where a single shock motion occurs the Type 1 gage gives a satisfactory record for analysis, however, when more complicated types of motion are present a more satisfactory analysis can be obtained from the Type 2 reed gage record.



(a) Sinusoidal shock motion, theoretical relative deflection of reed for impressed shock motion defined by the equation  $x = \sin \omega t$  for  $c/c_c = 0.1$  and for  $p/\omega = 0.5$

$$x_1 = \frac{x_0}{\frac{p^2}{\omega^2 - 1}} \sin \omega t + \frac{x_0}{\frac{p}{\omega}} \sin pt$$

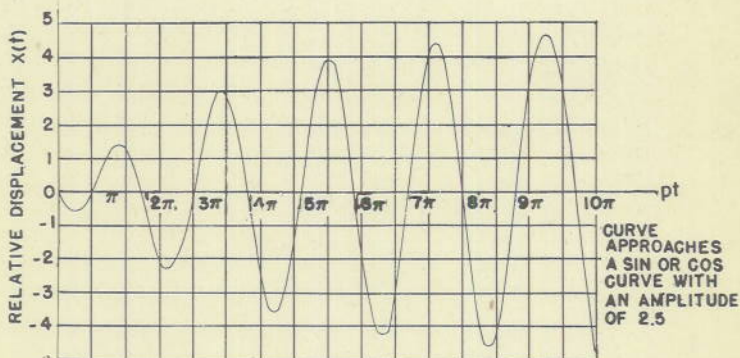
(undamped reed)

$$\frac{x_1}{x_0} = -e^{-\frac{c}{c_c} pt} \left[ \sin \omega_n t \left( \frac{\omega}{\omega_n} - \frac{p}{\omega_n} \frac{c}{c_c} \frac{\sin \phi}{R} + \frac{\omega}{\omega_n} \frac{\cos \phi}{R} \right) - \frac{\cos \omega_n t}{R} \sin \phi \right] + \frac{\sin(\omega t - \phi)}{R}$$

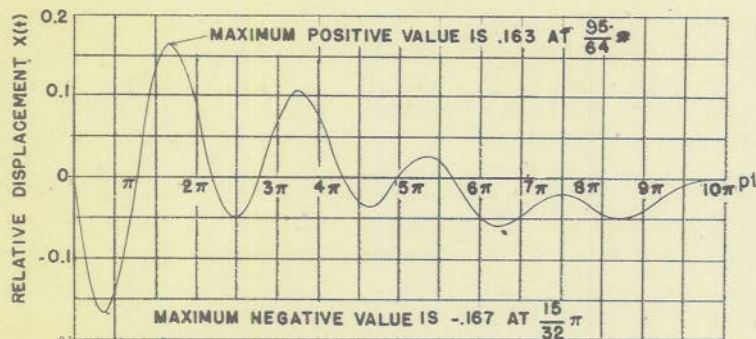
For  $x = x_0 \sin \omega t$  (damped reed)

$$\text{Where } R = \sqrt{\left[ \left( \frac{p}{\omega} \right)^2 - 1 \right]^2 + \left[ \frac{2c}{c_c \omega} \right]^2} \text{ and}$$

$$\phi = \tan^{-1} \frac{2p/\omega \cdot c/c_c}{(p/\omega)^2 - 1}$$



(b) Sinusoidal shock motion, theoretical relative deflection of reed for impressed shock motion defined by the equation  $x = \sin \omega t$  for  $c/c_c = 0.1$  and for  $p/\omega = 1.0$



(c) Sinusoidal shock motion, theoretical relative deflection of reed for impressed shock motion defined by the equation  $x = \sin \omega t$  for  $c/c_c = 0.1$  and for  $p/\omega = 1.0$

Fig. 13 (Sinusoidal shock motion, reed's relative deflection for shock motion defined by equation  $x_0 = \sin \omega t$ ).

An approximate calibration can be made in the field by plucking a relatively high frequency reed and simultaneously setting the record platen in motion. This records a number of oscillations of the reed at its natural frequency, from which one obtains the approximate timing of the travel of the record platen. The reed gages incorporate design and mechanical features which are desirable in an instrument for recording shock motions on shipboard. These features are wide amplitude and frequency ranges, mechanical operation without the use of power supply, direct recording, nonphotographic, and ease of operation and installation.

Work is continuing on the experimental study of the reed gage. The results will be presented in a separate report now under preparation.

## DISCUSSION

J. P. Walsh, NRL: The author should be congratulated for an interesting discussion of the reed gage and a method of analyzing the data from the instrument.

In the writer's opinion, there is another way of treating the reed gage data which makes the results more useful. It is felt that these more useful results are obscured by the method of analysis presented by the author.

The method of analysis presented in this paper is essentially that of seeking motions, for which the response of the reed gage is known (velocity changes, step velocities, etc.) which most closely approximate the shock motion to which the reed gage has been subjected. It is probably safe to say that "practical" shipboard shock motions are not "simple" shocks, and attempting to find the "simple" shock which most closely approximates the actual shock is not of great value, since a designer cannot use even these simple approximate shocks directly.

In our work on design at NRL, we have found (as others have before us-- M. A. Biot, for example) that one of the most useful concepts is that the response of single-degree-of-freedom systems to the shock motion is a measure of the probability of damage resulting from the shock motion under study. When given a velocity-time curve of a shock motion or even the "simple shock which most closely approximates the actual shock," we get the response of single-degree-of-freedom systems to the shock motion by using the torsion pendulum or the graphical methods. The real value of the reed gage, we think, is that it gives directly the response of single-degree-of-freedom systems to the actual shock motion.

The method of calculation is as follows: With each first mode natural frequency of the reeds we associate the static deflection,  $\delta_{st}$ , of a single-degree-of-freedom system having the same frequency. Let the deflection relative to the base of the reed under shock be  $X$ , then  $X/\delta_{st} = N$  where  $N$  is the number of "g" which, if applied statically, would produce the same deflection as was produced under shock. We then plot  $N$  as a function of the natural frequency of the reed. This plot is the so-called frequency-response spectrum of the shock motion, which is of use in design problems.

The method we use does not exclude the method described by the author, for, if one wishes, he can plot on the frequency-response spectrum the values of response to those "simple" shocks mentioned by the author and find the simple shock which most closely approximates the actual shock.

J. T. Muller, Bell Telephone Laboratories: About the instrument itself, the automatic release, which is caused by the shock, delays the starting of the recording paper by approximately 15 milliseconds. It is easy to see from displacement records that nearly half of the transient displacement of the shock is thus lost. It is of no value, therefore, as far as determining the actual displacement-time curve of the shock. Yet the author presented a rather incomplete mathematical analysis of the relative motion of the reed mass to show how this motion is related to the shock

displacement. A more complete analysis is given in great detail by W. P. Welch, of the Westinghouse Research Laboratories, in a paper, "A Proposed New Shock Measuring Instrument," presented at the Chicago Convention of 1947 of the S.E.S.A. and to be found in the proceedings of the Society.

What the author actually showed in the high-speed motion pictures was the steady state condition after the shock had passed. This has been treated mathematically in a paper, "Transients in Mechanical Systems," which I wrote for the 1947 Winter Convention of the S.E.S.A. There is a precise relationship between the reed amplitude and the duration of the shock, such that the amplitude may vary from zero to 1.7 times the pulse amplitude, depending upon the frequency ratio of the pulse frequency and the natural frequency of the reed. For instance, if this ratio is approximately .75, then the amplitude will be a maximum; while if it is 2, the amplitude may be zero. These values will change somewhat as the shape of the displacement pulse is varied, but it illustrates the point. As far as I see it, about the only thing multifrequency reed gages are good for is that they might indicate some critical values of natural frequency.

## PEAK-READING ACCELEROMETERS

by

R. E. Blake, NRL

Mechanical gages designed to indicate the maximum acceleration of a shock motion are discussed. These are: (a) the putty gage, (b) the Buchanan gage, a weighted-end cantilever of the same type as the putty gage, (c) the indenter gage, and (d) the ball-crusher gage. The principle of operation of each gage is outlined. Sources of error are discussed, particularly those due to the duration of the acceleration. The necessary conditions for minimizing these errors are pointed out, and the work yet to be done is indicated.

The four gages discussed here were all designed to measure the maximum acceleration of a shock motion. With one exception, they are entirely mechanical and self-contained, requiring no power or auxiliary recording apparatus. They record automatically the maximum acceleration to which they are subjected between the time they are assembled and the time the record is removed from them. Occasionally this has been a disadvantage, but more often it is an advantage, since it is not necessary to synchronize their recording action with a shock.

The relative ease in using peak accelerometers has sometimes resulted in their use in work for which more complex gages and more complete records were actually required. In most tests, peak acceleration alone is not an adequate definition of a shock motion--though it is a useful part of that definition. Some conception of the time history of the motion is usually essential. When this deficiency of these gages has been kept in mind, peak accelerometers have been used to obtain useful data. Some examples of such use are the following:

1. The gages are used to check on the operation of shock machines. When a combination of machine settings produces the same peak acceleration as it did on a previous test, it is reasonable to suppose that the shock motions were identical.
2. The gages are used alone for simple shocks whose time history can be estimated, such as in the water-entry shock of torpedoes, launching of rockets, etc.
3. The gages are used adjacent to velocity meters and reed gages, since those two meters do not define peak acceleration very accurately.

**Putty Gage.** - The putty gage shown in Figure 1 represents an important type of peak accelerometer. The cross-section shows two of the eight masses used in this model. Each mass is pressed against the top of the gage by a precompressed coil spring. If the gage were subjected to a gradually increasing acceleration in the direction indicated, an acceleration for each mass could be found which is just sufficient to separate the mass from the top of the gage. This separation is recorded by indentation of the plasteline insert by the conical-tipped shaft. This "critical acceleration",  $A_c$ , can be calculated from the formula  $A_c = F_s / M_w - \frac{1}{2} M_s$ , where  $F_s$  is the pre-compression force of the spring,  $M_w$  is the mass of the weight, and  $M_s$  is the mass of the spring.

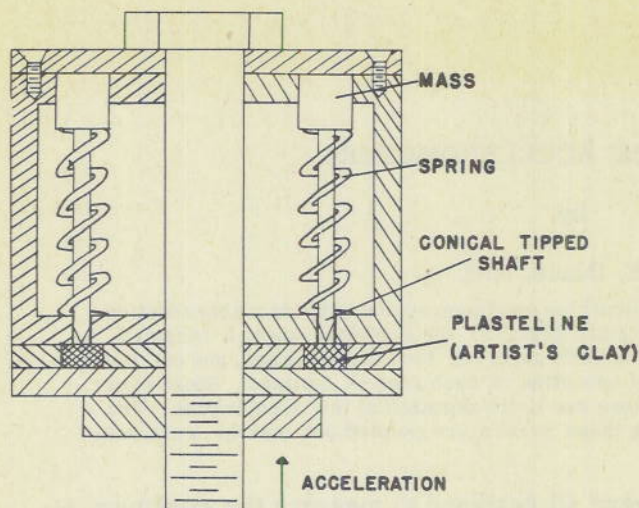


Fig. 1 - "Putty gage" peak acceleration indicator

stiffness,  $k$ , and length,  $l$ . Each end of the spring went through the shock acceleration  $a(t)$ , and the variation of  $F_s$  with time, was studied. The theory of longitudinal stress waves in uniform bars was used. According to this theory, the longitudinal stress in a rod due to a step velocity change,  $V$ , of the tip of the rod is  $\sqrt{klm} V$ . A stress wave moves at a uniform speed along the spring, without change in shape or intensity. The time,  $T$ , required for a stress wave to move from one end of the rod to the other is  $\sqrt{m/k}$ . A wave is reflected from the end of the rod without change of sign and moves back along the rod.

Using the effect of a velocity change as the indicial admittance of the spring, it is seen that the force at one end of the spring due to any velocity  $v(t_1)$  alone at time,  $t_1$ , is  $\sqrt{klm} v(t_1)$ . Other forces to be superposed upon this force are the precompression force  $F_s$ , the stress which originated at the other end at time  $t = t_1 - T$ , the stress wave which originated at the same end at time  $t = t_1 - 2T$ , and so on until the earliest stress force initiated by the shock has been taken into account. The formula for the spring force on the mass is:

$$\begin{aligned}
 F(t) &= F_s - \sqrt{klm} v(t) - [\sqrt{klm} v(t-T) + \sqrt{klm} v(t-2T) + \sqrt{klm} v(t-3T) \\
 &\quad + \sqrt{klm} v(t-4T) + \dots + \sqrt{klm} v(t-nT)] \\
 &= F_s - \sqrt{klm} \sum_{k=0}^n v(t-kT)
 \end{aligned}$$

where  $t - nT \leq$  the time at which the shock motion started.

The graphical analysis indicated by this equation has been carried out for the case where the shock acceleration,  $a(t)$ , is a half-sine pulse, with the result shown in Figure 2. The ratio of indicated to actual peak acceleration is seen to vary and to depend upon the duration of the shock pulse. In an NRL putty gage, mass ratios,  $M_s/M$ , up to one were encountered, and the surge time,  $T$ , was on the order of one millisecond.

After being subjected to shock, a gage is read by noting which inserts were indented. The peak acceleration of the shock is said to be greater than the critical accelerations of those masses which caused indentations and less than those which did not. By a suitable choice of critical accelerations, the peak acceleration can be bracketed between rather narrow limits.

One source of error of the above method is the variation of spring pressure during the shock because of the surges of stress in the spring. This was studied by Dr. Paul Symonds and the writer as outlined in the following. For a theoretical analysis, the spring was assumed to be a rod of uniform mass per unit length,  $m$ ,

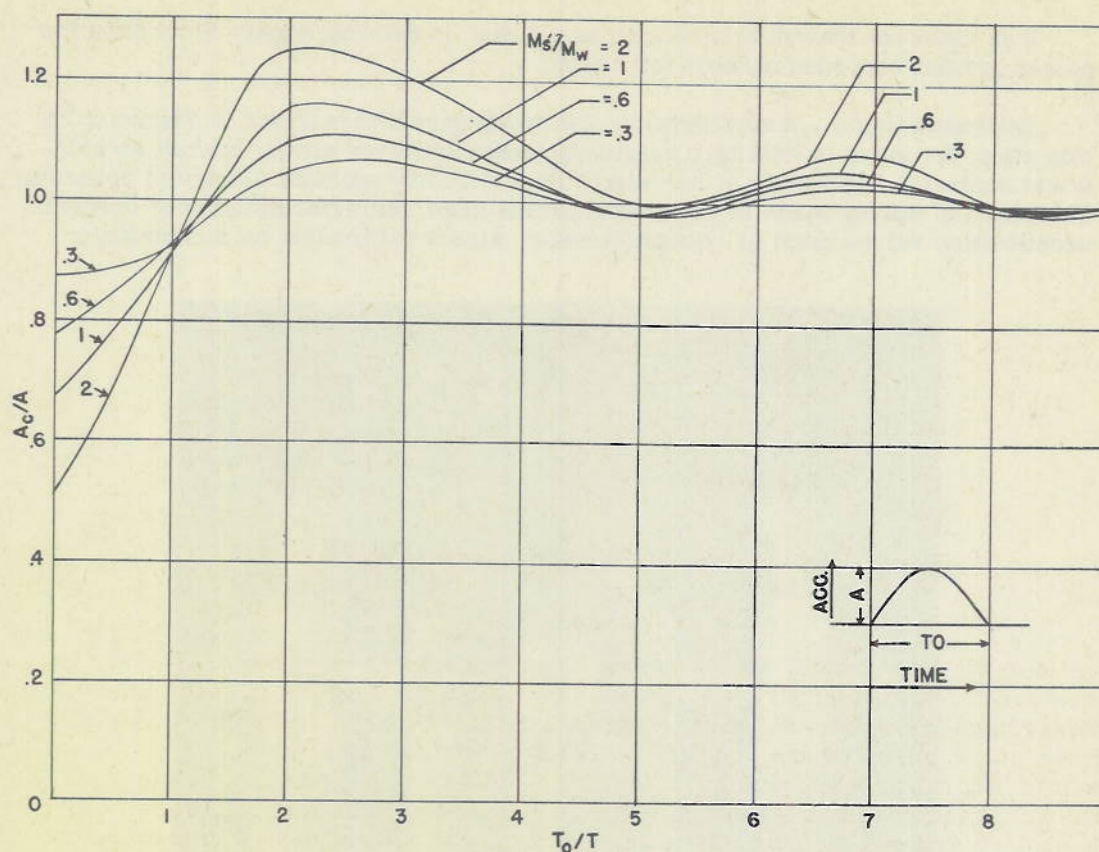


Fig. 2 - Response of putty gage to half sine wave acceleration

It is evident that lighter springs will result in smaller errors due to spring surge, but it can be shown that this means stiffer springs. In order to exert a precompression force within a certain range of accuracy, the precompressed height of the stiffer spring must be set with greater accuracy. Thus, greater accuracy requires greater precision in manufacture.

The preceding analysis was based upon the assumption that there was no clearance between the conical tip and the plasteline surface. A clearance of .002 inch existed in the gage used by NRL. This means that  $\int a(t)dt - \frac{1}{2} A_c t^2$  when  $a > A_c$  must exceed .002 inch in order that the clay will be indented. An approximate idea of the influence of this clearance upon the gage readings can be obtained by considering  $a(t)$  to be a "square" pulse of duration,  $t_0$ . Assuming the spring force to be constant, the gage will read too low by the amounts tabulated below.

Square Pulse  
Duration-Millisec.

Gage Error  
in G

|    |      |
|----|------|
| 1. | 10   |
| .5 | 41   |
| .3 | 115  |
| .1 | 1000 |

It is apparent that clearance acts as a filter, in that extremely short duration pulses are not recorded at their full value.

**Buchanan Gage.** - A modification of the putty gage idea is shown in Figure 3. In this gage, the mass is held by a relatively light cantilever spring against an electrical contact. Separation of the weight is detected by suitable electrical apparatus. The effect of spring mass is minimized in this gage, but determination of critical acceleration values must be accomplished by direct calibration on a centrifuge.

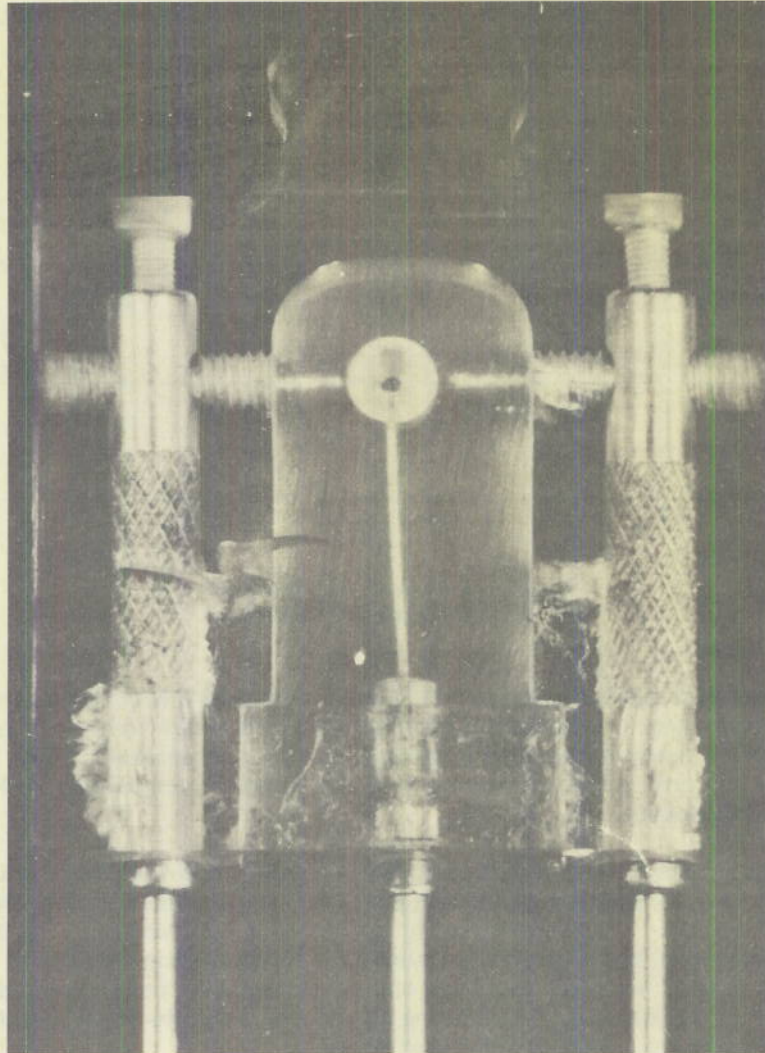


Fig. 3 - Modification of putty gage

The elimination of the clearance effect has permitted true peak-acceleration indications of pulses shorter than .05 millisecond. This brings up the question of whether or not the peak value of an extremely short acceleration pulse is of interest.

In many problems, the very short pulses are of no interest, and some sort of filtering of the higher frequencies is desirable. A filtering effect is provided in the Buchanan gage by a second electrical contact whose circuit is closed after the weight has traversed a known clearance. This, of course, duplicates the putty gage behavior.

A putty-gage-type instrument with electrical contacts was used by a group at California Institute of Technology.\* By means of a network of resistors and a battery it was possible to obtain an increment of voltage across the output terminals as each contact was opened by the shock. This voltage was proportional to the highest critical acceleration of the contacts which were open and was applied to the input of a string oscillograph. A rough time record of the acceleration was thus obtained. The allowable travel of a weight after opening of a contact was restricted to less than .01 inch to minimize the delay in closing of a switch as the acceleration fell off. Accelerations along only one sense of one direction were indicated, just as in the putty and Buchanan gages.

**Indenter Gage.** - The indenter gage represents an important class of peak accelerometers which are based upon the indication of peak force by the plastic deformation of a soft metal element. An indenter gage element utilizes a soft aluminum pellet, into the flat surface of which a hard-steel conical point is forced. The diameter of the resulting crater is related to the peak force exerted. As shown in Figure 4, one or more of these elements is used to support a weight of known mass in the interior of the gage. Each indenter element records the maximum force in one direction exerted on the weight during the shock motion. Other elements can be used to record other components of force. The peak force measured is equal to the peak acceleration of the weight times its mass, so the maximum acceleration of the weight can be easily calculated. This is the "indicated acceleration" of the gage.

Calibration of the indenters can be effected in two different ways. Using a static loading machine, one may find a peak load versus crater diameter curve. It is possible that such a curve will result in readings which are too low, because the metal is subjected to a high strain rate under shock. The California Institute of Technology group used a dynamic calibration by dropping weights on an indenter element to obtain a curve of energy absorbed versus depth of penetration. The slope of the energy-indentation curve was measured to plot a force-indentation curve. Since crater diameter and depth of penetration are uniquely related, a dynamic calibration curve was obtained. No comparison has been made of dynamic and static calibration curves, as far as the writer knows.

The indicated acceleration of the gage was stated previously to be the peak acceleration of the internal weight. Since the process of indentation involves relative motion between the weight and the gage base, it is not surprising that the indicated acceleration is not always the maximum acceleration of the gage base. In order to illustrate the error due to this effect, Dr. Paul Symond, at NRL, made a theoretical study of the gage when it is subjected to a half-sine pulse of acceleration. The peak load,  $F$ , was assumed to be related to the depth of penetration,  $d$ , by the formula  $F = Rd^2$ .  $R$  was determined from a static calibration of indenter elements used in some NRL gages. The curves applicable to this NRL gage are shown in Figure 5. The ratio of the indicated number of  $g(N_i)$  to the actual peak number of  $g(N_p)$  varies from 0 to 2.5, depending upon the pulse duration. This gage is seen, then, to have a filtering effect upon the high-frequency pulses and to magnify pulses of a certain frequency range.

\* Impact Decelerometers, OSRD Report 3430

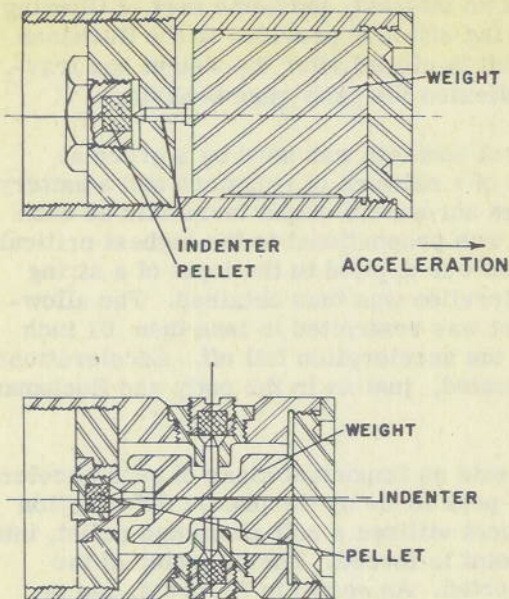


Fig. 4 - One directional and five directional impact gages indenter type

A serious limitation of the indenter gage is due to its behavior under a vibratory type of shock. The first acceleration pulse of a shock motion will move the internal weight toward one side of the gage, indenting one aluminum sample. This creates a clearance between the weight and the opposite indenter element. If the acceleration reverses direction, the weight may move through this clearance before striking the opposite indenter. Successive reversals of the acceleration can cause the weight to rattle back and forth, driving the indentations deeper with each impact. Such a situation has not been analyzed further than this or studied experimentally, but use of this gage should be confined to one-directional accelerations until it has been proven satisfactory for vibratory shock.

comments regarding indenter gages apply equally to the crusher gage, except for small changes to Figure 5.

**Ball Crusher Gages.** - In Figure 6 is shown the ball crusher type of gage which is very similar to the indenter gage. Force is recorded by the amount a soft copper ball is flattened by the internal weight. All the

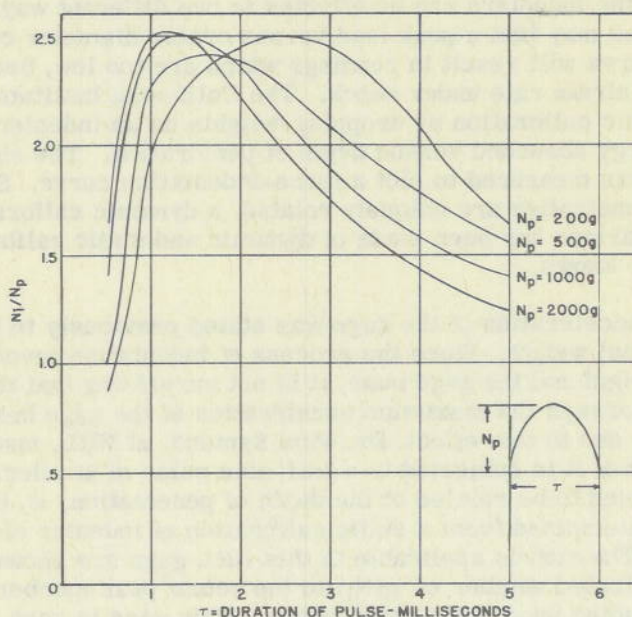
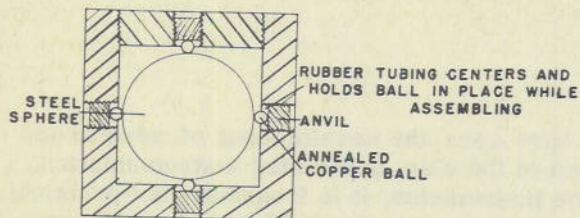


Fig. 5 - Indenter gage ratio of indicated to peak acceleration of sine pulse



NOL SIX DIRECTIONAL BALL ACCELEROMETER

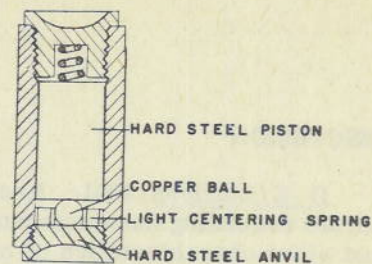
UNIDIRECTIONAL BALL ACCELEROMETER  
(About 2X Full Scale)

Figure 6

The load-compression curve for copper balls has been found to be a straight line for compression up to about 30 percent of the ball diameter. This simplifies the analysis of the effect of simple shocks. The differential equation of motion is the same as that for a simple oscillator, as long as the ball is continuously compressed. The curve for a half-sine pulse differs only in a minor way from that for the indenter gage. The filtering of high-frequency pulses and the too-large reading for a certain frequency range occur in the crusher gage as well as in the indenter gage.

## DISCUSSION

D. E. Sullivan, NOL: Recent years have seen the development of several accurate recording accelerometers. Because of the size, associated instrumentation, and other characteristics of many of these instruments, it is frequently not practical to employ them to obtain data; this is particularly true in the field testing of aircraft launched mines. The instrument preferred in many cases is the crusher accelerometer, also called the copper-ball accelerometer. Closely related to it are the copper-ball pressure gages and the crusher gages which are widely used in field tests.

The principal advantages of this accelerometer are its compact size, its ruggedness, and the ease and precision with which it can be read. The principal disadvantages are that it is a peak reading instrument, recording only the greatest acceleration encountered, and that it has a tendency to give spurious readings (to magnify under certain conditions).

The Shock Test and Design Section of NOL has recently been faced with several problems in the laboratory where it was not practical to use recording accelerometers and where information was desired. Techniques using the copper-ball accelerometer were employed which, it is hoped, may be of interest and use to readers of the Bulletin.

Considering the accelerometer as an elastic system (which is justifiable, since the load-deflection curve is linear) it is possible to calculate its natural frequency (sometimes referred to as pseudo-natural frequency). NOL at present has eight accelerometers designed for recording accelerations ranging from 10 to 30,000 g. The natural frequencies of these range from 150 to 1650 cps.

Having such a range of natural frequencies available has enabled us to turn the tendency of the accelerometer to "over-shoot," or "magnify" to our advantage in cases where the duration of the acceleration is known to be long. As is brought out in Figure 7, the amount that an accelerometer will magnify an applied acceleration of given "buildup" time,  $T_0$ , depends upon the natural period,  $T$ , of the accelerometer. In other words, two accelerometers of different natural frequencies will record differently the same applied acceleration.

This principle has been applied recently to determine the time of "buildup",  $T_0$ , of the acceleration exerted by the five-inch, high-g air gun. In this case, the magnitude of the acceleration and the natural periods,  $T$ , of several accelerometers are known. The magnification factors are readily obtainable from the recorded accelerations, and thus the time,  $T_0$ , is obtained with little difficulty.

The use of several accelerometers is also desirable when neither the magnitude of the applied acceleration nor the time of buildup,  $T$ , are known. The method in this case is to assume various values of  $T_0$  and to solve for the magnitude of the acceleration for each  $T_0$ , using the curve shown in Figure 7. In this manner, a curve of values of acceleration vs time,  $T_0$ , is obtained for each accelerometer. The point where these curves intersect is the true value of applied acceleration and of time of buildup,  $T_0$ . It is possible for the curves to intersect at several values of  $T_0$  (all integrally related). However, judicious choice of accelerometers will eliminate all but one of these.

The multiple frequency reed gage of DTMB can also be used in this fashion. The relatively small size of the accelerometers and the precision of measurements in the high frequency range make the use of the accelerometers preferable, however, in some instances.

The copper-ball accelerometer can also be used to advantage as a velocity meter. A large-mass, low-frequency accelerometer is used when the applied acceleration is of high magnitude and very short duration. The deformation of the copper-ball in such cases will be a measure of the energy imparted, rather than acceleration, and thus can readily be converted to "velocity change".

In conclusion, let us say that the copper-ball accelerometer, when properly used, can be a very useful instrument. While it is still true that "if you know everything about the applied shock, you can build a copper-ball accelerometer to measure the shock," it is also true that with judicious use, the accelerometer--in its various modifications--can give a wealth of information where little existed before.

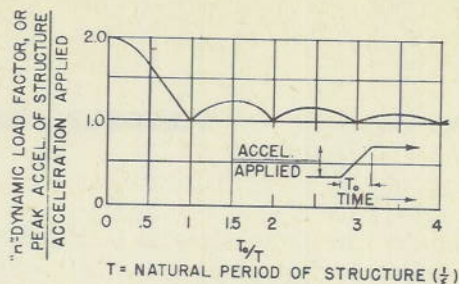


Fig. 7 - Effect of rapidity of application of sustained acceleration on response of an elastic structure

## PIEZOELECTRIC ACCELEROMETERS

by

J. L. Jones, NOL

The basic principles of the mass-spring accelerometer system are discussed in relation to the phenomenon of piezoelectricity which is used to convert acceleration effects to electrical transients. The advantages and limitations of crystal pickups are discussed, and the influence of these characteristics on the instrumentation techniques employed is shown. A few applications of corrected and uncorrected crystal accelerometer circuits at NOL are discussed in relation to the type of acceleration pattern being studied.

The Naval Ordnance Laboratory has been engaged in the field of acceleration measurements for many years. Before the last war, most of the effort had been directed toward obtaining measurements indicative of peak acceleration only. Various types of peak-reading accelerometers were used for this purpose, but, in general, no attempt was made to calibrate the instruments dynamically or to study the effect of different wave shapes upon the results. Continuous recording accelerometers were seriously considered in 1941 when it became evident that a more complete knowledge of impact shocks was necessary in order to design shock mounts for delicate mine mechanisms and to devise laboratory simulation tests to obviate tedious and expensive field testing. The first continuous recording accelerometer developed by the Laboratory was a mechanical type consisting of a spring-driven drum holding a celluloid tape, and a small mass carrying a stylus and suspended on a cantilever spring. Relative displacement between the mass and the tape was recorded on the celluloid; under suitable conditions this record could be interpreted as an acceleration-time curve of impact. Development of a crystal accelerometer began in the fall of 1942, after a survey of available recording accelerometers showed that none were suitable for measuring impact shocks associated with aircraft-launched mines.

Crystal accelerometer pickups depend on the well-known piezoelectric properties of some crystals; viz., that mechanical force applied along certain axes of such crystals produces an electrical charge proportional to the applied force. Many types of crystal pickups are available. Basically, the operation of all types is similar, however, so a description of a pickup designed at NOL will suffice. Figure 1 shows the construction of the NOL pickup. The spring-loaded piston provides the inertia force when the pickup housing is accelerated. The pickup is sensitive only to the component of acceleration parallel to the axis of the piston, and in the qualitative description which follows it is understood that only this component of acceleration is under consideration. Since the spring constant of the crystal stack is much greater than that of the plate spring, the force between the plate spring and piston remains essentially constant when the pickup unit is accelerated, the inertia force of the piston being

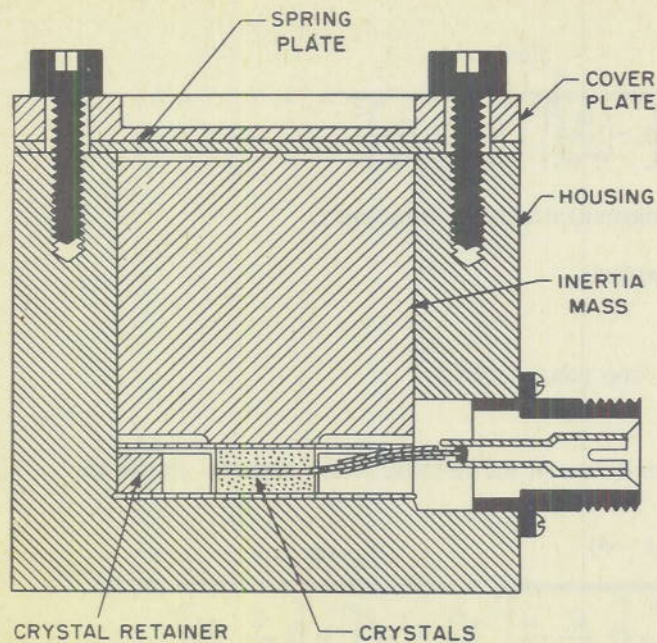


Fig. 1 - NOL crystal accelerometer pickup

balanced by a change in force between the crystal stack and piston. The preceding statement is true, regardless of the sense of the applied acceleration, as long as the inertia force does not exceed the spring load of the plate spring. If the applied acceleration is so large that the inertia force exceeds the spring load and is in such a sense that the force between crystal stack and piston is decreased, the piston will leave the crystal stack, and the unit will no longer function as an accelerometer until the acceleration is reduced. The maximum acceleration in the opposite sense which the units will record is limited only by the strength of the crystal stack. The choice of piezoelectric crystal depends upon the application. Low-g accelerometers generally employ Rochelle salt because of its large piezoelectric constant, which results in good sensitivity. In such applications, the poor mechanical

properties of Rochelle salt are of no consequence, but variation of the piezoelectric constant with temperature is sometimes troublesome. X-cut quartz crystals are used in high-g pickups where sensitivity is no problem. Quartz has excellent mechanical properties, and the piezoelectric constant of x-cut quartz is essentially independent of temperature. ADP (PN) crystals are sometimes useful in accelerometer pickups because they have piezoelectric, mechanical, and thermal properties intermediate between Rochelle salt and quartz.

A quantitative discussion of pickup operation will now be presented. As previously mentioned, the pickup is so designed that the spring constant of the plate spring is negligible compared to that of the crystal stack. Consequently, the mechanical system may be simplified to that shown in Figure 2. Consider a sinusoidal acceleration,  $A(t) = A_0 e^{j\omega t}$ , paralleled to the piston axis applied to the pickup housing. Assuming viscous damping, the differential equation of motion is:

$$m \frac{d^2 y}{dt^2} + C \frac{dy}{dt} + ky = ma(t) = mA_0 e^{j\omega t},$$

where

$m$  is the mass of the piston,

$k$  is the spring constant of the crystal stack,

$C$  is the coefficient of viscous damping,

and

$y$  is the RELATIVE displacement between inertia mass and housing.

The steady-state solution is:

$$y = \frac{A_0}{\omega_n^2} \cdot \frac{e^{j(\omega t - \phi)}}{\sqrt{\left[1 - \left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left[2 \frac{C}{C_c} \frac{\omega}{\omega_n}\right]^2}},$$

where  $\omega_n = \sqrt{\frac{k}{m}}$ , the undamped natural circular frequency,

$C_c = 2m\omega_n$ , critical damping,

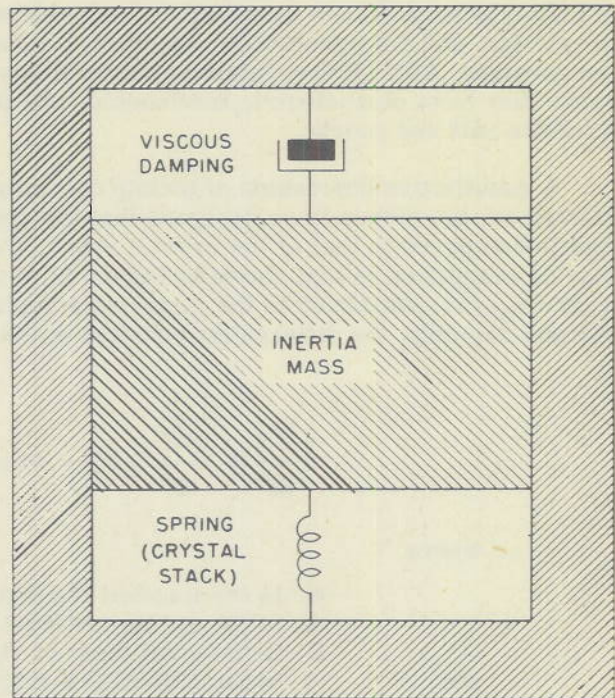
and  $\phi = \arctan \frac{2 \frac{C}{C_c} \frac{\omega}{\omega_n}}{1 - \left(\frac{\omega}{\omega_n}\right)^2}$ , the phase shift.

We thus see that the change in force between the crystal stack and the piston is:

$$F = ky = \frac{kA_0}{\omega_n^2} \cdot \frac{e^{j(\omega t - \phi)}}{\sqrt{\left[1 - \left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left[2 \frac{C}{C_c} \frac{\omega}{\omega_n}\right]^2}} = \frac{e^{-j\phi}}{\sqrt{\left[1 - \left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left[2 \frac{C}{C_c} \frac{\omega}{\omega_n}\right]^2}} ma(t).$$

The term  $\frac{1}{\sqrt{\left[1 - \left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left[2 \frac{C}{C_c} \frac{\omega}{\omega_n}\right]^2}}$  is sometimes called the magnification factor.

Fig. 2 - Mechanical system of crystal accelerometer



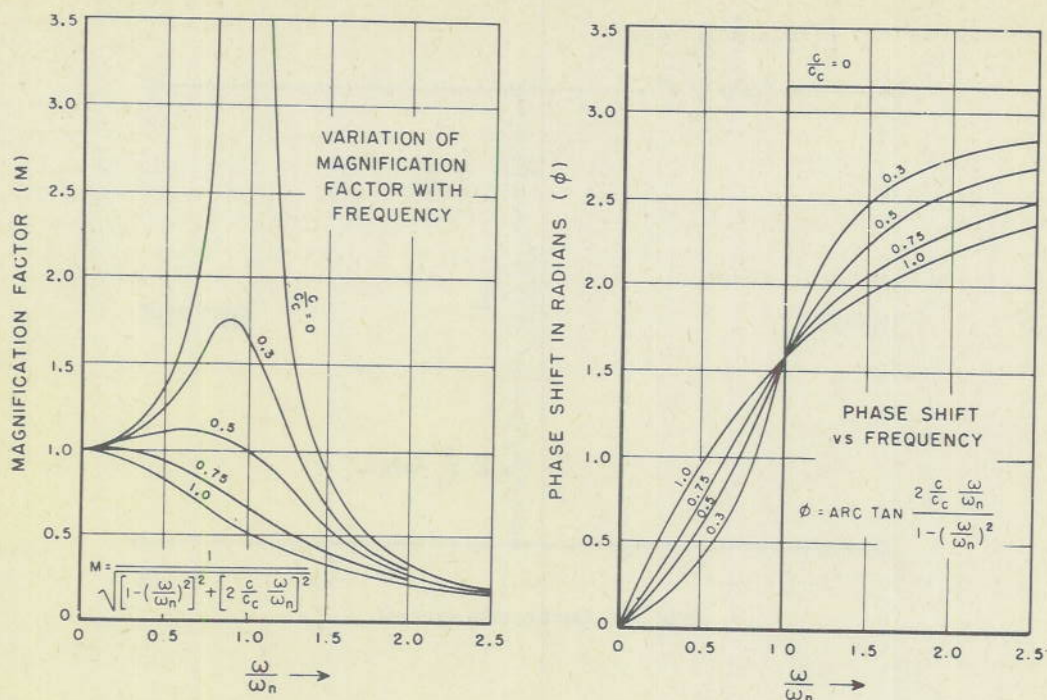


Fig. 3 - Frequency characteristics of viscously damped mass-spring system

Curves showing the variation of this factor and of the phase shift with frequency are shown in Figure 3. Since the charge produced by the crystals is proportional to the force,  $F$ , it is evident that these curves can be interpreted as the amplitude- and phase-response functions of the pickup. The requirements for distortionless transduction into an electrical signal are a flat amplitude response and a linear phase shift characteristic over the range of the frequency spectra of the applied acceleration. From the curves of Figure 3, two conclusions are immediately evident:

- The higher the natural frequency of the pickup the greater its range of applicability.
- approximately 75% of critical damping is optimum.

The natural frequency is limited in practice by sensitivity requirements. For instance, referring to Figure 1, the natural frequency of the pickup could be increased by reducing the size of the inertia mass or by diminishing the number of crystals, but both of these modifications would reduce the sensitivity of the unit. Realization of 75% of critical damping by mechanical means is quite impractical for a crystal pickup of the type shown in Figure 1. However, the desired end can be accomplished electrically by means of a corrective circuit. It can be shown that the overall response of a system consisting of a viscously damped pickup with damping ratio  $C/C_c$  ( $< .75$ ) and the corrective circuit shown in Figure 4, is identical to the response of a similar pickup having optimum damping (75% of critical).

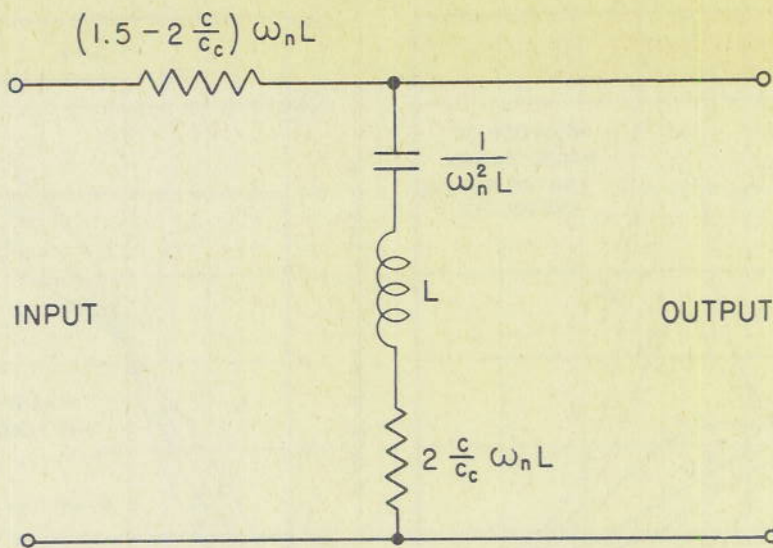


Fig. 4 - Corrective circuit

The preceding analysis has been concerned with the high-frequency response of crystal pickups. The low-frequency operation will now be discussed. The usual circuit into which a crystal pickup feeds is shown in Figure 5. The total shunt capacity,  $C$ , is the sum of the pickup capacity, shunt condenser capacity, and the capacity of the cable leading from the pickup to a preamplifier. The resistance,  $R$ , is the input resistance of this preamplifier. It is the leakage of charge through the resistor that results in poor low-frequency response. Consider, for instance a constant acceleration suddenly applied to the pickup so the crystals generate a charge  $Q$ . This charge instantaneously distributes itself so the voltage across the condenser is  $Q/C$ . Immediately, however, this charge starts to leak through the resistor so an exponential trace is recorded for a constant applied acceleration. As long as the time constant ( $R$ - $C$  product) is at least 20 times as long as the duration of the applied acceleration pulse, it is certain that distortion will be negligible. In cases where it is not possible to realize a sufficiently long time constant, a method of correcting the recorded trace is necessary. It can be shown that:

$$A(t_0) = A_r(t_0) + \frac{1}{CR} \int_0^t A_r(t) dt,$$

where  $A_r(t)$  is the recorded acceleration, and  $A(t)$  is the true acceleration.

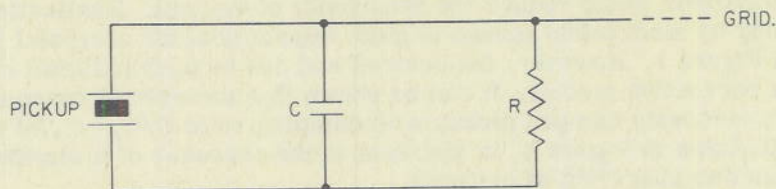


Fig. 5 - Input circuit

According to this formula, the correction to the recorded acceleration at any time  $t_0$  is found by dividing the area under the curve from  $t = 0$  to  $t = t_0$  by the time constant of the circuit; this correction is then added to the recorded acceleration to indicate the true value. In the derivation of this result, it is assumed that the point  $t = 0$  is so chosen that  $A_T(0) = A(0)$ .

It is evident that the larger the resistance and capacitance of the input circuit, the longer the time constant. The capacitance is limited by sensitivity requirements, but the highest possible value of resistance is desirable. This suggests the use of a cathode-follower preamplifier as an impedance matching device between pickup and recorder. A two-stage cathode follower is often convenient. A high- $\mu$  triode is suggested for the first stage to obtain a high input impedance, and a high transconductance triode is most desirable in the second stage in order to realize a low output impedance. Obtaining an input impedance as high as 500 megohms and an output impedance of less than 100 ohms is no problem. Having a low output impedance is necessary in applications where a long cable from the recorder loads the preamplifier. The corrective circuit mentioned previously can be inserted between the stages.

Figure 6 shows a schematic diagram of a recording accelerometer designed at the Naval Ordnance Laboratory primarily for the purpose of studying impacts associated with aircraft-launched mines (with parachute). The accelerometer, consisting of a recording unit, battery box, and delay device, is completely self-contained. Neither a preamplifier nor other amplifiers are used because of adverse microphonic signals produced by amplifying tubes when subjected to shock. A signal of about 75 volts is required to produce full-scale deflection of the cathode-ray beam. Since the peak acceleration encountered is usually less than 150 g's, a sensitivity of  $\frac{1}{2}$  volt per g is required. The desired sensitivity was attained using a crystal pickup having a natural frequency of 6000 cps by minimizing the shunt capacity to about 100  $\mu\text{mf}$ . A resistance of 1000 megohms was satisfactory for stabilizing the cathode-ray spot, so a time constant of about 100 ms was realized. This was, in general, satisfactory for the short-duration impacts encountered.

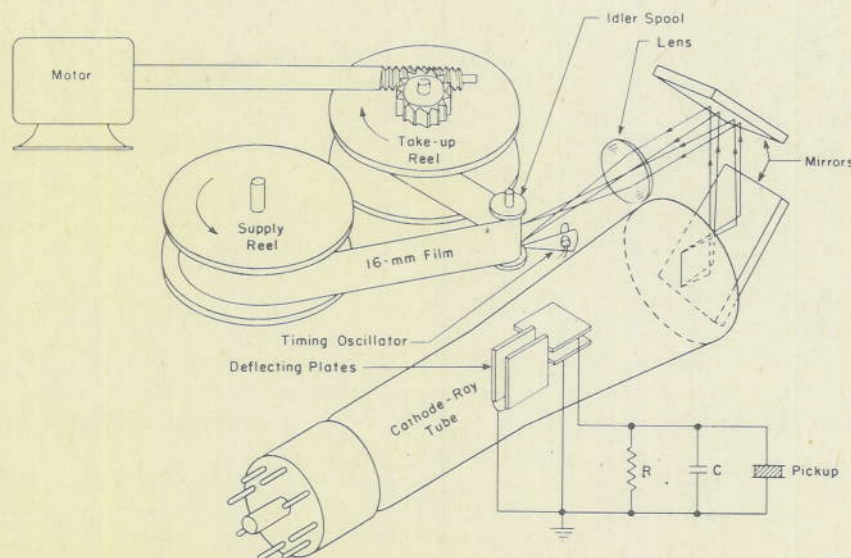


Fig. 6 - Schematic diagram of recording accelerometer

The application described in the preceding paragraph demonstrates one advantage of crystal pickups; viz., good sensitivity. A second advantage is the fact that the construction is such that a high natural frequency is attainable. In addition, it is pointed out that the same crystal pickup can be used over a wide range of acceleration by simply adjusting the input capacity to such a value that a signal of desired amplitude is obtained. For instance, the pickup shown in Figure 1 has been used over a range of acceleration from 20 g's to 3000 g's. For small accelerations, where a small input capacity is required, it may be difficult to realize a sufficiently long time constant so the low-frequency response is adequate. The poor low-frequency response is a fundamental limitation of crystal pickups. Perhaps a more serious limitation is the high impedance of the input circuit. This feature is characteristic of all crystal gages and makes the elimination of false signals difficult. All cables generate a signal when subjected to shock. If a low-impedance gage is used, these signals are effectively shorted out, but with high-impedance gages such as crystal accelerometers these signals are sustained. The only solution seems to be in minimizing the cable signals by special cable design. Much effort has been directed along this approach both in this country and in England, and considerable success has been experienced. At the present time, one of the most satisfactory shock-proof cables is manufactured by the Simplex Wire and Cable Co., Cambridge, Mass. However, the problem is still troublesome and is the reason for the choice of low-impedance strain-gage accelerometers for many applications.

## STRAIN GAGE AND CARRIER-TYPE ACCELEROMETERS

by

E. W. Kammer, NRL

A commercial accelerometer which utilizes a wire strain gage as its sensitive element is described. Its operational characteristics are explained. A model of the NRL strain gage accelerometer, which employs commercially available strain gages, is demonstrated. In addition, an accelerometer depending upon a change in reluctance of a magnetic path is discussed. Another depends upon the change in position of a movable magnetic core with respect to two transformer-like windings. In addition several methods of accelerometer calibration are described.

### INTRODUCTION

For the study of shock waves in structures, it is often desirable to measure the motion of various members subjected to the shock. When the displacement of the point in question is of the order of inches in magnitude, it is difficult to construct a low-frequency, seismic-type pickup of convenient size which can handle these amplitudes. Although the information desired may be confined to the initial portion of the motion, it is undesirable to have the movable element strike the excursion limits soon thereafter and obscure the remainder of the record. When an accelerometer is used, there are no relatively flexible parts of the pickup, and thus there are no problems that large relative motions might introduce.

The ranges of vibration frequencies and of accelerations anticipated in the intended use of the instrument at this Laboratory were up to 5000 cps and 5000 g, respectively. In this range, successful accelerometers have been made which depend upon the piezoelectric effect for their operation. The high impedance of the piezoelectric accelerometer introduces many difficulties, such as shielding from electrical noise interference, distortions if long lengths of cable are used, and excessive signal loss if much additional capacity is introduced. In the crystal-type accelerometer, when constant acceleration is applied to the unit, the charge developed initially gradually leaks off through a grid resistor at the input to an amplifier, or it dissipates through the insulation of connecting cables and fittings or around the edges and through the body of the crystal element itself. This means that a piezoelectric unit introduces serious errors when the output is integrated electronically over appreciable time intervals. In addition, tribo-electric effects produced by movement of the cable are more pronounced. This is particularly true under field test conditions which are not as favorable or controllable as those in the laboratory.

Accelerometers that depend upon the action of electrical resistance change in a conductor, due to strain during acceleration, are free of some of these disadvantages because of their low-impedance and power-output capabilities. A few models operating on this principle are now available commercially. In the present case the resistance, in the form of bonded-wire strain gages, was found to be a practical sensitive element for a wide-range accelerometer having sufficiently low impedance to reduce cable effects and electrical interference to a negligible degree. Furthermore, the output under constant acceleration is maintained indefinitely at a value proportional to the acceleration. Hence, this instrument can be used as a basis for an integrating

velocity meter even though the acceleration lasts for a considerable length of time.

Aside from the electrical characteristics already mentioned, certain other qualifications for the proposed instrument must be considered. In general usage, the instrument should be as light and small as possible. An instrument measuring accelerations must have all self-resonant frequencies of its own, well above the maximum significant observed frequency in order not to introduce amplitude distortion. In the present instance, the applications were such that a limit of 5000 cps could be placed on the important component contributions to the resultant accelerations. A low-pass filter with this cut-off frequency is used in the amplifier channel. This is the same filter suggested as a standard for shock test analysis described in NRL Report No. 0-2645 and references cited therein. A lowest self-resonant frequency of about twice that of the highest frequency to be observed should generally be designed into the pickup unit; however, the shape of the filter transmission curve makes it possible to relax this requirement. Finally, the instrument should discriminate as much as possible against accelerations in directions other than those along its axis and should have as great a sensitivity as is practicable along the direction of its axis.

A successful line of development leading to instruments depending upon a different signal source uses the carrier technique. The acceleration is able to change an inductance, capacitance, or resistance and thereby modulate this carrier, either in amplitude or frequency. The accelerometer is thus no longer required to provide any power, since the carrier is supplied from an external source. Furthermore, a low impedance is readily achieved. The signal is practically independent of cable effects, and shielding is no longer a serious problem.

The variety of instruments operating on the carrier principle is great and no attempt will be made to review the field exhaustively. Three successful models manufactured commercially are described briefly below. Accelerometers depending upon the change of a capacity with acceleration will not be considered here.

A typical example of an instrument which depends upon change in inductance under acceleration is the Glenn L. Martin Type B accelerometer. In this type, the reluctance of a magnetic path is changed by movement of a core, and the inductance of a coil associated with this magnetic path is thereby changed. This inductance forms an element of an alternating current bridge. (See Figure 1.) The change in the inductance causes unbalance in the bridge and hence modulates the applied carrier.

A second type in this same category is made by the Schaevitz Company. In this instrument, a core (common to three coaxial coils) changes its position during acceleration. The carrier is applied to one of the coils located centrally with respect to the other two. (See Figure 2.) Motion of the common core will induce a greater voltage in one of the remaining coils and less in the other. This, in turn, results in a modulated carrier when the secondary coils are connected so that the induced voltages oppose each other.

Still another instrument, which may or may not be used with an alternating current carrier, depends upon the change in resistance of a fine wire due to the relative movement of an inertia mass with respect to the frame of the accelerometer. Instruments of this type are made by the Statham Laboratories. Figure 3 illustrates the essential components.

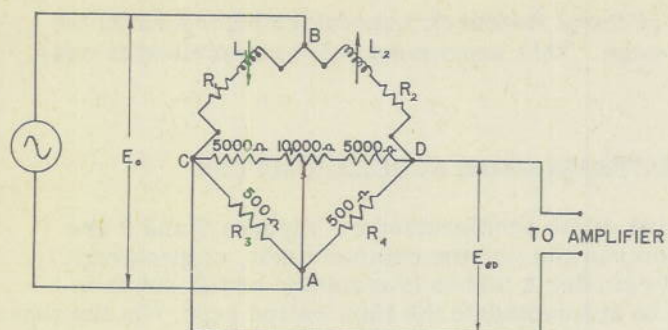


Fig. 1 - G.L.M. accelerometer bridge circuit

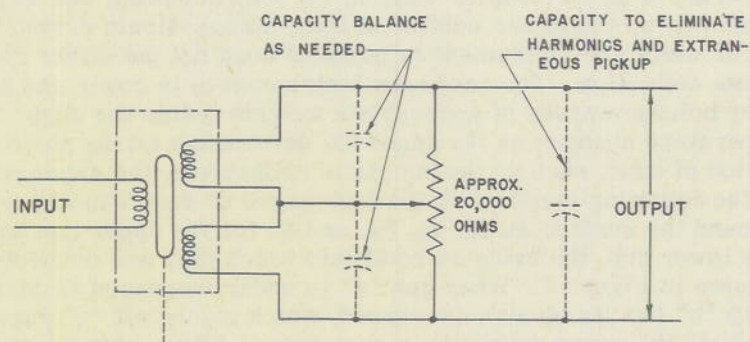


Fig. 2 - Q-method for reducing minimum balance below .001 volt

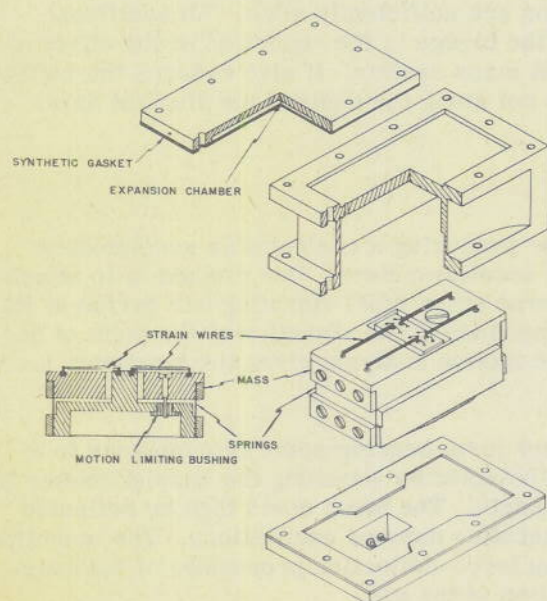


Fig. 3 - Statham accelerometer model AP

A development along this line at Naval Research Laboratory incorporates the standard SR-4 bonded-wire strain gage. This instrument will be described in considerable detail.

#### DESCRIPTION OF THE NRL WIRE STRAIN GAGE ACCELEROMETER

Figure 4 is an assembly drawing of the accelerometer. Figures 5 and 6 are photographs showing the various components and the complete unit, respectively. The accelerometer body is a dural cylinder 2 inches in diameter and 2 inches in length. This cylinder is bored out to accommodate the thin-walled cups. On the outside walls of these cups and equally spaced around the circumference are cemented four fine-wire resistance strain gages of approximately 1000 ohms each. These gages have an active length of  $5/16$  inch. The length of the cups is made in such a way that tightening the central bolt places the cup walls under tension. This tension is designed to produce a strain of about .0013 to .0015 in the wall material, corresponding to a point about half way up the linear portion of their stress-strain curve. The maximum acceleration for which the instrument is intended does not put either cup in compression at the maximum deflection. The range the instrument is to cover can be selected to some extent by bolting masses of appropriate weights inside the cups. To reduce the effect of temperature changes on the gages for deflections taking place over a considerable period of time, such as during static calibration, the gages on the cups are connected in the following manner: If the four gages on each cup are numbered consecutively around the surface as 1a, 2a, 3a, and 4a for the upper cup and 1b, 2b, 3b, and 4b for the lower cup, the bridge circuit into which they are connected would be labelled as shown in Figure 7. When cup "a" is under increased tension during acceleration, cup "b" has its tension decreased, which means all "a" gages increase in resistance and all "b" gages decrease in resistance. Their connections in the bridge circuit are such as to produce maximum unbalance in the bridge for this condition. However, a temperature increase tends to affect all gages equally and, if all the gages have the same temperature coefficient of resistance, will either increase the resistance of both sets of "a" and "b" gages or correspondingly decrease the resistance of both sets for a decrease in temperature. This type resistance change will produce practically no unbalance of the bridge if the gages are sufficiently alike. An additional feature of this arrangement of the gages in the bridge is the reduction of the effect of undesired modes of vibration of the cups and mass system. It also reduces the unit's sensitivity to accelerations in any direction not along the desired longitudinal axis.

#### CALIBRATION OF ACCELEROMETERS

There are several practical methods for producing a controllable acceleration for the purpose of comparing or calibrating accelerometers. One method is to attach the accelerometer to the center of a transverse (free-free) vibrating bar driven at its resonant frequency. If the frequency and amplitude of the vibration at the point of attachment are known, then a simple harmonic motion analysis gives the maximum acceleration for the steady-state vibration.

An experimental situation yielding a wave form bearing some resemblance to a practical transient shock vibration could be obtained by attaching the accelerometer to a cantilever beam of suitable stiffness and length. The beam could then be deflected statically and released suddenly, thereby executing damped oscillations. The magnitude of the acceleration can be computed from the force-deflection properties of the beam for any sufficient number of modes of vibration of the bar.

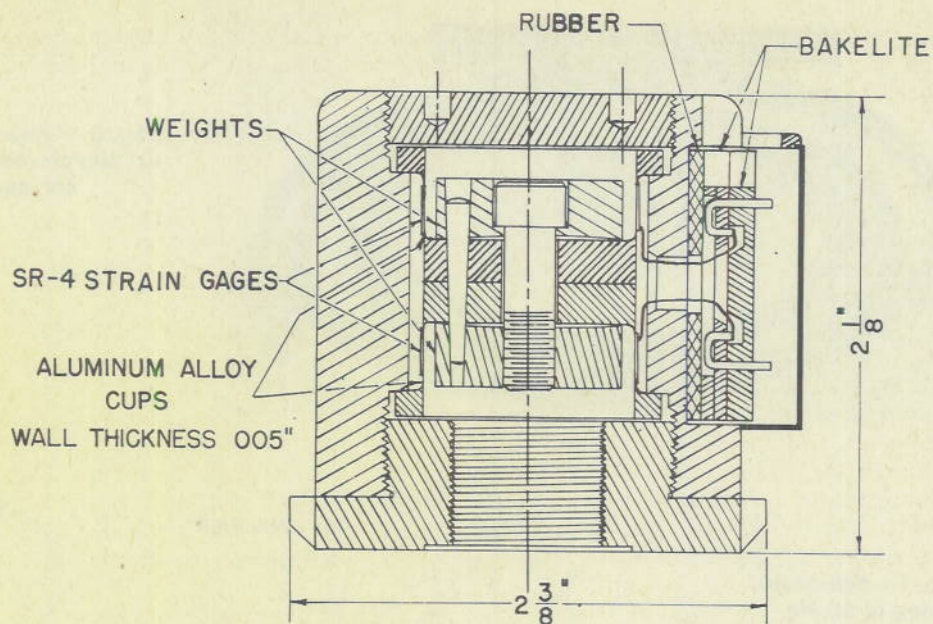


Fig. 4 - Cross-section view of strain gage accelerometer

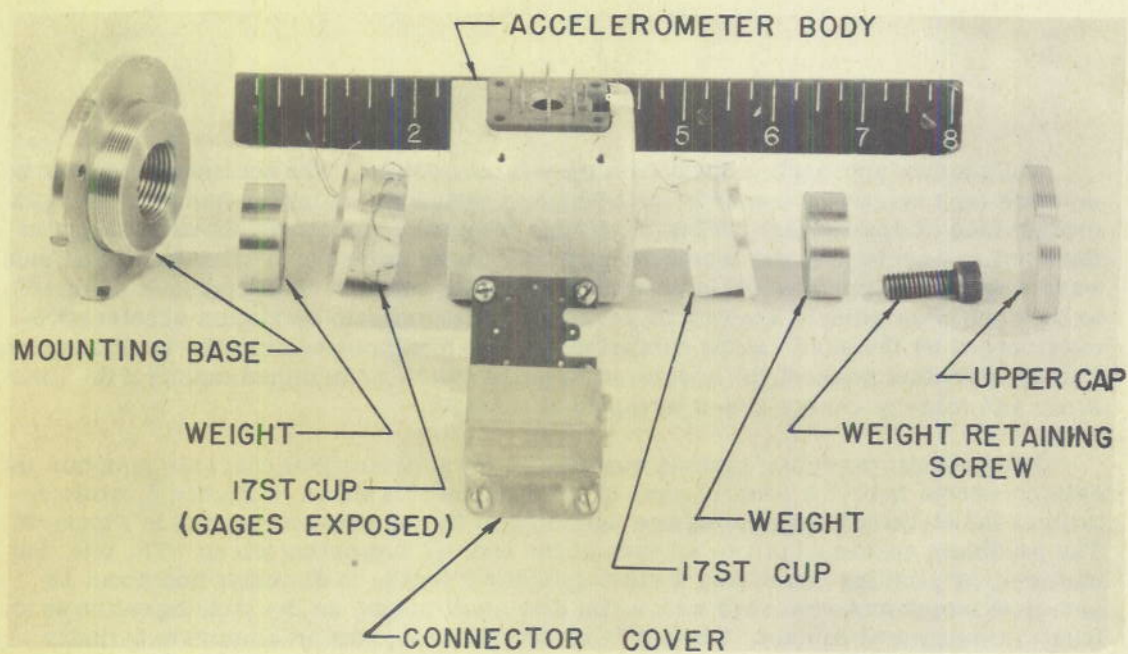


Fig. 5 - Exploded view of accelerometer components

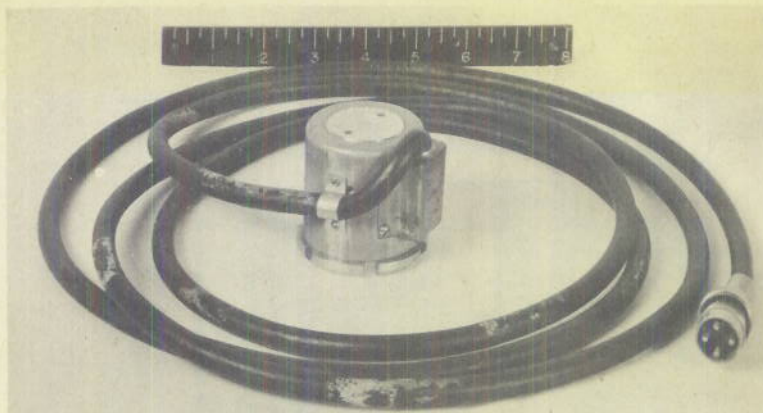
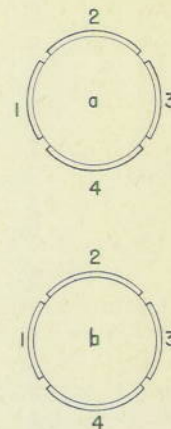
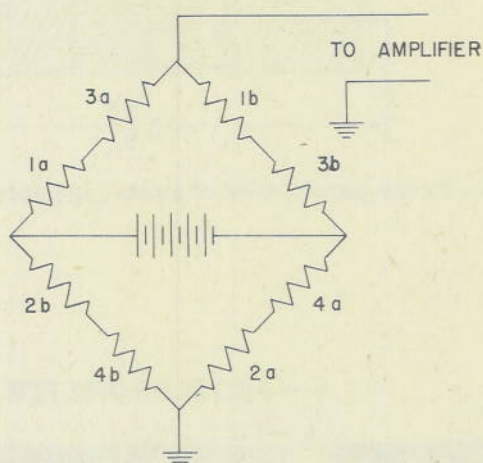


Fig. 6 - Assembled accelerometer, ready for use

Fig. 7 - Schematic wiring of strain gage accelerometer bridge



Yet another approach is the ballistic pendulum method. The accelerometer can be mounted on a relatively massive anvil which is struck by a similar hammer having a contact face of appropriate stiffness. If high frequency vibrations above 5000 cps in the anvil are neglected, the acceleration is of a pulse nature approximating a half sine wave when a stiff spring disc is used on the striking surface. If the pulse is assumed to be a half sine wave of known half period, the approximate maximum acceleration experienced by the anvil can be computed from the maximum velocity the anvil acquires. The acceleration scale of the acceleration pulse can be determined exactly if the time scale and velocity change are determined accurately.

The ballistic pendulum method was used as the acceleration generator to obtain the data presented here. A general view of the equipment is shown in Figure 8, while details of the electronic equipment are indicated in the schematic diagram in Figure 9. The pendulum consisted of two masses at the ends of supporting wires. The one, the hammer, or striking mass, was a steel cylinder 5 inches in diameter and about 12 inches in length and was fitted with a flat disc steel spring on the striking end so as to lengthen the time of contact. The anvil, or struck mass, was an aluminum cylinder of about the same dimensions with a convex hardened steel face bolted to the struck end. On the opposite end tapped holes were made to fit mounting studs for the accelerometers to be tested.

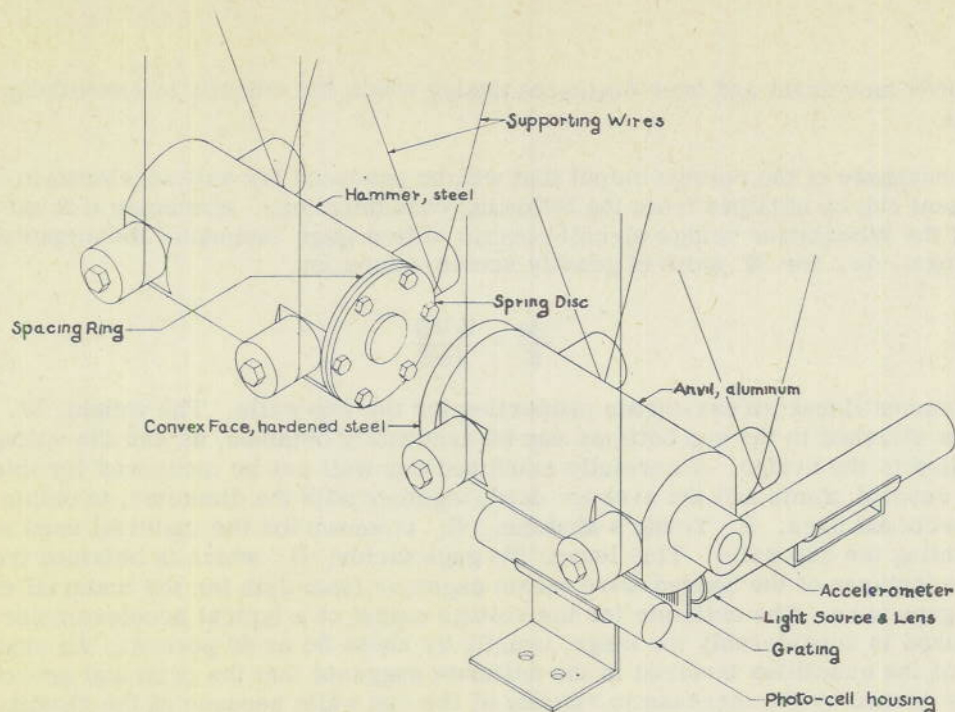


Fig. 8 - Details of pulse pendulum and photocell velocity measuring device

The velocity of the anvil was obtained by photoelectric means. A grating with vertical slits 0.010 inch wide and separations of 0.100 inch was attached to the under side of the anvil mass. Figure 8 shows the position of the essential components. A line filament lamp is focussed on the slits by a lens. When the anvil moves, due to the impact of the hammer, the slits will interrupt the light reaching the photocell at a rate directly proportional to the velocity of the anvil. By recording the signal from the photocell on the same trace immediately after the accelerometer signal has passed through the amplifier, it is possible to determine the frequency of the light interruptions and hence the velocity. This record is made for the initial portion

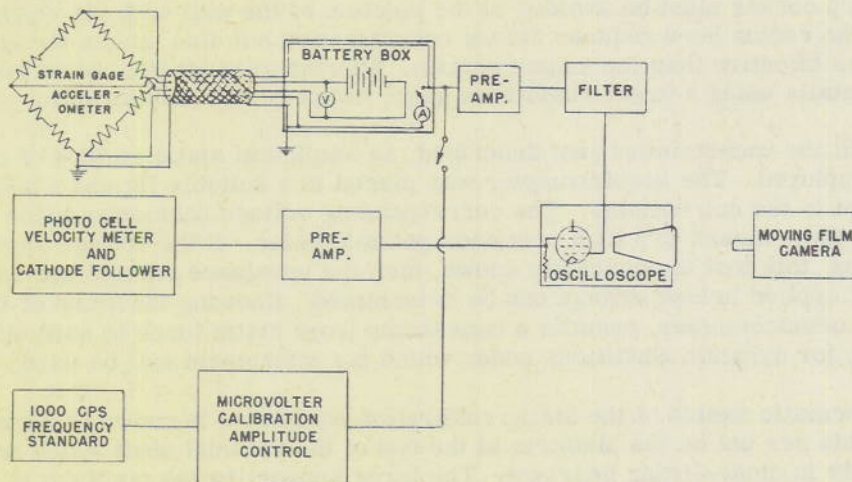


Fig. 9 - Electronic circuit details

of the anvil movement and over distances during which the velocity is essentially constant.

An estimate of the voltage output that will be produced for unit acceleration of the instrument can be obtained from the following considerations. Assuming that all four arms of the Wheatstone bridge circuit contain active gage elements, the output voltage change,  $\Delta e$ , for  $N$  units of gravity accelerations is:

$$\frac{\Delta e}{N} = \frac{WVG}{2AE}$$

This assumes linear stress-strain properties for the cup walls. The weight,  $W$ , of the mass attached to the cup bottoms can be accurately obtained, as can the voltage,  $V$ , applied to the bridge. A carefully machined cup wall can be measured for thickness at several points and the average used, together with the diameter, to obtain the cross sectional area,  $A$ . Young's Modulus,  $E$ , is known for the material used in constructing the cup walls. This leaves the gage factor,  $G$ , which is obtained from the manufacturer of the bonded-wire strain gages or from data for the material used for the gage wire. The estimate for the voltage output of a typical accelerometer by this method is considerably too large, usually by about 30 or 40 percent. An examination of the quantities involved in the estimate suggests that the principal source of error is caused by the increase in rigidity of the cup walls because of the rigidity of the attached gages. Since the base on which the gage is cemented is thin (.005-inch wall thickness for the particular gage desired) the gage with its associated paper backing and insulation is of the same order of magnitude in thickness. Hence, when the cup wall is coated with these strain wires, the average cross-sectional area of the tube is not that of the metal alone. Furthermore, the Young's Modulus for the strained portion of the cup is no longer that for the pure metal only but for some composite mixture of wall metal, cement, paper, and resistance wire. This complicates the estimate of the product  $AE$ . The gage factor,  $G$ , is reliable when the gage is cemented to a relatively large, uniformly strained area; however, the present application of small gages to the curved surface of a short tube introduces practical difficulties affecting the nominal gage factor. It is not always possible to cement the gages exactly parallel to the axis of the tube. Furthermore, the active length of the gage is reduced when the gage length is about the same as the length of the tube wall, since a sharp corner must be avoided at the junction of the wall with the cup bottom and rim. The radius here reduces stress concentration but also makes the ends of the gage less effective than the center portion. Shorter gage lengths would reduce this error but entails using a lower resistance gage, which is also undesirable.

To avoid the uncertainties just described, an empirical static method of calibration was also employed. The accelerometer was placed in a suitable jig and a known static force applied to the cup bottoms. The corresponding voltage unbalance of the bridge was measured by means of a high precision potentiometer. If the voltage applied to the bridge during this test is accurately known, then the unbalance voltage per unit force and per unit applied bridge voltage can be determined. Knowing the value of the central mass in the accelerometer, permits a conversion from static force to units of gravity acceleration for dynamic conditions under which the instrument will be used.

The schematic sketch of the static calibration equipment is shown in Figure 10. Known weights are put on the platform at the end of the vertical shaft which is free to slip axially in close-fitting bearings. The force supporting the platform is thus transferred to the accelerometer cups in the same positions they occupy when dynamically deflected. A Type K Leeds and Northrup potentiometer measures the bridge

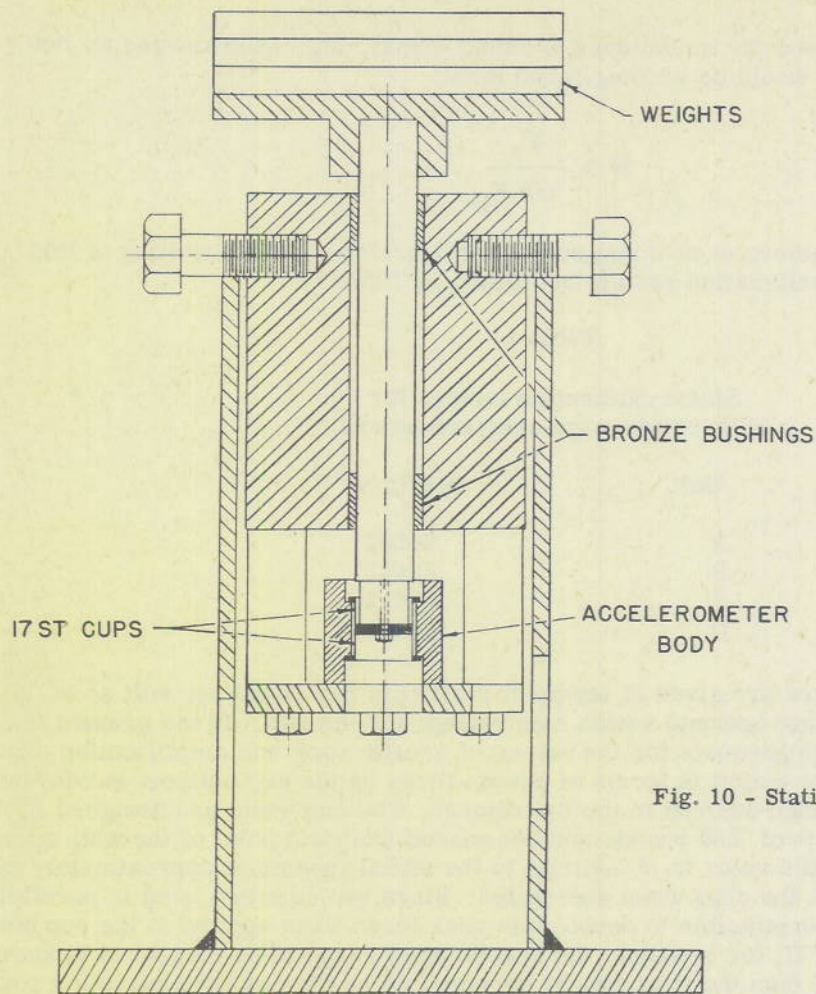


Fig. 10 - Static calibration apparatus

unbalance, and a precision voltmeter indicates the battery voltage. Several medium capacity 45-volt "B" batteries in parallel provide a sufficiently stable source of bridge voltage. The push-pull connection of the strain-gage elements in the accelerometer bridge circuit provided adequate temperature compensation for the accuracy of calibration desired in this instance.

The data obtained from this test determines a constant,  $k$ , such that

$$k = \frac{\Delta v}{F \cdot V_B}$$

where  $\Delta v$  is the unbalance voltage of the accelerometer bridge caused by the application of a force,  $F$ , to the cup bottoms when a battery voltage,  $V_B$ , is applied to the bridge. In terms of acceleration or number of  $g$  units,  $N$ , and the weight,  $W$ , of the central mass, the expression for  $k$  becomes:

$$k = \frac{\Delta v}{N \cdot W \cdot V_B}$$

In dynamic applications the instantaneous output signal,  $V_S$ , is measured so that the evaluation of  $N$  would be accomplished from:

$$N = \frac{V_S}{k \cdot W \cdot V_B}$$

Four accelerometers were completed and calibrated up to the writing of this report. The static calibration results are listed in Table 1.

Table 1

Static calibration factors for  
NRL strain-gage accelerometers.

| Unit | Mv/lb/v |
|------|---------|
| A    | .00683  |
| B    | .00683  |
| C    | .00619  |
| D    | .00791  |

The calibration factors are given in terms of millivolts per pound per volt so as to leave the bridge voltage optional within safe ratings for the unit. In the present instance, 45 volts were adequate for the values of acceleration and amplification available. Expressing the output in terms of pounds force on the cup bottoms is convenient for estimating the range desired in the instrument. The cup walls are designed to withstand a peak force of 250 pounds and not exceed the yield point of the wall material (17ST aluminum). This value is in addition to the initial tension of approximately this same value placed on the cups when assembled. Since two cups are used in parallel mechanically, it is permissible to double this peak force when applied to the cup bottoms or 500 pounds. If, for instance, the acceleration range to be measured is known not to exceed 1000 g, then the total weight permissible on the cup bottoms is 0.5 pound. Actually it is not physically possible to put that much weight into the cup if steel is used, due to lack of space. The maximum weights used have been about 0.19 pound, which fills both cups. The lowest practical weight permissible is about 0.1 pound, which gives a maximum acceleration range for the present design of about 5000 g. The natural frequency of vibration of the cup and weight system will be different when the weights are changed. For the heaviest weights used (0.19 lb.) the natural frequency is 7300 cps. Figure 11 displays a typical set of records from the accelerometer.

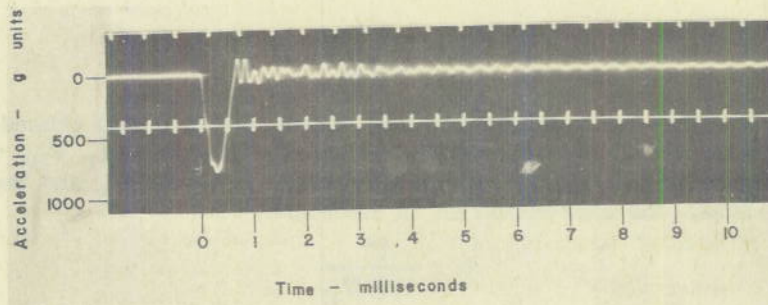


Fig. 11 - Oscillogram of pulse pendulum test

If one neglects the superimposed vibrations on the filtered signal as shown in Figure 11, the form of the acceleration pulse closely resembles a sine wave. This suggests the use of the contact time and the independently measured velocity, as read from the photocell velocity meter, to obtain the acceleration for comparison with the accelerometer output. An auxiliary electrical circuit was arranged through the contact area of the hammer and anvil so as to make two very brief deflections of the oscilloscope indicating the beginning and end of the contact period. If the acceleration of the anvil be assumed sinusoidal, the velocity can be obtained by integration.

$$a = a_0 \sin \omega t$$

$$v = \int_0^{\tau} a_0 (\sin \omega t) dt$$

or

$$a_0 = \frac{\pi v}{2g\tau} \quad \text{if } \tau = \frac{\pi}{\omega}$$

A graph of peak acceleration read from the strain gage accelerometer against the computed peak acceleration using  $a_0 = \pi v / 2g\tau$  is shown in Figure 12.

A further comparison can be made independently of any assumption regarding the wave form of the acceleration experienced by the anvil. This is done by integrating the acceleration signal from the accelerometer and comparing the resultant velocity with the measured velocity of the anvil. A typical set of data is displayed in graphical form in Figure 13.

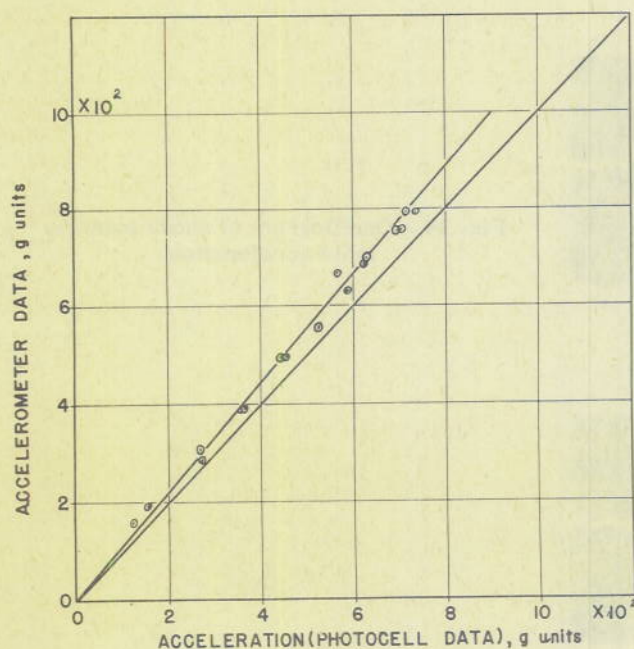
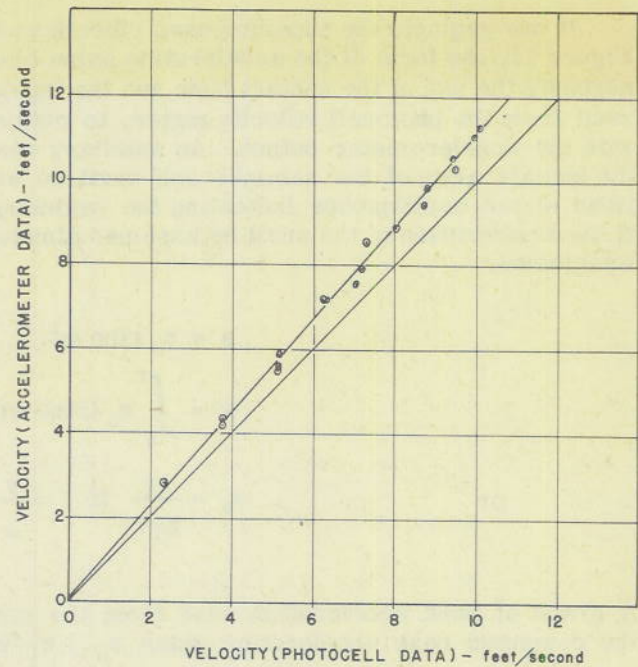


Fig. 12 - Correlation of acceleration calibration data

In both Figure 12 and Figure 13, ideal correlation would have required that the points lie on the 45-degree line indicated on each graph. The deviation from this line is approximately 10 percent. The fact that the scatter of the points is not wide enough to extend below this line indicates a systematic error of some sort. A probable cause may be that the static calibration constant cannot be applied to dynamic conditions without modification when greater accuracy is desired. However, it is gratifying to observe that the static test, a completely independent method of calibration, can yield results of this degree of accuracy for dynamic application of the instrument. An adjustment of the sensitivity constant can be made from the pendulum data for each instrument to make the velocity correlation perfect.

Fig. 13 - Correlation of velocity data



In Figure 14 is shown an oscillogram of the acceleration experienced by the table of a small shock machine when struck by a hammer. The velocity of the table was measured simultaneously by the same photocell technique as was employed on the pulse pendulum. Agreement within 5 percent was obtained between the velocity indicated by integration of the accelerometer trace and the independently measured velocity.

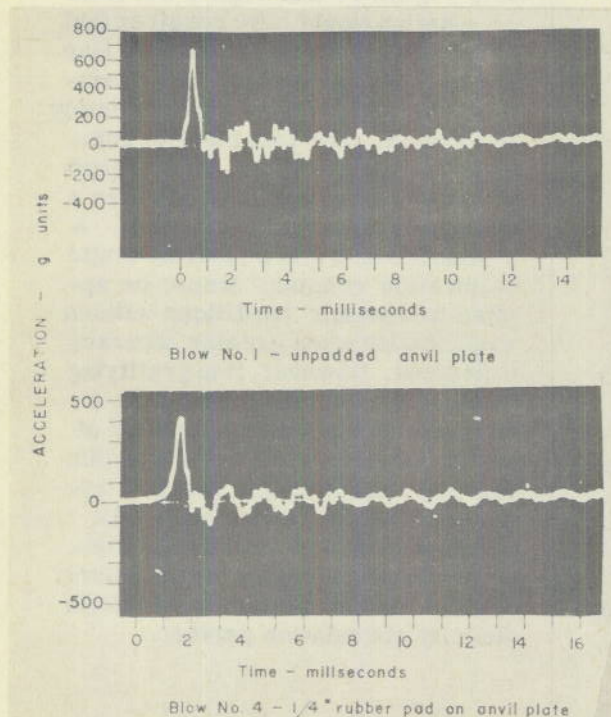


Fig. 14 - Oscillogram of shock machine table acceleration

## **S A V I B U L L**

In the Shock and Vibration Bulletin No. 6 (pages 102-3) a letter to the editor was published concerning the "Chamberlain shock-wave absorber". The letter included a description of the principle upon which the absorber is proposed to operate. This Laboratory has recently completed a comparative study of this absorber and the results may be found in NRL Report No. O-2976, dated 23 September 1946. Briefly the results may be summarized by repeating here the concluding paragraph.

"The theoretical and experimental work presented in this report indicates that the Chamberlain mount is not appreciably more effective as a shock and vibration absorber than a similarly shaped piece of copper, steel or other material of comparable rigidity. It is recommended that this type of mount not be used unless specific test data show its effectiveness in a particular application."

