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SHOCK AND VIBRATION BULLETIN

NO. 6

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NAVAL RESEARCH LABORATORY
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THE SEVENTH SYMPOSIUM
Naval Research Laboratory
5 November 1947

VEHICULAR SHOCK AND VIBRATION INSTRUMENTATION AND MEASUREMENTS WITH SPECIAL CONSIDERATION OF MILITARY VEHICLES

By

Frank K. Priebe

Squier Signal Laboratory

Experimental measurements are described, which were made during the war in order to determine the frequencies and amplitudes of the vibrations and road shocks encountered in motor vehicles and trailers. A review of the instrumentation and the results obtained from prior investigators are given. The measurements were made in the Aberdeen Proving Ground, since it was felt that this road represents the most common road conditions that give rise to large transient and steady state vibrations in vehicles.

The purpose of these measurements was to obtain design data for electronic equipment to be transported in vehicles over rough terrain.

It was found that three frequency ranges predominated:

The Spring Frequencies – 2 to 4 cycles
3 to 5 G max. amp.

The Tire Frequencies – 8 to 15 cycles
2 G max. amp.

The Structural Body Frequencies – 60 to 100 cycles
1 G max. amp.

The limitations of the instrumentation and difficulties encountered during these measurements, which must be taken into consideration in the interpretation of the results, are pointed out.

On the basis of these investigations, the performances of several machines available for the purpose of simulating vehicular transportation are analyzed.

Procedures for obtaining calibrations of instruments and checking the performance are discussed.

Plans for future investigations are discussed.

PURPOSE

Various investigators have approached the problem using widely varying tools. Therefore, the results obtained must be interpreted on the basis of the tools used and, whenever possible, on the basis of exact conditions under which the measurements were made, such as location of the point where the measurement was made. It is the purpose of this report to analyze the test results, taking the above-mentioned facts into consideration. It is also the purpose of this report to list instruments that are or have been used here and abroad for measurements, their characteristics, applicability, and limitations.

On the basis of these measurements, machines are described that simulate field conditions. These machines are intended to be used as testing machines for the purpose of shock testing electronic instruments or equipments to be installed or transported in vehicles. Several types of these machines have been developed by the Armed Forces and by industry. The characteristics of these machines are illustrated and discussed. The scope of the report is restricted to land vehicles only, with particular emphasis placed on road motor vehicles.

DISCUSSION

Requirement Placed on an Instrument to be Used for Measuring Road Shock and Vibration During Vehicular Transportation and Handling.

Only within the last two decades has equipment been made available for the study of shock and vibration in vehicles. Not all instruments used for laboratory experiments are suitable for use in moving vehicles. In the laboratory, experiments can be conducted under ideal conditions - for instance, a fixed set of reference axis is generally available. This is not the case in a vehicle, and measurements are restricted to relative motions only. None of the instruments have all the characteristics desired. Compromises have to be made or the characteristics of the instrument have to be taken into consideration in the interpretation of the results obtained. For instance, not all the instruments cover the desired frequency range with the required sensitivity.

Listed below are the requirements that must be met by instruments in order to make them ideal for use in vehicles:

1). The instrument must be able to withstand the maximum vehicular vibration encountered without any damage to it.

2). The instrument must be of the recording type.

3). The mechanism outside the vibration sensitive element must be so sturdy that the error caused by the vibrations of it is a minimum.

4). The instrument itself, or if the vibration sensitive element is separate from the complete instrument (vibration pickup), should be light in weight and physically small.

5). If possible, the frequency range from at least 0 to 400 cycles should be recorded.

6). If displacements are measured, it should be possible to record displacements up to 6 inches. In general, such an instrument is cumbersome. If this is not possible, due to the space limitations, the characteristics of the displacement pickup should be such that the low-frequency, high-amplitude displacements are attenuated and do not saturate the instrument for higher frequency displacements.

7). If velocities are measured, velocities up to 100 inches per second should be recorded.

8). If accelerations are measured, the instrument should at least withstand an acceleration of 10 g for vibration measurements and 200 G for occasional shock measurements.

9). The minimum value of displacement to be recorded should be at least .1 inch.

10). The minimum velocity to be recorded should be at least 20 inches per second.

11). The minimum acceleration to be recorded should be at least .1 g.

12). All recording instruments should provide a known time scale; a minimum speed of the recording medium of 5 inches per second is desirable.

Static Measurements on Vehicles

The following information is of great use in studying vibrations on vehicles (see Ref. 21, Chapter 10):

1). Weight of vehicle (with load).
2). Center of gravity.
3). Moment of inertia.
4). Spring constants.
5). Frame and body stiffness.

Center of Gravity

The following quantities can be measured readily:

1). The total weight of the vehicle = $W_1 + W_2$.
2). The weight under the front wheels = W_1.
3). The weight under the rear wheels = W_2.
4). The length of the wheel base = L.

Let (a) and (b) be the co-ordinates (distances) of the projection of the center of gravity onto the wheel base. Measure from the front and rear axle,

$$\text{then } a = \frac{W_2 \cdot L}{W_1 + W_2}$$

$$b = \frac{W_1 \cdot L}{W_1 + W_2}$$

Height of Center of Gravity

For the purpose of determining the height of the center of gravity, it is necessary to block the springs of the vehicle and jack up the one side of the vehicle a distance h = about 8 inches, then the height of the center of gravity above the plane through the contact point of the tires is

$$H = \Delta W \cdot \frac{G^2}{Wh} \text{ approx.}$$

where ΔW = the difference between the weight on the low wheel and the wheel that has been raised

G = the distance between wheels

W = total weight

h = the height the wheels on one side of the vehicle have been raised.

The above method is simple but not necessarily the most precise one. General Motors, in their proving ground, use the following method (see Ref. 22).

Alternate Method of Determining the Centroid of a Vehicle.

In this test the vehicle is suspended on a platform in such a manner that the platform and the vehicle form a pendulum, which is free to swing about an axis perpendicular to the wheel base of the vehicle, and the period T_1 of the free vibration is measured. The test is repeated with a platform which has different pendulum length; again the period T_2 of the free vibration is measured. Then the height h_0 of the center of gravity above the platform (if the weight of the platform is small compared with the weight of the vehicle)

$$h_o = \frac{(T_1^2\, y_1 - T_2^2\, y_2) - \frac{4\pi^2}{y}\,(y_1^2 - y_2^2)}{\frac{8\pi^2}{g}\,(y_1 - y_2) - (T_1^2 - T_2^2)}$$

where y_1, is measured from the point of suspension of the first pendulum to its baseline. Y_2 is the corresponding measurement on the second pendulum. T_1 = the period of the second pendulum. In a similar manner, the radius of gyration and the moment of inertia can be determined.

Moment of Inertia Determination

The springs of the car are blocked so that the springs and the tires cannot deflect, and the vehicle is supported on a platform which rests on a knife-edge below the rear axle and on a spring below the front axle. The natural frequency of the assembly and the spring constant are determined.

$$\text{Then } I = \frac{82\ k\ L^2}{F^2}$$

L = Wheel base length

F = Natural frequency

The Spring Rates k

It is customary to measure the spring rates directly on the vehicle. It is done by placing the four wheels of the vehicle on scales and changing gradually the load on the scales by jacking up the vehicle. In this manner, the spring rates for up and down motion, roll, pitch and the individual rates of the springs and tires can be determined.

Frame and Body Stiffness

For this test the vehicle is supported at the point where the springs are connected to the body. A Lazan Oscillator can be placed on the vehicle and the speed adjusted as required to determine the resonant frequency of the individual parts of the body. The stiffness of the frame can be determined by measuring the force required for twisting it. (It should be noted that these spring rates are not linear functions of the applied forces.)

Wheel Alignment

The alignment of the wheels and the effect of unbalance of the wheels as causes of vibrations are briefly mentioned.

Dynamic Laboratory Measurements

All the dynamic characteristics of a vehicle may be determined by laboratory drop tests.

Figure 1 illustrates the test procedure. The vehicle is jacked up on blocks either under one wheel or under the axle, and the blocks are then knocked out from under it. By means of a recording device described later in the test, the amplitude of the displacement of either a point of the body or the acceleration of a point of the body is obtained. The frequencies which predominate can be identified as frequencies of the tires, the vehicle springs, and the natural frequencies of the body.

Wheelhop

Wheelhop is the vertical vibration of the unsprung masses of the vehicle between the tire and the suspension. Two different types of vibration can be noted:

1) The tire leaves the ground and then the frequency of the vibration becomes higher, since the spring rate changes. This vibration is called "tramp."

2) The tire stays in contact with the ground. Then the frequency is the one which is generally designated as the tire frequency.

In general, it can be stated that an obstruction greater than 1" in a road and a

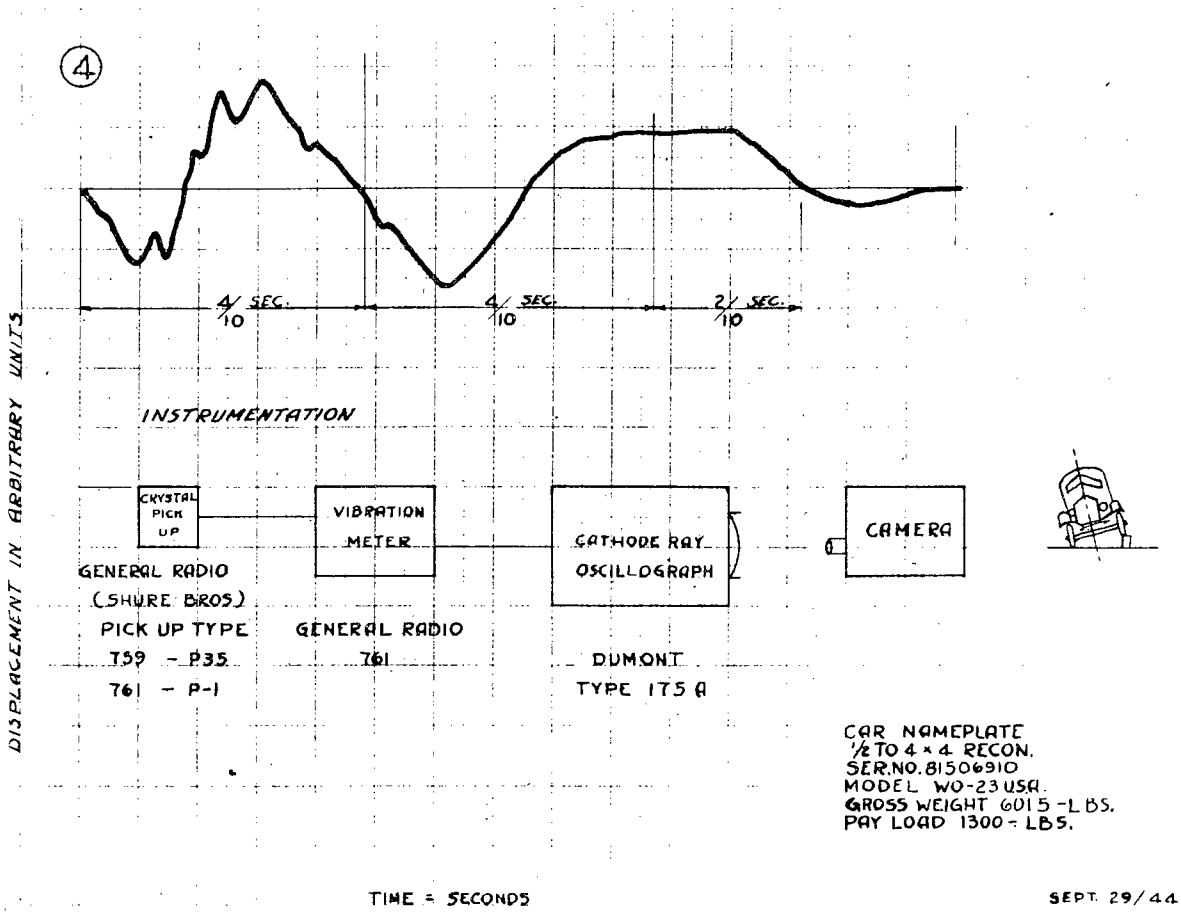

Figure 1

speed of the vehicle greater than 15 miles per hour will cause the tire to jump off the road. Figure 2 illustrates this phenomenon as measured on the General Motors Bump Rig. The Bump Rig consists of a large drum. On this drum cams are mounted, and the vehicle is placed so that the rear wheels rest on the drum. If the car is running and the speed increased, the wheels will leave the drum until the amplitude becomes less than plus or minus 1 inch. If the speed is then decreased, the amplitude will increase and jumping will continue until it stops suddenly at 3 inches.

Early Field Measurements by Other Foreign and Domestic Laboratories

Field measurements have been made by various investigators. However, very few of the records have been correctly interpreted.

Observations can be made, for instance, simply by driving behind the vehicle to be tested and watching the reflection of a light beam from the rear window. The shape of the Lissajous' Figures observed in the reflection of the rear window gives an estimate of the wave form of the relative motion between the two vehicles. If the light source is stationary, a good estimate of the amplitude of the motion can be obtained.

As mentioned above, the absence of a fixed reference system in a moving vehicle increases greatly the difficulties in making any measurements in a moving vehicle.

Since it is not possible to make an absolute measurement with an accurate system of reference axis, most measurements that have been made are based on the re-

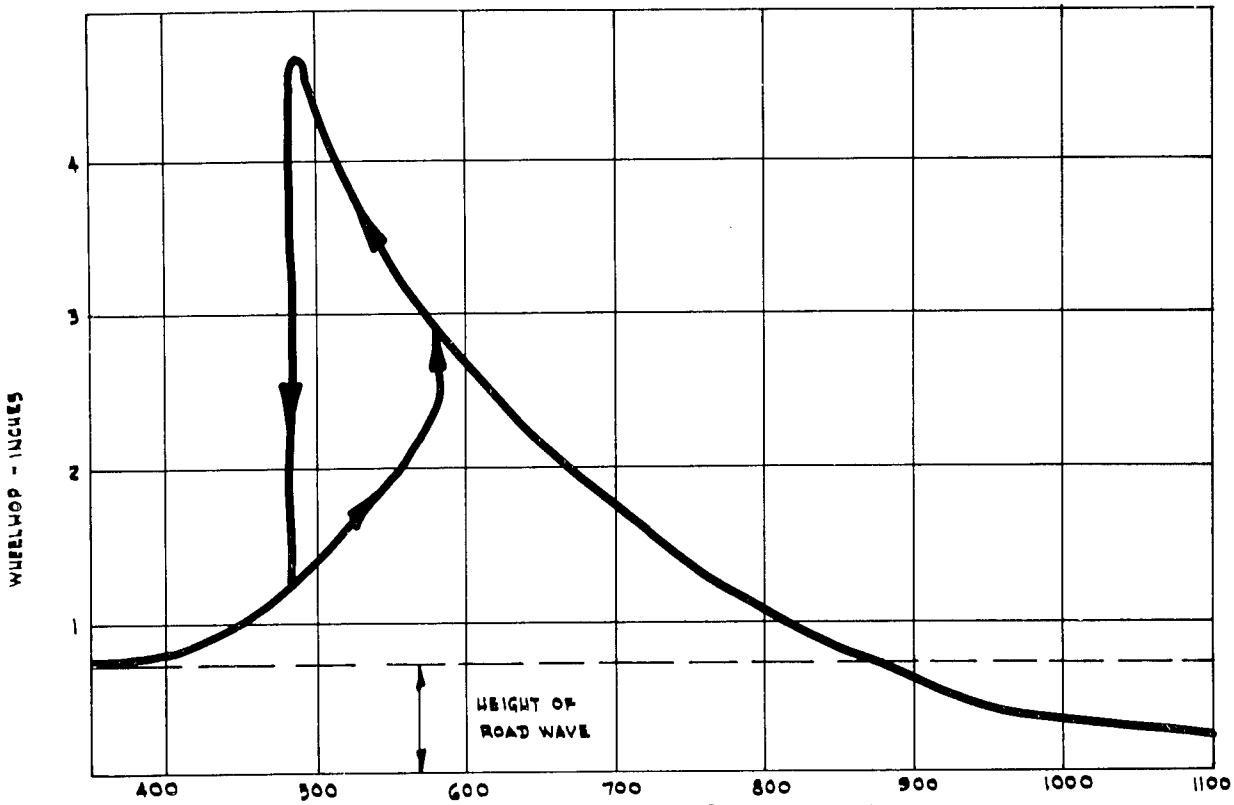

Figure 2 Frequency of Road Wave-Cycles per Minute
Wheelhop Due to Road Wave - 3/4" Cams

cording of the relative motion of a seismic mass with respect to the vehicle or are the recording of the forces that exist in certain structural members of the vehicle. Since the forces involved, at which certain structural members of the vehicle fail, can be very accurately determined, it is possible to estimate from the field failures on vehicles the maximum acceleration the vehicle itself can withstand. For instance, the deformation on tank type T26E1 due to shock suspension is such that an overall acceleration of 6g will put the tank out of operation. (see Ref. 25)

On the basis of these observations, General Motors reports the following information:

Spring Frequencies	Max. accelerations 2.2g
Passenger cars (civilian type)	1 to 1.6 cps
Tanks, pitch frequencies	.83 to 1.66 cps
Tanks, bounce frequencies	1.16 to 2.0 cps
Cross country vehicles and trucks	1 to 2.5 cps

Information is available that some vehicles will withstand as much as 9g, based on measurements of deformation of parts.

Wheelhop	
Passenger cars	(13 cps (1.2 inch displacement (2.25 g acceleration
Engine Vibrations	
Soft engine mountings	6.6 to 20 cps
Hard engine mountings	30 cps

High-Speed Motion Picture Studies

Very valuable information can be obtained concerning the spring frequency and tire frequency type of vehicular vibration from the study of high-speed motion pictures. Typical of examples of such films are the following ones:

Ordnance Research & Development Center, Aberdeen Proving Ground

1) 40 mm Gun Mount M5 test of suspension for designated APG-809.
2) Naval Research Lab. and Squier Signal Laboratory.

For the purpose of evaluation of such films, it is desirable to have a scale in inches available in the field of view of the camera. This can be achieved by simply pasting alternately black and white tape to the body of that part of the vehicle that is to be studied. Several types of cameras are available for such studies, such as:

1) Eastman Kodak 16 mm High-Speed Camera
2) Western Electric Fastex Camera

It is desirable not to let the speed of the camera exceed 400 frames per second. In some cases, the camera has to be modified in order to have a steady speed at this picture frequency.

Measurements made by the Chrysler Ordnance Laboratory

During the war the Chrysler Ordnance Laboratory made some tests in connection with failures of some automotive components such as fuel gauges, headlights, relays, etc. (Ref. 3 and 4).

All the components which were investigated by Chrysler were very light and, also, rigidly mounted to the chassis of the vehicle. It follows that the frequencies observed were more or less the frequencies of the supporting structure. Some of the individual structural parts of the vehicle may have resonant frequencies as high as 800 cycles per second and 20 g in acceleration peak amplitude.

Chrysler used a Bell Laboratory Erpi Frequency Recording Analyzer and Sperry-MIT vibration velocity pickups. Typical records are shown in Figure 3. Measurements were made either by striking the tire with a ball-peen hammer or by driving the vehicle over a rough cobblestone road.

Chrysler made other measurements in order to eliminate the failures of headlights and gasoline tank sending units. The maximum total acceleration measured on a tank, Chrysler Ser. 634 (the roughest riding tank in the Chrysler Proving Ground) was 20.1 g at 325 cycles. This acceleration is abnormally high. Steel tracks and steel bogie wheels were used. The remarkable thing about these measurements is their similarity when taken under the same conditions, even though the instrumentation is not particularly designed for these measurements. Another interesting feature is the saw-tooth nature of the records. This feature has also been observed with other types of instrumentation. For each vehicle and for each individual installation of the measuring equipment, certain frequencies will predominate and others will be entirely absent.

General Electric Displacement Recorder

1) Brief Description and Theory of Operation

A seismic mass is suspended by a soft spring inside a brass tube. The weight is centered by means of some friction dampers made out of felt. Attached to the mass is a stylus that records the motion of the weight on a wax paper record. The wax paper moves with constant speed past the stylus; simultaneously a timing signal can be recorded.

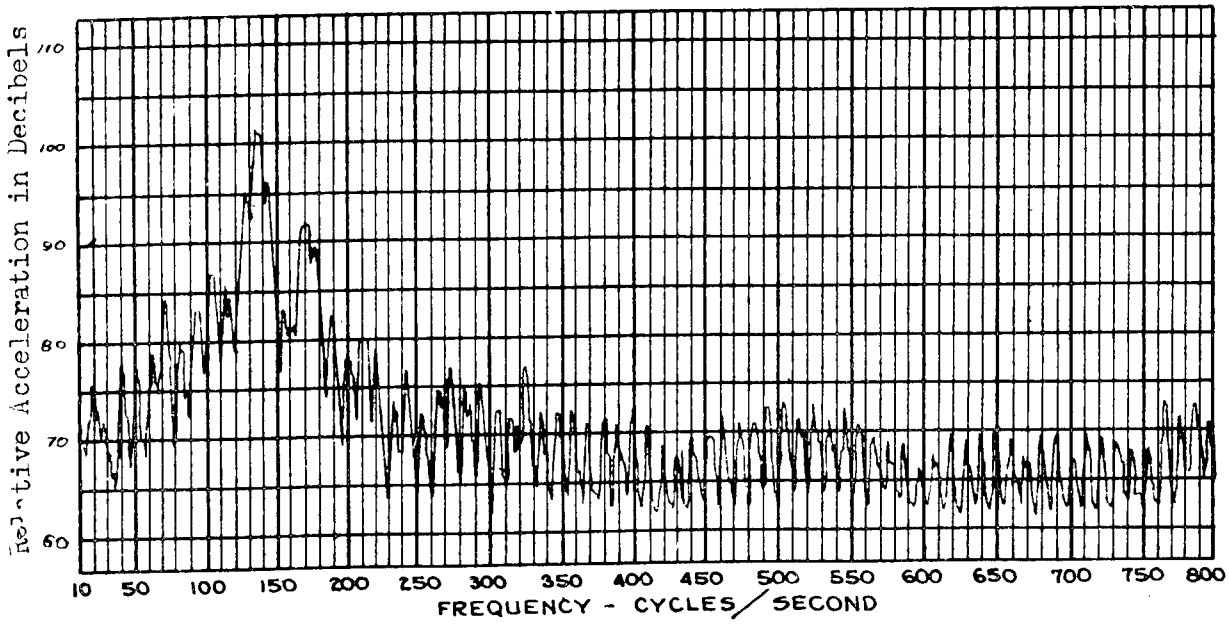

Figure 3 Typical Record made by Chrvsler with Frequency Analyzer Setup

2) Schematic Diagram. See Figure 4.

3) Amplitude Range. Amplitudes up to three inches may be recorded.

4) Frequency Range. From less than 1 to 35 cycles per second.

5) Resonant Frequency. Less than 1 cycle/sec.

6) Sample Records. See Figure 5.

7) Outstanding Characteristics, Advantages, and Disadvantages.

This instrument has the following advantages:

a) It has a very low resonant frequency.
b) It has a sufficient amount of damping so that the amplification is not excessive at resonance.
c) The maximum amplitude to be recorded is limited only by the width of the paper available.
d) The only power required is the one necessary for the motor of the paper drive of the paper transport. This electric motor could be replaced by a spring motor in order to make the instrument independent of any electrical power source.

The following are the disadvantages:

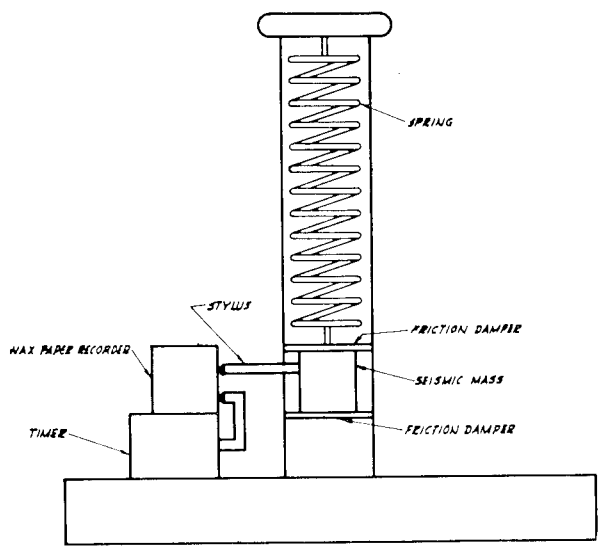

Figure 4 Schematic Diagram of G. E. Vibration Recorder (Vertical Displacement Recorder)

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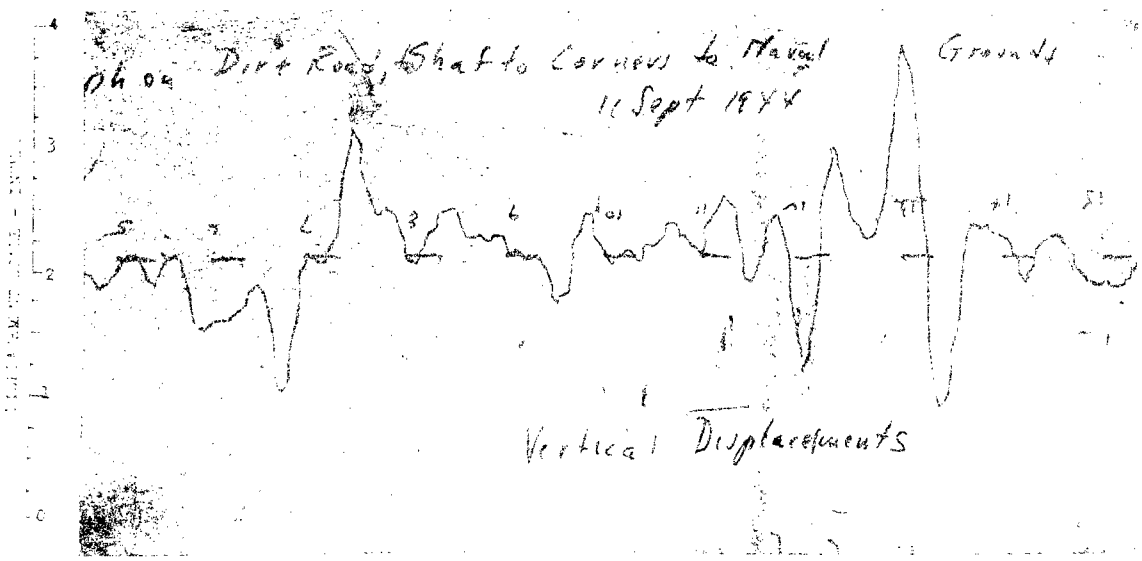

TIME IN SECONDS

Vehicle: USA W2051215 MT 53
Road: Dirt Road
Vibration Measured: On floor of truck, rear, right hand side
Metering System: G.E. Vertical Displacement Recorder

Figure 5 Vehicular Vibration Test on Truck, 1/2 Ton, 4x4, Recon on Roads in Monmouth County, New Jersey

a) Great bulkiness. (Instrument cannot be used in installation where space is at a premium.)
b) The recording of high frequencies is inaccurate, due to the response limitations of the stylus and spurious vibration of the spring.
c) At high amplitudes and high frequencies the spring is vibrating in many different transverse modes which interfere with the recording.
d) In the model used, the amplitude range was 3 inches. This is not sufficient, since the very low frequencies greatly exceed this range.
e) The damping was not readily adjustable.
f) At low frequencies, the resonant frequency of the instrument is easily excited.

The Schrader Three-Component Accelerometer

1) Description and Theory of Operation.

This accelerometer is an entirely mechanical system using optical and photographic means for recording. The seismic mass is mounted on a cantilever spring. An adjustable damper in the form of an air bellows is provided to cut down the amplitude at the resonant frequency. Figure 6 shows the schematic functional diagram of the instrument. The translatory motion of the weight mounted on the cantilever spring is transformed into a rotational motion of a steel shaft. A mirror is mounted on this steel shaft to provide a reflecting surface for the light beam that is projected from a fixed stationary lamp. The light is re-

flected into a photographic surface for recording purposes and onto a ground glass screen for visual observation purposes. The motion of the light spot on the screen or on the film is proportional to the angle of rotation times the radius. In such manner, an amplification of the recorded vibrations is obtained.

2) Schematic Diagram. See Figure 6.

3) Frequency Range. 0 to 75 cycles.

4) Resonant Frequency. Between 75 and 150 cycles per second.

5) Amplitude Range. 2 to 3 g.

6) Sample Record. See Figure 7.

7) Outstanding Features, Advantages, and Disadvantages.

The following advantages are in favor of the instrument:

a) Three components of acceleration perpendicular to each other are recorded.
b) Air damping is provided so that the instrument can be used satisfactorily up to the vicinity of resonant frequency.
c) The magnification of the amplitude is obtained by mechanical and optical means; no electronic devices are required.
d) The only electric power source used serves for the purpose of film drive, timing, and excitation and heating of the recording lamp. Variations of the voltage are permissible within wide limits.

The disadvantages are:

a) The photographic film used is not universally available.
b) The instrument weighs several pounds and, when placed on a vibrating object, changes the vibration characteristics of the object.

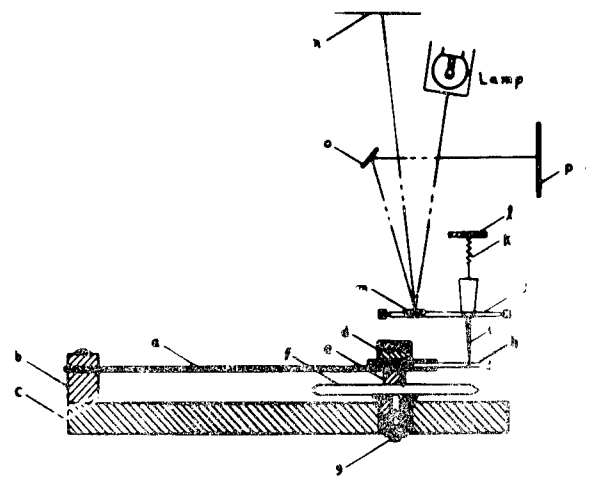

Figure 6 Schrader 3 Component Accelerometer

c) The optical system is affected by the temperature. The mirror changes its focal length.
d) The calibration, especially at the higher frequencies, is quite difficult.
e) The loading of the film in the camera is difficult.
f) The instrument lacks the rigidity required to withstand vibrations and shocks encountered

The Shrader, Tri-dimensional Vibrograph, made by the Vibration Specialty Co., Philadelphia, Pa.

1) Description and Theory of Operation.

This instrument operates on the same principle as the Shrader three-dimensional accelerometer system made by the Visration Specialty Company. The only difference from this type of accelerometer is that the resonant frequency of the cantilever springs is very low, so that displacements are recorded.

The Vibrograph Made by the Westinghouse Electric & Mfg. Co.

1) Description and Theory of Operation.

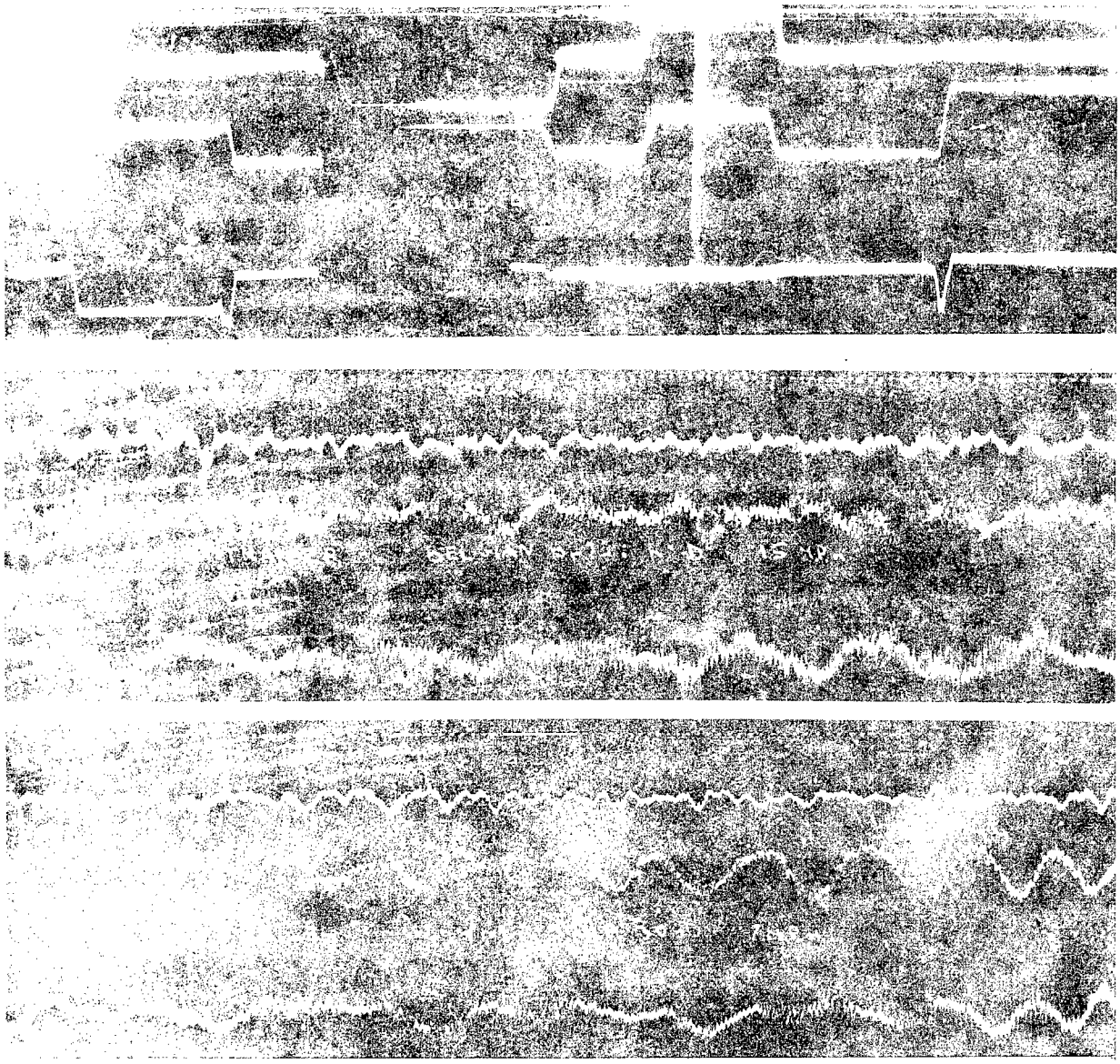

Figure 7 Snrader Accelerometer Record

The vibrograph made by the Westinghouse Company is an instrument that can be used for recording displacement amplitudes encountered in vehicles.

Figure 8 illustrates the operation, in schematic form, of this instrument. For such an arrangement, the following formula describes the relative motion of the seismic mass with respect to the frames:

$$\frac{y_o}{a_o} = \frac{W^2/W_o^2}{\left(1 - \frac{W^2}{W_o^2}\right) + \left(2\frac{C}{C_c}\frac{W}{W_o}\right)^2}$$

where y_o = relative motion of mass and frame

a_o = vibration amplitude

W = vibration frequency

W_o = natural frequency of the vibrograph between mass and frame

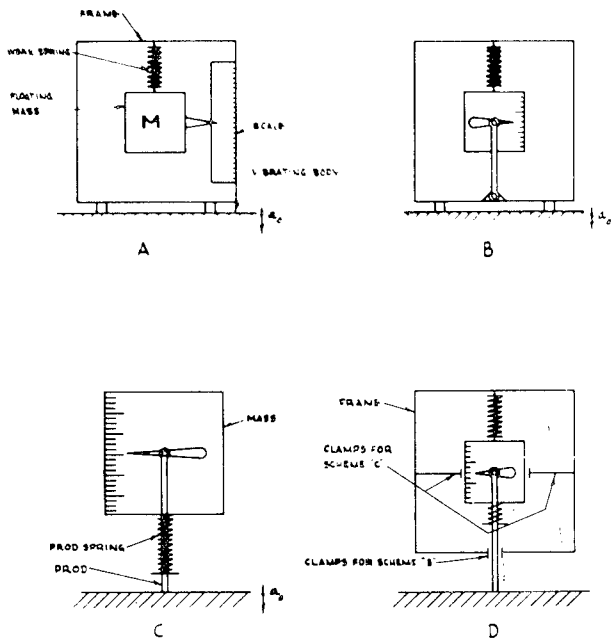

Figure 8 Westinghouse Vibrograph Principle of Operation

C = equivalent damping of vibrograph between mass and frame

C_c = value of critical damping

This relation is shown plotted on Figure 9. It can be seen that for very

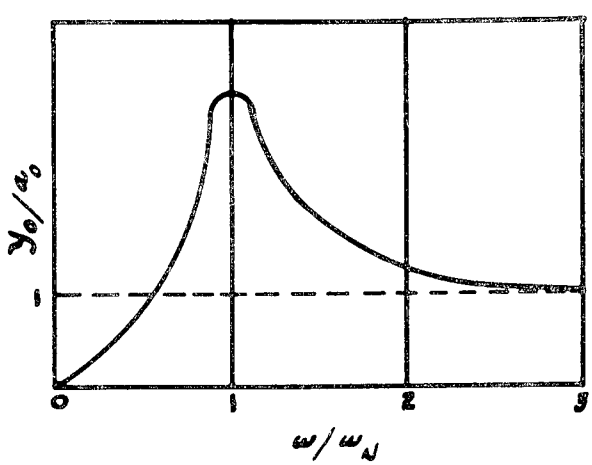

Figure 9

high frequencies this ratio approaches unity very rapidly; this means $y_0 = a_c$, or the mass is standing still with respect to a fixed system of reference axis. The recording is made by impressing a fine groove in a cellulose film. At the same time a timing wave is recorded, which timing wave is obtained by plucking a stiff reed at frequent intervals. The frequency of this reed is tuned to 30 cycles. The film speed is approximately 1.5 inches per second. For study, the record must be observed under a light that falls on the film almost parallel with the surface of the film. The overall dimensions of the Westinghouse Vibrograph are 4-1/4 inches x 4 inches x 6-3/4 inches. The magnification due to the stylus is 8.5 (mechanical amplification). The record can be studied under a low-power microscope furnished with the instrument, with magnification of 40. Therefore, the total magnification is 340 times this, and displacement amplitudes of 1/1000 inch or less can easily be read within 5% accuracy. The reproduction of the records has to be made with a photographic microscope or by tracing the curves directly on a Recordac enlarger.

2) Frequency Range. 10 to 250 cycles/sec.

3) Natural Frequency. 6.66 cycles per second for vertical operation

6.25 cycles per second for horizontal operation.

4) Amplitude Range. At 250 cycles per second the maximum amplitude range recorded is .002 inches; at 6.66 cycles per second the maximum amplitude recorded is .025 inches.

5) Outstanding Characteristics, Advantages, and Disadvantages.

The Westinghouse Vibrograph has the following advantages:

a) No electric power supplies are required, since the instrument is entirely spring operated.

b) The inertia of the recording stylus is as low as possible to prevent local resonance of parts.
c) The amplification is constant over a wide range of frequencies.

The following are disadvantages:

a) The records require special optical equipment for examination.
b) The instrument has the tendency to "bottom" at very low frequencies and at large displacements; however, during a test made at Coles Signal Laboratory at normal driving speeds in a tank M4, bottoming never occurred.

The Brush System of Recording Vibrations

i) Description and Theory of Operation

The Brush system of recording vibrations consists of the following component instruments:

12 volt storage battery
Dynamotor, Jeanette Genemotor, 12 volt input, 110 volt 60 cycle output.
Amplifier, Brush type 0A-1 is used.
Direct-inking crystal oscillograph type 0BC or 0BD is used.
Integrating box used only with Rochelle salt pickups.

The component part which is unusual for the Brush system and in which this system deviates from any other direct recording system is the oscillograph. The oscillograph is so constructed that the record is instantaneously visible without any further treatment.

The record is made with ink on paper by means of a very fine and light-weight pen. The pen is actuated by a piezo-electric crystal. If a signal is applied to the electrodes of this crystal, the crystal element twists about its axis. This motion is transmitted by a neoprene shaft to the pen. For full scale deflection 900 volts are required; therefore, special amplifiers are necessary. Two types of oscillographs were used; one, type 0BC, is twice as sensitive as the other one, type 0BD. Type 0BC has two 3 inch pens and type 0BD has two 2 inch pens. Type 0BC covers a frequency range that is flat from 1 to 60 cycles; type 0BD has a flat characteristic from 1 to 120 cycles. Frequencies of several hundred cycles may be detected if the input signal is strong enough.

Both oscillographs, type 0BC and type 0BD, have two channels available so that two vibrations can be recorded at the same time.

The damping of the pen is such that no free vibrations can be observed when the pen is removed by hand and then suddenly released.

The overall characteristics of the type 0A4 amplifier and the 0BC and 0BD oscillographs are shown in Figures 10 and 11. The second curve in two graphs gives the deflection in mm/g if a piezo-electric Rochelle salt crystal is used. In vehicular measurements of vibration it is necessary to use an integrating circuit in combination with a crystal, not for the convenience of measuring displacement or velocity but in order to avoid pickup of stray voltages by the long crystal cords and, also, in order to minimize the effect of erroneous recordings due to the motion of the connecting cords of the pickups.

Very good shielding is required for all cords in the input circuit of the amplifier, since its input impedance is very high (over 10 megohms).

2) Frequency Range. The frequency range of the 0BC type oscillograph extends nominally from 1 to 60 cycles. The frequency range of the 0BD oscillograph extends from 1 to 120 cycles.

3) Resonant Frequency. The resonant frequency depends on the type of pickup

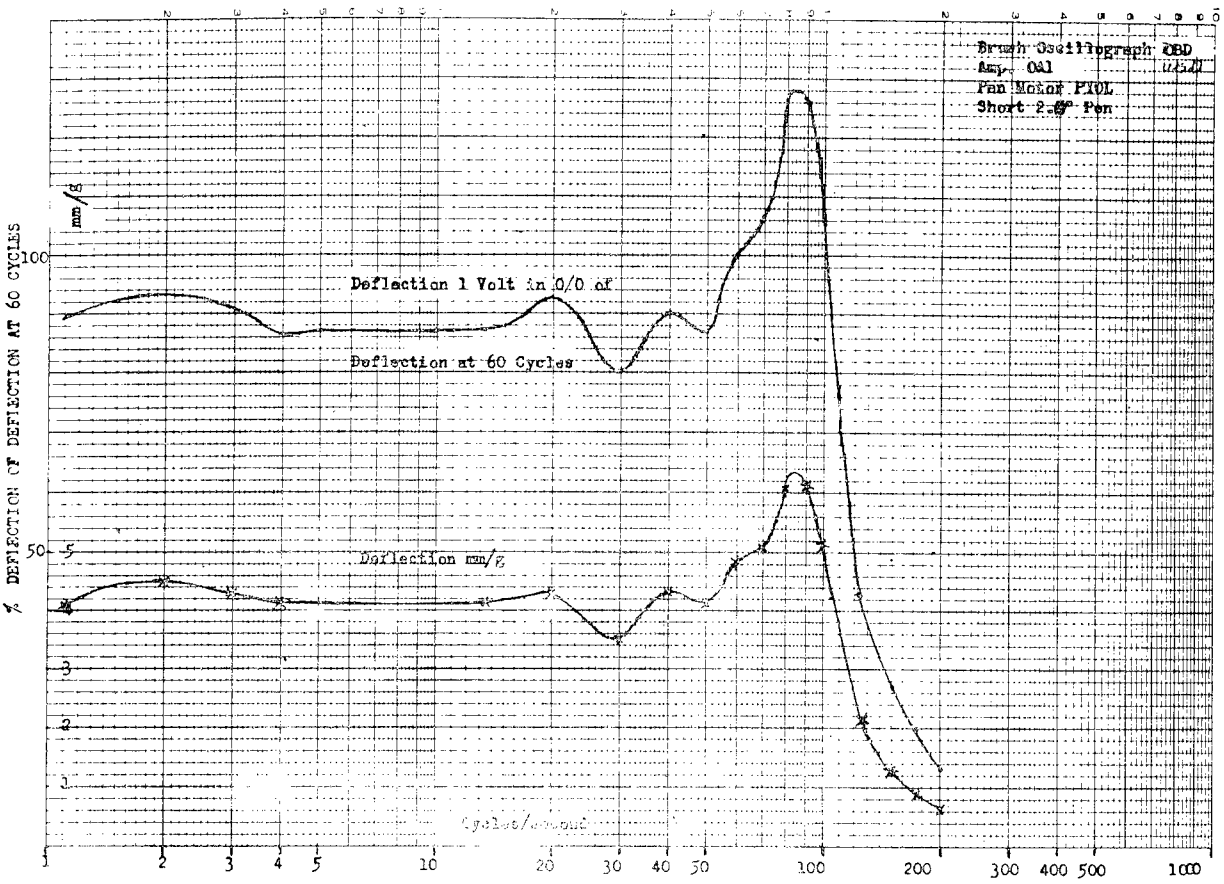

Figure 10

used. However, the [illegible] oscillograph has a definite peak at 85 cycles, and the OBC type of oscillograph has slight rise in characteristic at 30 cycles.

4) Amplitude Range. The amplitude range depends on the sensitivity of the pickup used. The voltage range which can be recorded extends from .001 V to 700 V.

5) Outstanding Characteristics, Advantages, and Disadvantages.

The following features made the Brush vibration recording system highly [illegible] table for exploratory or preliminary investigations:

a) The record is instantly available after it has been taken; no additional processing of the record is [illegible].

b) The system is very simple to operate. No special skill is required.

c) The speed with which the paper moves through the oscillograph is constant. Three speeds are available; 125, 25, and 5 mm/sec.

The following disadvantages restrict the application:

a) The system is not very easy to calibrate.

b) The power consumption during operation is quite high.

c) The frequency range is quite limited.

d) The Rochelle salt crystal and the Pyrex pen are very fragile, easily damaged, and are affected by temperature variations.

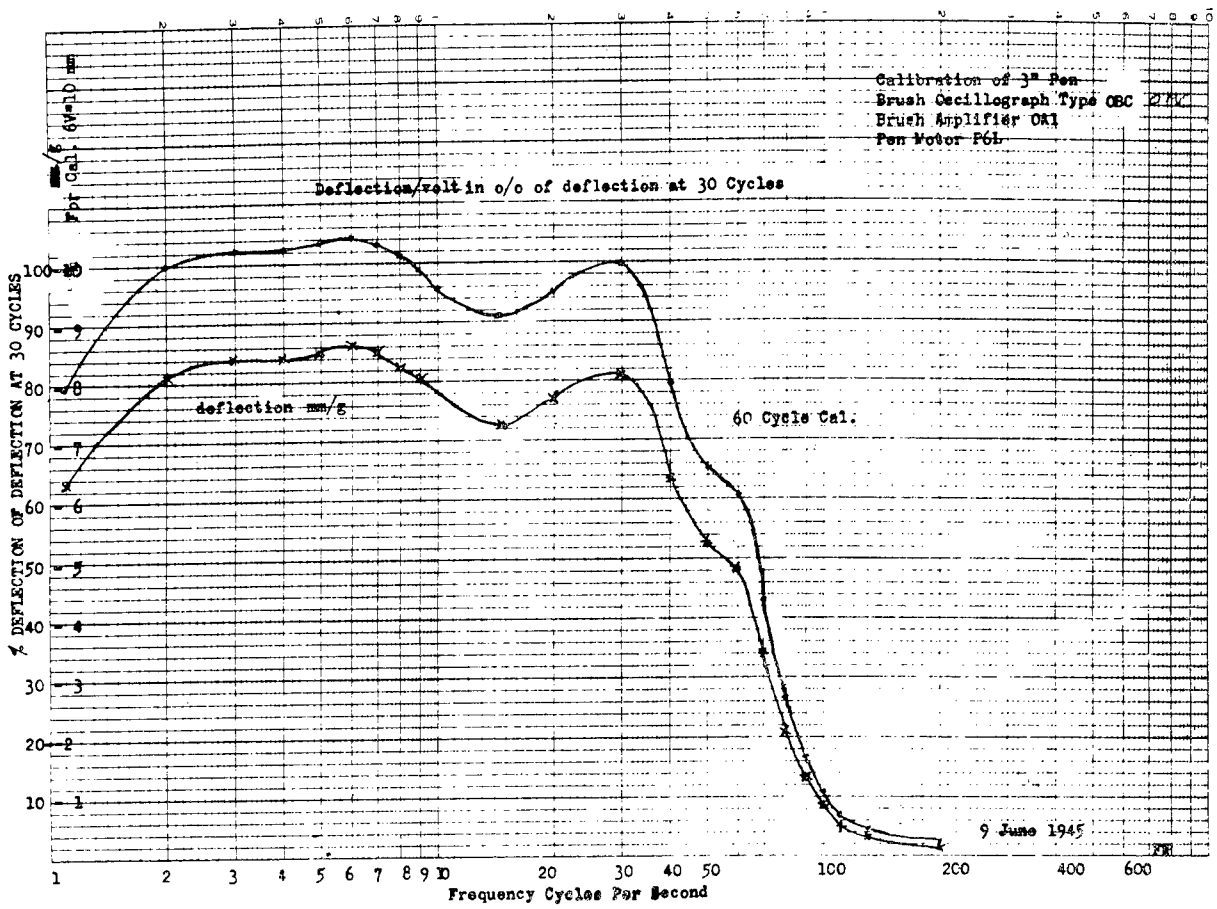

Figure 11

The Brush Magnetic Penmotor BL902

Some of the disadvantages of the Brush Crystal Penmotor are eliminated by the Magnetic Penmotor.

This oscillograph element has the following characteristics:

a) Frequency range 0 to 70 cycles/sec.
b) Sensitivity 1.1 mm/volt, 1.6 mm/ma.
c) Impedance 1500 ohms resistance.
d) Damping 250 ohms critical - 500 ohms satisfactory.
e) Amplitude 23 mm max. 0-70 cycles.
f) Pen length 3 inches.

The impedance is much lower than the impedance of the Rochelle salt motor; therefore, the magnetic device is not subject as much to electrostatic pickup as the Rochelle salt motor. The construction is much more rigid, and the damage done to it, due to accidental electrical overloads, can be neglected.

This device comes with an assembly of six pen motors so that at least five simultaneous records can be made.

The Glenn L. Martin Dynamic Measurement Equipment.

1) Description and Theory of Operation

The equipment consists of the following basic components:

a) A carrier signal source (oscillator) operating at about 1500 cycles per second.
b) A modulating bridge circuit.

c) An amplifier.
d) A full wave rectifier (rectifying the output of bridge circuit).
e) A galvanometer type recording oscillograph.
f) A camera containing the recording film.

The system is a carrier system. The frequency amplified remains constant, while the amplitude of the signal fed from the modulating bridge to the amplifier (the carrier) changes with the instantaneous value of the vibration. The envelope going through all the peaks of the carrier is proportional to the vibration acceleration. The carrier frequency is amplified, and the output of the amplifier is rectified and fed into a recording galvanometer. If the characteristic of the galvanometer is such that it does not respond to the carrier frequency, no filter has to be used for filtering out the carrier signal. This can be achieved in almost every case; since the system is limited in its usability up to only one-fourth of the carrier frequency.

All the electric power used for the operation of this system is furnished from a 6V storage battery. The voltage is converted to the proper AC value by means of a rotary converter contained in the DC power supply.

This system is not limited to the use of reluctance variation pickups. Vibration pickups of other types and constructions can be used, such as displacement indicators, strain gauge circuits, and other gauges that are suited for carrier operation.

2) Frequency Range. The frequency range extends from 0 to 400 cycles per second, depending on the galvanometer and vibration pickup used.

3) Resonant Frequency. The resonant frequency depends on the type of galvanometer and the vibration pickup used.

4) Amplitude Range. The amplitude range depends on the particular pickup used. The G. L. Martin reluctance variation accelerometers can be used as high as 12 g in acceleration.

5) Outstanding Characteristics, Advantages and Disadvantages.

The following characteristics are in favor of the G. L. Martin dynamic measuring system:

a) The frequency range extends from 0 to 1/4 of the carrier frequency. This makes the instrument very suitable for low-frequency measurements and easy to calibrate.
b) Independence of the system from the frequency characteristic of the amplifier.
c) Adequate control of the amplification of the system. The amplifier can be adjusted to any desired value.
d) Versatility of the system. Various types of pickup devices may be used. The following have been successfully used at the Squier Signal Laboratory:

G. L. Martin Reluctance Variation Accelerometer
Statham Strain Gauge Accelerometer
Modified Shrader type Accelerometer
Tuning Fork Accelerometers
Displacement Indicators

e) The input impedance of the system is very low (500 ohms); therefore, the danger of interference due to magnetic or electrostatic fields of erroneous voltages is minimized.
f) As many channels may be arranged for as are desired or as galvanometers are available in the oscillograph. It is possible to keep an accurate record of the correct operation of the system by using "dummy pickups" and timing oscillators.

The factors listed below limit the use of the Glenn L. Martin dynamic recording system in certain instances:

a) The great bulk and weight of the galvanometer oscillograph, power supply amplifiers and associated equipment.
b) The sensitivity of the galvanometer and vacuum tubes to mechanical vibrations.
c) Upper limit of the frequency range is given by the carrier frequency and the sensitivity of the galvanometers used.

For further information see Ref. 15 and 16.

Calibration of Vibration Pickups

The calibration of a vibration pickup has to be made on a device which permits the quantity to be calibrated to be measured accurately by some other basic means, or permits the quantity for which the pickup has to be calibrated to be held constant over a wide range of frequencies.

Displacement pickups are generally calibrated on any vibration table for frequency ranges from 10 up to 60 cycles. If this is done, the following has to be kept in mind:

The amplitude of a vibrating table varies with frequency.
If the vibrating table is large, the amplitude varies across the table with the position of the pickup.

In order to account for these difficulties, the following expedients are used:

The table is built as light as possible.

In order to obtain absolute measurements of amplitude, a Gaertner Scientific Company travelling microscope, with filar micrometer and scale, is used for optical measurements.

1) Standard Procedures for Calibrating Vibration Pickups.

The procedure of calibrating is very simple, provided a vibration machine is available which has a large flywheel. The amplitude of the machine is determined as a function of frequency. Next, the pickup is mounted on the machine and a recording device connected, and the operation is started. Then the machine is brought up to speed - for instance, 40 cycles per second - and the power shut off on the vibration machine motor. As the vibration machine coasts to a stop, the machine will pass through all frequencies - from, for instance, 40 cycles to 0 cycles - and enough information can be obtained to plot frequency against displacement.

This procedure is possible only for displacement pickups, since all vibration machines have a vibration wave shape that is rich in high-frequency components. If accelerometers are used and if these accelerometers have a natural frequency inside the range of these high-frequency harmonics, then the higher frequencies will always be accentuated. The natural frequency of the accelerometer will also always be shock excited and predominant. (See Ref. 11)

2) Precision Method of Calibrating Acceleration and Velocity Pickups.

In order to obtain a calibration for a frequency range as large as possible, it is necessary to have a sinusoidal driving force of as pure a wave form as possible. Devices with such characteristic, designed on the basis of the principle of operation of the dynamic loud-speaker, are available from several manufacturers. For measurement purposes, it is desirable to have the natural frequency of the spring suspension of the moving coil outside the range of the measurement.

3) Precision Calibration Method for Pickup used by General Radio.

Instrumentation. The instruments used were as follows:

a) Audio Oscillator, Gen. Radio Corp. Type 713B, Serial No. 623
b) "D.C." Power Supply, Gen. Radio Corp. Type 672A, Serial No. 101
c) Sound Level Meter, Gen. Radio Corp. Type 759B, Serial No. 718
d) Attenuation Box, Gen. Radio Corp. Type 249T, Serial No. 428
e) Vacuum Tube Voltmeter, Gen. Radio Corp. Type 727A
f) Primary Frequency Std., Gen. Radio Corp. Type C21HLD
g) *Current Meter, Weston Thermal A. C.
h) *Oscillograph, Three Inch R.C.A.
i) Constant acceleration vibration jig (as designed by General Radio Corp.). This item is under development and is not a commercially available product. All the above instruments, except those marked with an asterisk are within 2% accuracy. Because of their function, those instruments so marked do not necessarily have to have this accuracy. The primary standard has a possible error of one part in two million.

The constant acceleration vibration jig is a device resembling a loud-speaker and consists of a [illegible] field magnet and a coil to which a sinusoidal current is supplied. The moving coil assumes sinusoidal motion in the field of the magnet. Since the position and size of the coil is such that it is always in a strong, constant and uniform field, if the current through the coil is a sinusoidal current, it will move in a sinusoidal motion at right angles to the field of the strong constant magnet.

The magnetic field strength of any coil carrying current is directly proportional to the current. The force acting on the current-carrying coil is directly proportional to its magnetic field strength. Since by Newton's Laws force equals mass times acceleration, it can be shown that if the RMS value of the current is held constant with changing frequency, the field strength, hence the force, hence the peak value of acceleration of the moving coil, will be constant. Taking precautions to keep the current to the coils constant, we thus have a constant acceleration vibration jig.

Analysis of Records Obtained.

All records obtained with vibration recording instruments are quite complex. The recorded wave generally contains many frequencies. Strictly speaking, a harmonic analysis should be made, and the individual frequency components should be determined. This method is actually used. (See Ref. 3 and 4)

The instrument used for such automatic recording is the Erpi recording frequency analyzer with a frequency range from 10 to 5000 cycles.

A similar instrument that can be used for such applications is the General Radio 762-A vibration analyzer (See Ref. 5).

However, in every case a fair approximation can be made, and the frequencies and amplitudes can be determined if some simple rules are observed.

In most cases, the frequencies of the vibration can be obtained with the timing wave or by calculation from the tape speed, which is assumed to be constant.

Figure 12d shows the combination of two harmonic frequencies of a frequency ratio 3:1.

Figure 12e shows the combination of two vibrations of frequency ratio 2; the amplitude of the 30-cycle vibration is about twice the amplitude of the 15-cycle vibration. This record is a typical example to indicate the difficulties encountered in the analysis when the two frequencies are not widely separated.

If the frequency ratio is less than 2 or 3, then beat frequencies are formed. Figure 12f illustrates such a case. The

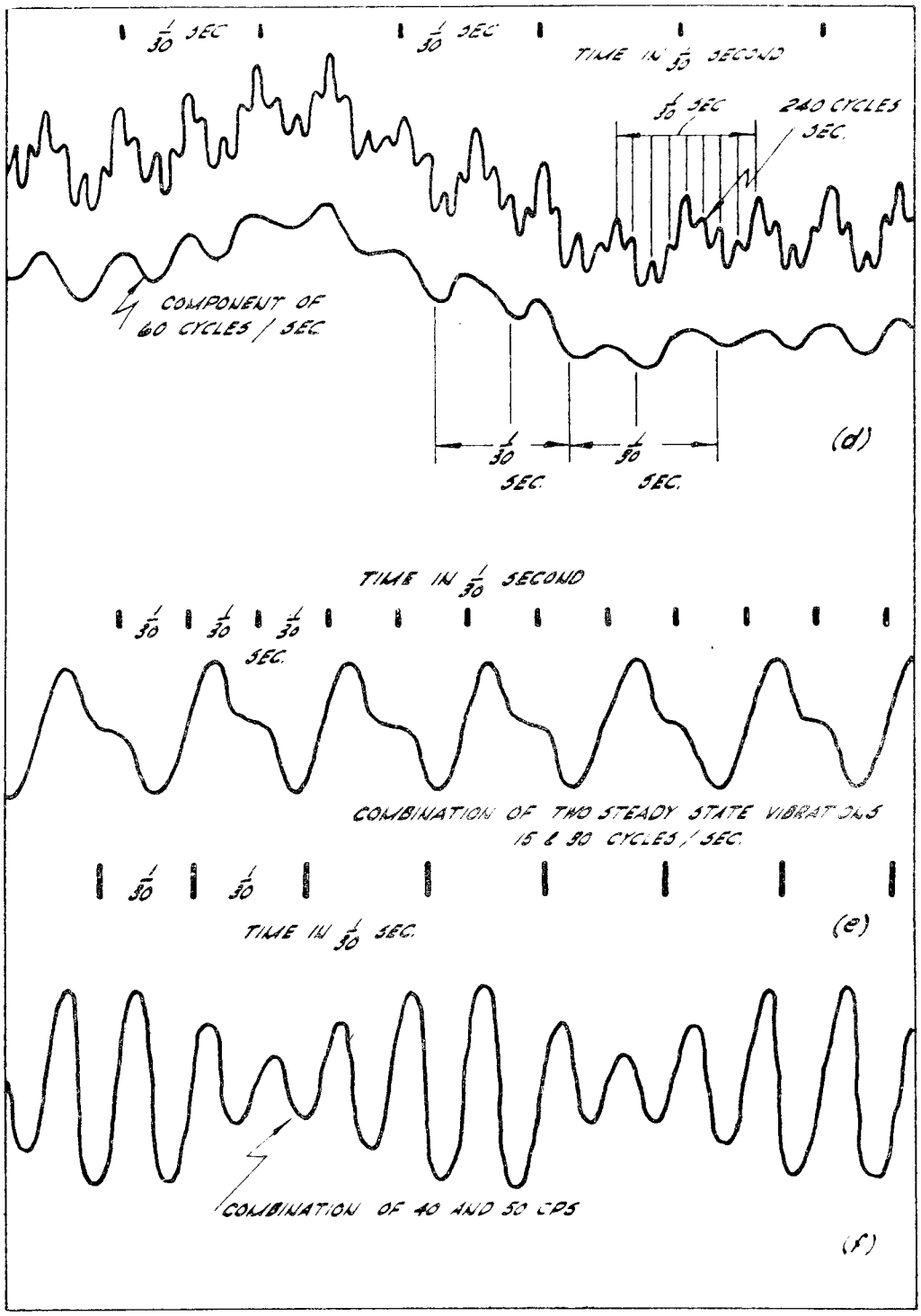

Figure 12 Vibration Analysis

beat frequency is produced by two frequencies 40 and 50 cps. The frequency of the beat effect is 10 cps. The amplitude fluctuates between the difference and the sum of the amplitudes of the frequency components.

The General Radio Type 761A Vibration Meter.

1) Description and Theory of Operation.

The General Radio Type 761A Vibration Meter (Figure 13) is essentially a vacuum tube volt meter that indicates the RMS value of the vibration under measurement in combination with a series of integrating circuits. No indication of the frequency of vibration is made. The detailed theory of the instrument is given in Ref. 5, 7, and 13. The instrument makes it possible to read either displacement, velocity, or acceleration. A Rochelle salt acceleration pickup is used as vibration sensitive element. Two integrating circuits are built in the amplifier and can be connected into the circuit by means of push-button switches. An attenuator system is push-button operated. It is designed to permit measurements of vibration over the following ranges:

Acceleration may be measured from 1 to 30,000 inches/sec.^{2}

Velocities can be measured from .001 to 300 inches/sec. (depending on the frequency).

Displacements can be measured from .001 to 30 inches (depending on the frequency). If the wave form of the vibration is sinusoidal, then the frequency of the vibration can be determined in the following manner:

Let X be the displacement measurement and V be the velocity reading made with the vibration meter

$$\text{then } V = 2\pi F X$$

$$\text{or } F = \frac{V}{2\pi X}$$

If an exact frequency analysis is to be made, a General Radio Type 762-A vibration analyzer should be used. With this device, a frequency spectrum can be obtained.

2) Frequency Range. Linear from 2 to 100 cycles.

3) Resonant Frequency. Approx. 1500 cycles for Rochelle Salt Crystal.

4) Amplitude Range. Displacement .001 to 30 inches (depending on the frequency). Velocity .001 to 300 inches/sec. (depending on the frequency). Acceleration 1 to 3000 inches/sec.^{2}.

5) Outstanding Characteristics, Advantages, and Disadvantages.

The instrument is self-contained (all batteries and power supplies are contained in the case of the instrument). A calibration which is accurate within a few per cent can be made by attaching the instrument to a 60-cycle source and adjusting the input voltage to a predetermined value.

Field Tests made by SCEL

The first field tests on vehicles were made on roads in the vicinity of Fort Monmouth. Secondary roads were selected which had either a large number of washboard formations or single road waves. However, the conditions of the secondary roads would be constantly changing, so that no comparative repetitive tests could be made. Therefore, it was decided to use the gun carriage test course of the Munson Test Course in the Aberdeen Proving Ground for any other additional tests.

The performance of a vehicle on this course is best illustrated by a high-speed motion picture film taken by the Automotive Research and Development Center entitled "40 mm Gun Mount M5, test of suspension for" designated APG809.

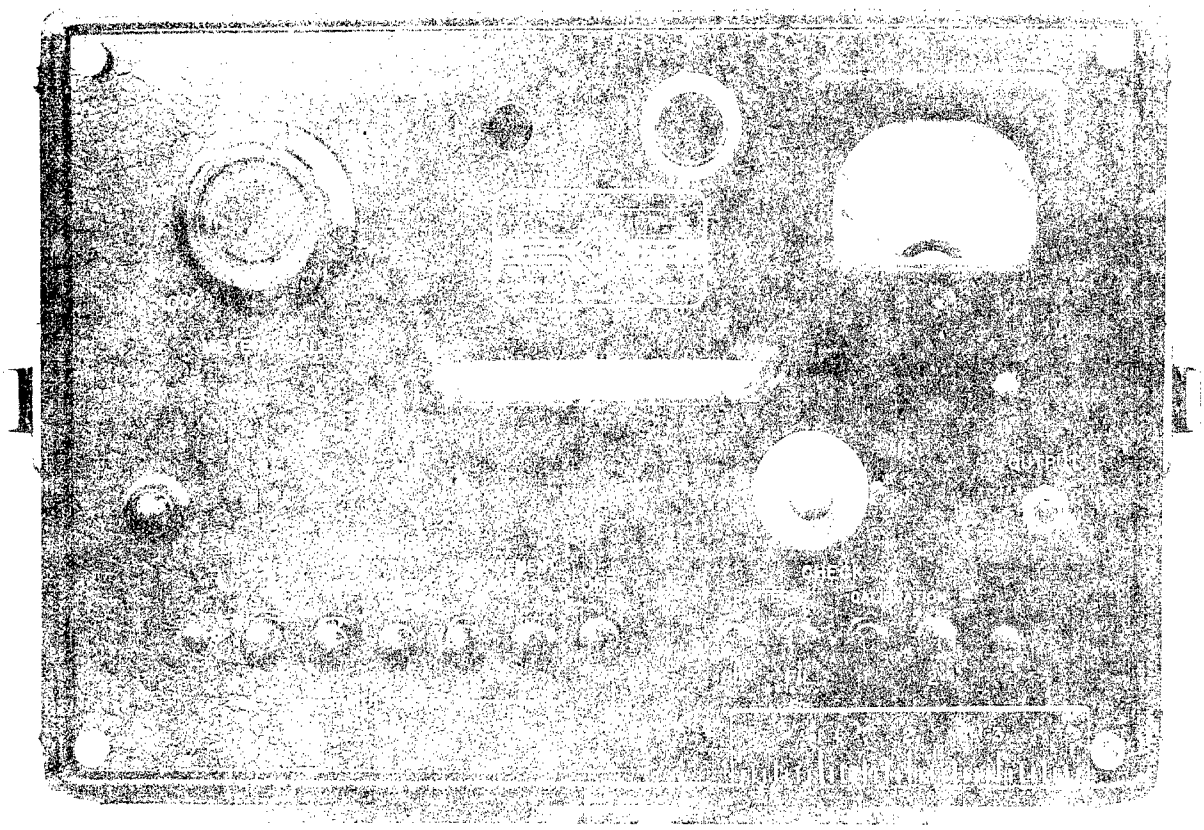

Figure 13 G. R. Vibration Meter. Part of G. R. Vibration Test Set

Description of Test Roads

1) Coarse Washboard or coarse corrugation road.

This is a concrete road with sinusoidal waves spaced about 6 feet apart.

2) Belgian Block Road. This road is constructed out of cobblestones (Belgian blocks). The blocks are not laid uniformly but in waves. These waves are 2 inches high and 8 feet crest to crest. The waves are so designed that they exist not only along the direction of travel but, also, perpendicular to it.

3) Radial Washboard Road. This road consists of diagonal washboards 1 to 2 inches high and separated 1 to 2 feet. This road introduces tramping action of the suspension.

4) Individual Corrugation. This is a smooth concrete road with transverse sinusoidal waves 3 inches high spaced 30 feet apart.

5) In addition to the four roads mentioned above, the Perryman cross country test course was used on one vehicle for overall comparison. This road is hilly and purposely maintained at its worst condition.

It may be stated that travel on these roads is so severe that a thousand miles of driving over them represents approximately 100 thousand miles of normal driving. Listed below are the normal speeds at which a vehicle may travel across these roads:

Coarse washboard road	5 mph
Belgian block road	20 mph
Radial corrugations	15 mph
Single corrugations	20 mph

Speeds higher than those indicated above result in destruction of the vehicle. This serves as an indication of the severity of the roads.

Machines Simulating Rough Rides.

On the basis of the measurements made on these roads, machines simulating the vibrations and shocks encountered have been built by various investigators. These machines are listed below:

The Canadian Indicating Instruments Testing Machine.
The SAE Automobile Headlight Testing Machine.
The Chrysler Headlight Testing Machine.
The General Motors Shake Rig.
The General Motors Bounce Rig.
The SCEL Bump Testing Machine.
The SCEL Tumbling Machine.

1) The Canadian Indicating Instrument Testing Machine.

This machine is excellent for production testing. It consists of a hinged board with a 1 inch square cam which is located 15 inches from the hinges under the board and is rotated at a speed of 120 RPM. In such way, repeated shocks are produced at the rate of 480 CPM.

The following comments are made concerning this machine:

a) The drop height is not high enough.
b) The impulse time is too short.
c) The repetition rate is too fast.
d) Results cannot always be duplicated with this machine.

2) The SAE Headlight Testing Machine.

This machine is shown in Figure 14. It has a cam actuating a springloaded cantilever which carries equipment to be tested. This machine was used for testing automotive suppressor equipment during the war. Up to this time, no performance records have been taken.

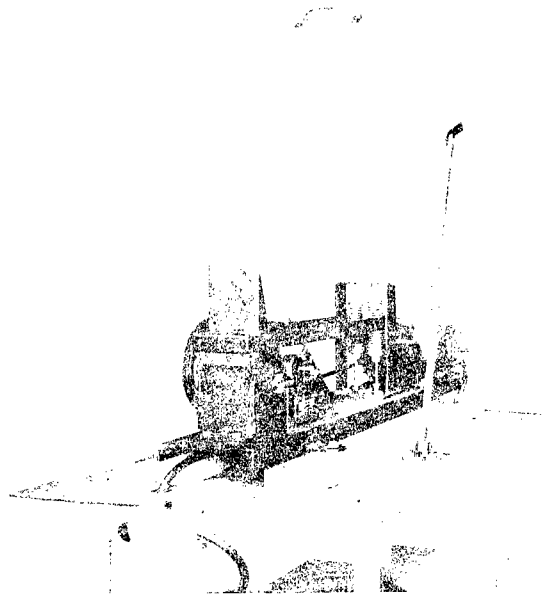

Figure 14 Vibration Machine, S.A.E. Design and Specifications

3) The Chrysler Headlight Testing Machine (Ref. 4)

This machine is a modification of the SAE machine.

4) The General Motors Shake Rig.

This machine is described in Ref. 34. The machine consists of a shaft, close to the ground, which is rotated at speeds up to 1600 RPM. At its end are links which connect to some point on the suspension of a truck system. This fixture pumps the wheels up and down with a fixed amplitude and at any desired frequency up to 1600 c.p.m.

5) The General Motors Bounce Rig.

This device is described in Ref. 22. It consists of a large rotating drum which supports the wheels of the vehicle and which has mounted on its circumference

wave-like cams which bounce the wheels up and down. Maximum amplitudes can be obtained at the resonant frequencies of the vehicle. This is an excellent device for studying the performance of the vehicle; however, the space requirements for such installation are quite large.

6) The SCEL Bump Testing Machine.

This machine is similar to the machine described under (4) but differs from it, in that the release of the truck suspension system after it has been pulled down by a cam is very sudden. This produces a sudden snap action which bounces objects placed in the cargo compartment of the vehicle up into the air, resulting in repeated impacts which are destructive to equipment stored in the compartment.

7) The SCEL Tumbling Machine.

The tumbling machine simulates transportation shocks encountered during transportation, when the objects are carried loose in the vehicle. Figure 15 shows the

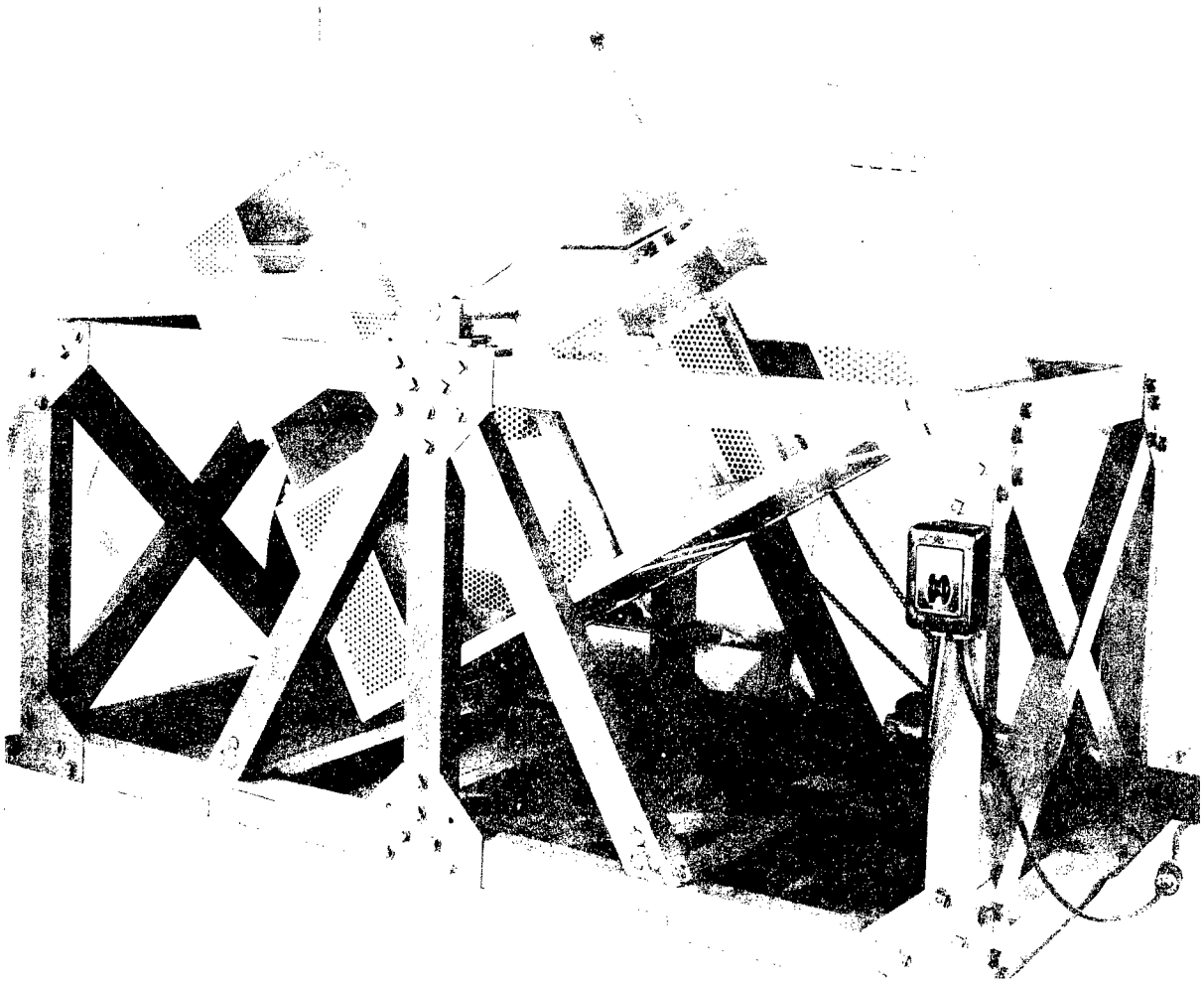

Figure 15 Tumbling Machine

construction of this machine. The machine is designed so that objects tumbled in it fall approximately the same distance as they would be thrown in the air if they were transported in a vehicle. For detailed information see Ref. 36 and 37.

CONCLUSIONS AND FUTURE WORK

Additional work is in progress on the following subjects:

Characteristics of various electrical and electronic accelerometers.

Methods of obtaining the frequency amplitude characteristic of various elastic elements as a function of the natural frequency of these elements.

Description of methods and procedures used for testing shock mounts.

A description of the operation of mechanical and electrical analogues.

Methods used for the graphical and numerical solution of differential equations.

The results of measurements made with the instruments described have been reported in test reports Nos. 1 to 7. See Ref. 27 to 33.

Upon the basis of these results, machines have been made available which will simulate road conditions encountered during transportation. In the future, tests will be made with some of these machines to determine the shock resistance of various components and the equivalent of field and laboratory tests (number of hours on laboratory test equivalent to number of miles of transportation over rough terrain).

Design and application principles for components and subassemblies will be established.

RECOMMENDATIONS

It is recommended that after completion of the basic research work being conducted now in various laboratories, a standardization program for shock and vibration testing be initiated for the purpose of simplifying the various shock testing procedures now in use for performance testing.

It is recommended that a unified production testing method be established which simulates transportation.

ACKNOWLEDGEMENT

The assistance rendered by the various commercial companies listed in this report, in supplying information, is greatly appreciated.

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SOME TESTS ON MILITARY LAND VEHICLES

By

C. D. Montgomery

Aberdeen Proving Ground

Shock and vibration of military automotive vehicles are grouped into three categories: those occurring during operation; those occurring when vehicle weapons are fired; and those occurring under impact of projectile hits. Each category is briefly discussed, with examples of the test work accomplished. Particular emphasis is placed on these problems as related to track-laying combat vehicles.

INTRODUCTION

During the war, the analytical approach to a problem was often subordinated to the need for immediate answers. Certain specific shock and vibration data were obtained to meet the requirements of the Signal Corps, but, in general, it was quicker to manufacture two comparable items for test in order to evaluate design changes. This was a quick but expensive way to make improvements which, however, did not produce quantitative results for post-war designers. Much of the military vehicle test work was based on the judgment of experienced engineering personnel. It was a wise decision to obtain speedy test results in this manner; however, at present we are only beginning to investigate thoroughly the shock and vibration characteristics of all standard military vehicles.

The general shock and vibration problems for military vehicles may be grouped, for test purposes, as:

1. Those occurring during vehicle operation.
2. Those occurring when vehicle weapons are fired.
3. Those occurring under impact of projectile hits.

For this discussion, shock will be considered as a severe transient vibration condition. Ride vibration is fairly regularly repeated vehicle motion, dependent upon the vehicle mass and suspension system, in conjunction with operating and road, or terrain characteristics (see test areas, Figures 1 and 2). Ride vibration may be thought of as the vibration limits within which the motion is not disquieting to a person riding in a vehicle. The frequency range of 50 to 150 cycles per minute of moderate accelerations, with small displacements, is not unpleasant. Unfortunately, other factors such as tire vibrations and spring bottoming cause ride vibrations which are not in the range of comfort.

The engine, transmission, differential, and wheels are all exciters of vehicle vibrations. In addition, there are the shocks and vibrations imposed by the motion of the truck over the road or terrain. At given speeds, the engine and power train components will vibrate at known values which can be determined and attenuated by rubber mountings for

Figure 1 Munson Automotive Test Area (Aberdeen Proving Ground)

the engine, interphones, instrument panels, etc.

In general, except where road irregularities occur in resonance with the vehicle suspension, road impulses are attenuated by the mainsprings and shock absorbers. Thus, when a truck is driven against a log (Figure 3), the mainspring will take the impulses; however, the shock imposed will cause the truck and its components to vibrate severely. It has been found that engine vibrations of a stopped truck do not correspond with the predominant vibrations recorded on an operating vehicle. The predominant engine vibrations, when underway, are affected by transmission and other power train harmonic vibrations, and road, or terrain, irregularities. Large accelerations or rate-changes of acceleration occur during cross-country operation, causing the destruction of vehicle parts. These large accelerations occur after the vehicle springs bottom against the spring stops. Bottoming is a more general occurrence during cross-country operation than during road operation.

During a co-operative Signal Corps and Ordnance project, it was determined that the maximum instantaneous vertical acceleration of a wheeled vehicle is dependent upon the forward speed of the vehicle, that the frequencies of vibration are almost independent of road profile, and that the effect of vehicle speed is small except for high speeds or resonant conditions. It was determined, also, that the vertical accelerations are two to three times the acceleration (except for high frequencies) through pitch or roll axis based on this test. In general, the low-frequency spring vibrations had the

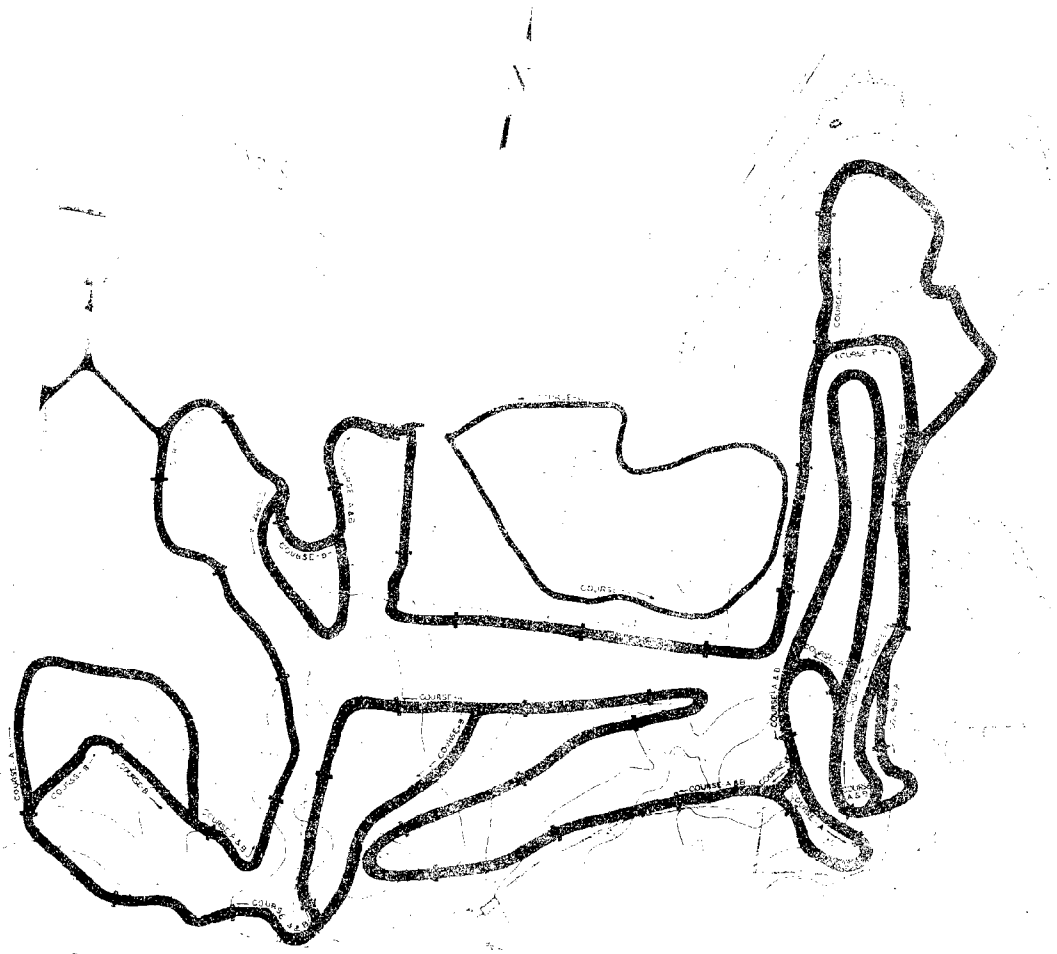

Figure 2 Churchville Automotive Test Area (Aberdeen Proving Ground)

greatest acceleration. However, in some vehicles the high-frequency structural vibrations were the most severe. It is believed that this is due to shock absorbers and the fact that road impulses excite large-amplitude natural vibrations. The transmission of vibration, theoretically, should be independent of road speed, because the road impulses should be of much higher frequencies than the natural frequency of springing. Results of the co-operative test project reported that the vertical accelerations of all wheeled vehicles tested did not exceed three g when operated on a typical stretch of Belgian Block road and a Single Bump road. It should be noted that *impact shocks* were not always recorded, and vertical accelerations could have occurred during this test.

For a representative group of wheeled vehicles, it was found that the predominant frequencies could be grouped into 2 to 5, 8 to 15, and 60 to 200 cycles per second, which are, respectively, the natural frequencies of the mainsprings, tires, and body structures. The effect of spring

Figure 3 Operation Over Concrete Humps (Aberdeen Proving Ground)

bottoming was not investigated. There were vibrations within the ranges noted which could not be correlated with the natural frequency of any springing system on the truck.

TRACKED-VEHICLE OPERATION

Only one tank (see Figures 4, 5 and 6) was investigated in the aforementioned co-operative project. One of the most serious exciters of tank vibrations is the track blocks when striking the road or terrain.

The track vibration frequency (which predominated at any speed) transmitted to the tank chassis is a function of the road speed and track pitch (distance between blocks).

It was found that rubber tread tracks caused smaller amplitude vertical accelerations than the steel track. Cushioning of the rubber, therefore, reduced the vertical accelerations at a given road speed.

The predominant frequency bands for the tank operation were 3 to 7 cycles per second - spring vibrations, and the groups 12 to 17, 38 to 60, 200 to 300, and 500 to 1000 cycles per second which represented the engine and power train vibrations. During earlier tests, with a dial indicator, maximum double displacement amplitudes on older medium and light tanks were 15 and 7 mils (at 16 and 12 miles per hour). Structural members were found to have double amplitudes of from two to five mils and stowage boxes from five to twenty mils at the predominant track frequency. When the tank was stopped with the engine racing, accelerations were small; however, even when operating, no accelerations were greater than three g.

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Figure 4 Special Test, Churchville Area. Light Tank M24 (T24)

The springs did not bottom during any of this operation.

At present, a study is underway to determine the effect of a light tank hitting a twelve inch log obstacle at speeds up to 15 miles per hour. Accelerations in three planes are being recorded of the center of gravity of the tank as well as at other points on the vehicle. Glenn L. Martin accelerometers are being used, with cables trailed from the test vehicle to a truck containing a Hathaway oscillograph and Glenn L. Martin amplifiers.

During this test and within the limits of the present instrumentation, accelerations as high as five g have been recorded at the top of the gun. This condition is aggravated by looseness in the elevating mechanism. Severe operation of this type also produces impact accelerations as high as four and one-half g at the center of gravity of the tank. In general, other values are less than three g. The spring suspension has been designed to withstand a four g shock load. Our problem at present is to refine the instrumentation to permit more reliable and higher frequency shock determinations; in particular, we are investigating the use of telemetering the acceleration data to a convenient recording point.

Prior to the work now being prosecuted, various mechanical devices were tested, all of which were excited by the initial shock, after which they would vibrate at their natural frequencies. Seismic recording devices are satisfactory for recording low-frequency ride vibrations. Peak reading accelerometers have been used. A considerable number of displacement-time studies have been taken using camera speeds ranging from normal up to 5000 frames per second. Displacements are then measured from the projected film. Although labori-

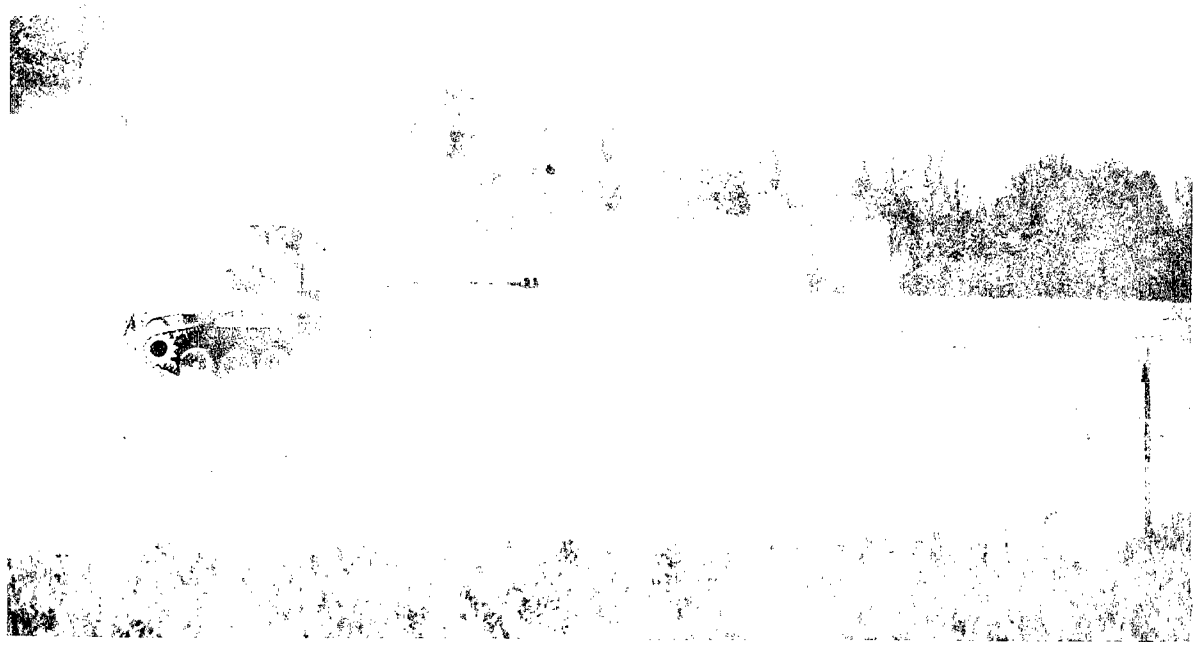

Figure 5 Heavy Tank, M26, Firing of 90 mm Gun M3 Using A.P.C. Ammunition

ous, this is a very reliable method for obtaining data, as well as a graphical illustration of the test.

Strain gauges have been used to measure the shock loads on test items. In general, electronic amplifying and recording equipment must be kept out of tanks.

FIRING SHOCKS

Displacement-time studies have been made of the firing shocks from tanks and gun motor carriages. It is a general practice that the displacements caused by firing shocks be recorded on all new mounts. There is a general trend in tanks for a completely gyrostabilized gun and turret. A knowledge of the firing shocks which occur would facilitate the design of these new fire control items. Frankford Arsenal now has an active test project on the determination of firing test shocks.

IMPACTS

Another group of shock and vibrations associated with military vehicles occur from projectile impacts and land mine blasts. There are two classes of tank equipment which must be protected from projectile impacts or mine blasts:

1. Those items *essential* to the operation of the tank which are needed to permit the tank to perform under its own power in order to return to its base.

2. Stowage items or *nonessential* functioning parts which must not become secondary missiles inside the tank. A secondary missile occurs from objects being propelled inside a vehicle by the force of projectile impacts.

From firing at light and medium tank hulls, it was determined that:

1. Brackets solidly mounted to the armor will become secondary missiles when impacted.

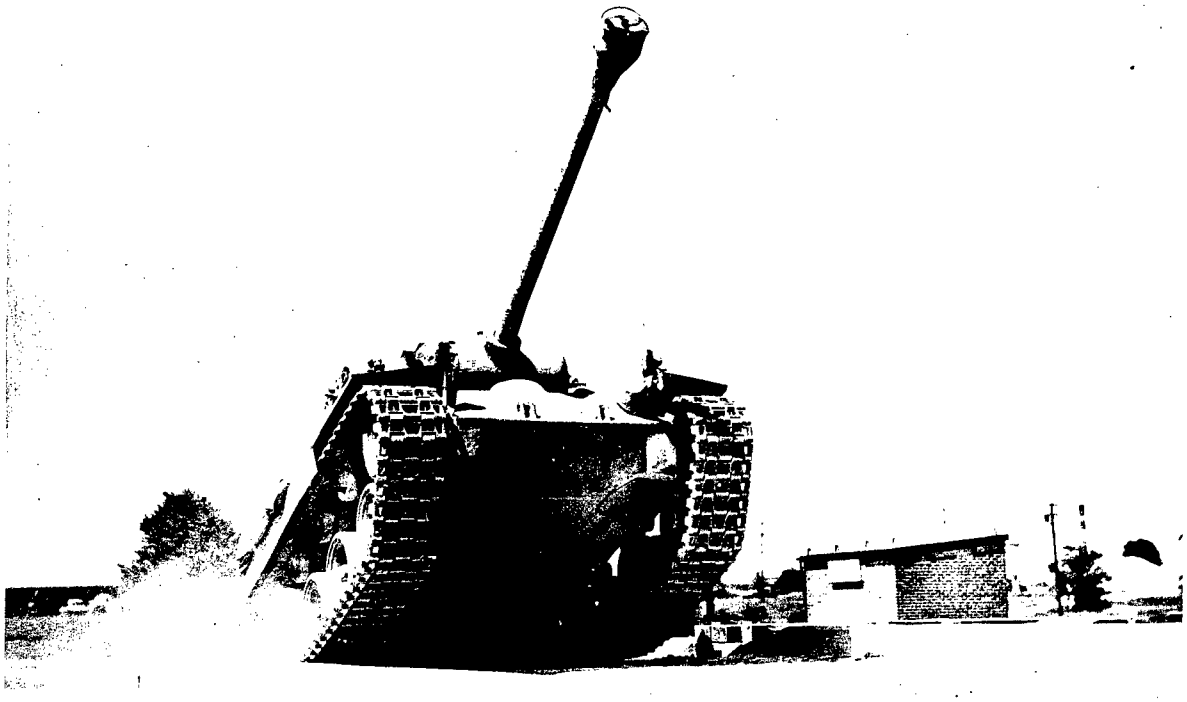

Figure 6 Heavy Tank M26, Operation on 36" and 48" Vertical Walls

2. Approximately 18 inches from the impact area the accelerations will be high, but the displacement will be low, and little damage will be suffered by mechanical equipment. At the point of impact, the high frequency of shock will, in turn, produce high accelerations in the order of 5000 to 8000 g with small displacements (except for impact area), along with low-frequency vibration with large-displacement amplitudes. Much test work was of a qualitative nature, with shocks produced by proof projectiles fired at known velocities.

3. Electrical apparatus mounted in armor plate is sure to be damaged.

4. Stowage and brackets should be strap mounted, spaced at least one and one-half inches from the armor.

5. Nothing should be mounted on the front armor plate, unless mounted in long, flexible straps supported over a considerable distance and allowing movement under an impact. The tank steering mechanism should not be mounted on the front plate.

6. Good welding is better than other fastening means; however, bolts or screws are better than poor welding.

The aforementioned conclusions were reached during a co-operative Ordnance project in which General Electric, Westinghouse, and Dr. Edgerton, of M.I.T., participated.

There are many improvements in tanks and military trucks which are traceable to the meager shock and vibration work accomplished to date. To mention a few - there are the torsion bar suspensions, rubber bushed track, gyrostabilized guns, and shock mountings of all sorts. Where possible, Ordnance has benefited by the shock and vibration work of the automotive industry. This applies particularly to

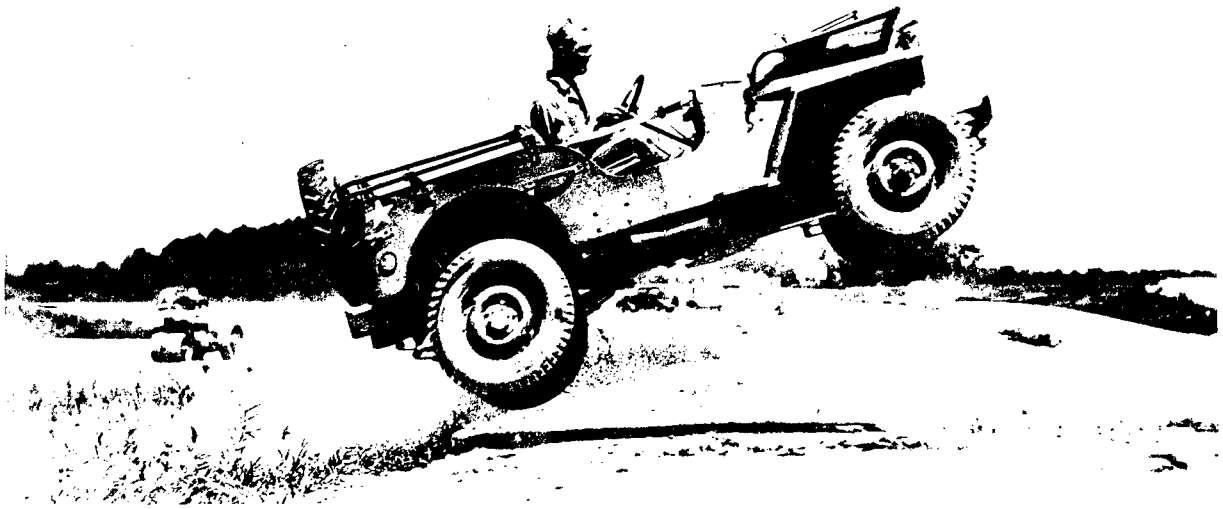

Figure 7 1/4 Ton Truck. Operation in Munson Test Area

the wheeled vehicles. The net gain from shock and vibration studies to date is an indication that more data may lead the way to further advances in military vehicle design. The newer vehicle designs will be capable of higher cross-country speeds (Figure 7), which will impose greater shock loads and vibrations. Although we are going through growing pains with our present instrumentation, we feel that the data obtained will eventually prove of value.

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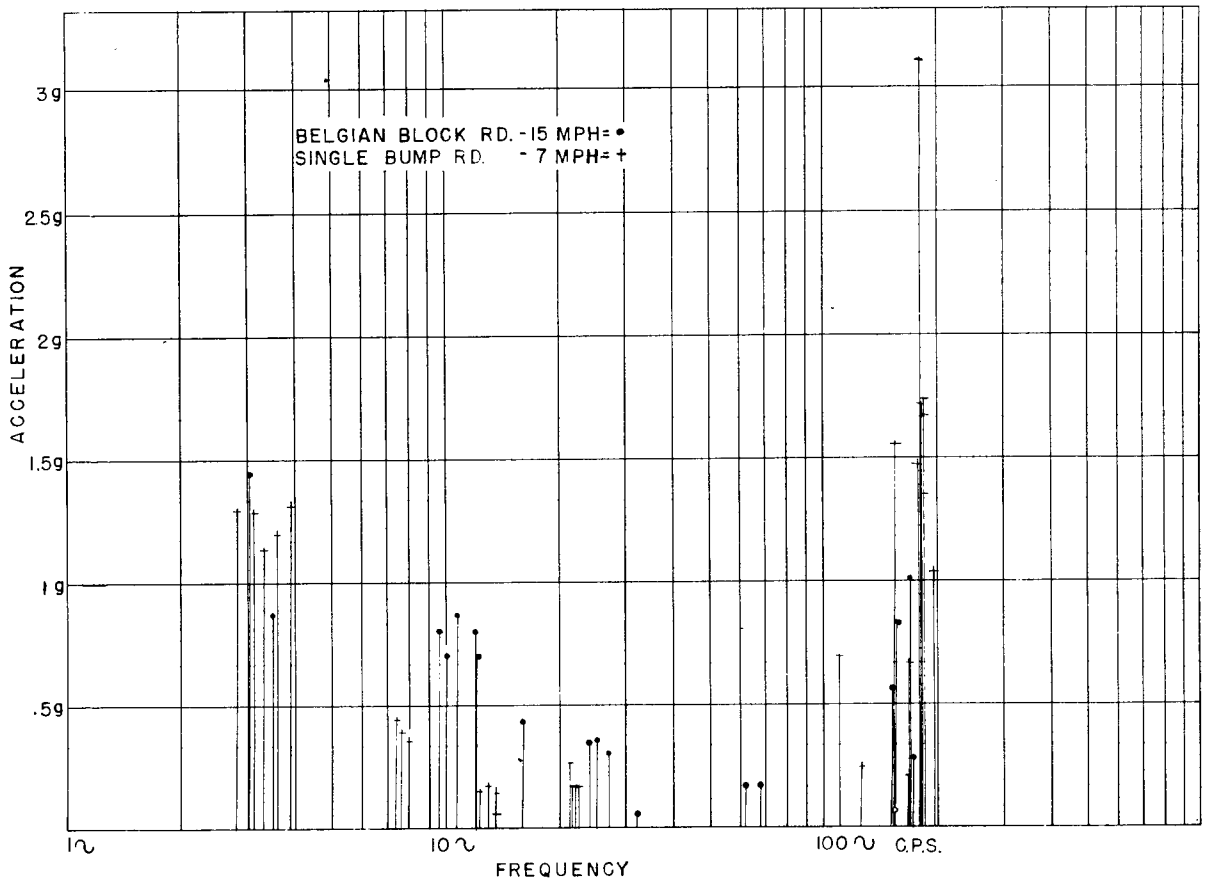
3. *Fire Control Engineering Report No. 1051, "Transmission of Road Impulses to Fire Control Instruments Transported in Military Vehicles."*
4. *Composite film on typical operations in Automotive Division, D&PS prepared by Mr. J. Beach, Project Engineer.*



Figure

lowing generalized results for wheeled vehicles were obtained:

(a) Road impulses are effectively attenuated by the mainsprings and tires, except when road irregularities are in resonance with either spring system. For the case of a single impulse such as is obtained over the "Single Bump" road, the form of the initial impulse is barely transmitted to the chassis, but the free vibration resulting from the impulse is severe. In general then, the effect of road irregularities on wheeled vehicles is to set the chassis vibrating at the natural frequencies attendant, with the spring systems of the vehicle.

(b) Vibrations in the vertical direction occur in three distinct frequency bands: 2-5 cps, 8-15 cps, and 60-200 cps. These predominant frequencies correspond, respectively, to the vertical and rocking modes of the chassis on the main springs, the vibration of the wheels and axles on the tires, and the natural frequency of the elastic structure of the vehicle chassis. The vibrations between 20 and 60 cps were due mainly to motor vibrations, although in some cases body structural vibrations ranged that low. Vibrations other than those due to the motor in this range were practically impossible to correlate with some spring system of the vehicle.

(c) Rocking modes of the vehicles were recorded mainly by the vertical pickups. Because the chassis motion is a mixture of the vertical and rocking vibrational modes, however, it is difficult to discern each mode individually on the records or on the vibration spectra.

(d) The low frequency vibrations, from 2-5 cps in most cases, have the largest associated accelerations. In some vehicles, however, the high frequency structural vibrations are more severe. Such exceptions can be noted by comparison of individual vibration spectra.

(e) Under the test conditions specified, the maximum transient or instantaneous acceleration of the chassis did not exceed 3.5 g for any vehicle tested. This value, however, was obtained when the recording pen struck the "stops", and a value of between 5 and 7 g could be extrapolated.

(f) Vertical vibrations are more severe than those in the crosswise or longitudinal directions. By comparing the vibration spectra of a given vehicle for the three directions, as shown in Figures 2 and 3, it will be seen that the vertical vibrations have accelerations of two to three times those of the vibrations in the other two planes, except for the high frequency vibrations. This statement however, is true only for the vehicles and test conditions reported. Vibrations in the crosswise and longitudinal direction are due mainly to tramp, yawing, and sideshake of the wheels and axles, and at certain critical speeds with certain suspension designs these vibrations can become appreciable.

(g) Effects of Loading - It can be seen that loading lowers slightly the frequencies of vibration in the two ranges, 2-5 cps and 8-15 cps, and decreases the accelerations associated with the 2-5 cps range. Both the accelerations and frequencies of the body structural vibrations are increased, probably because the loads were not secured to the vehicle. From these results, it is evident that the

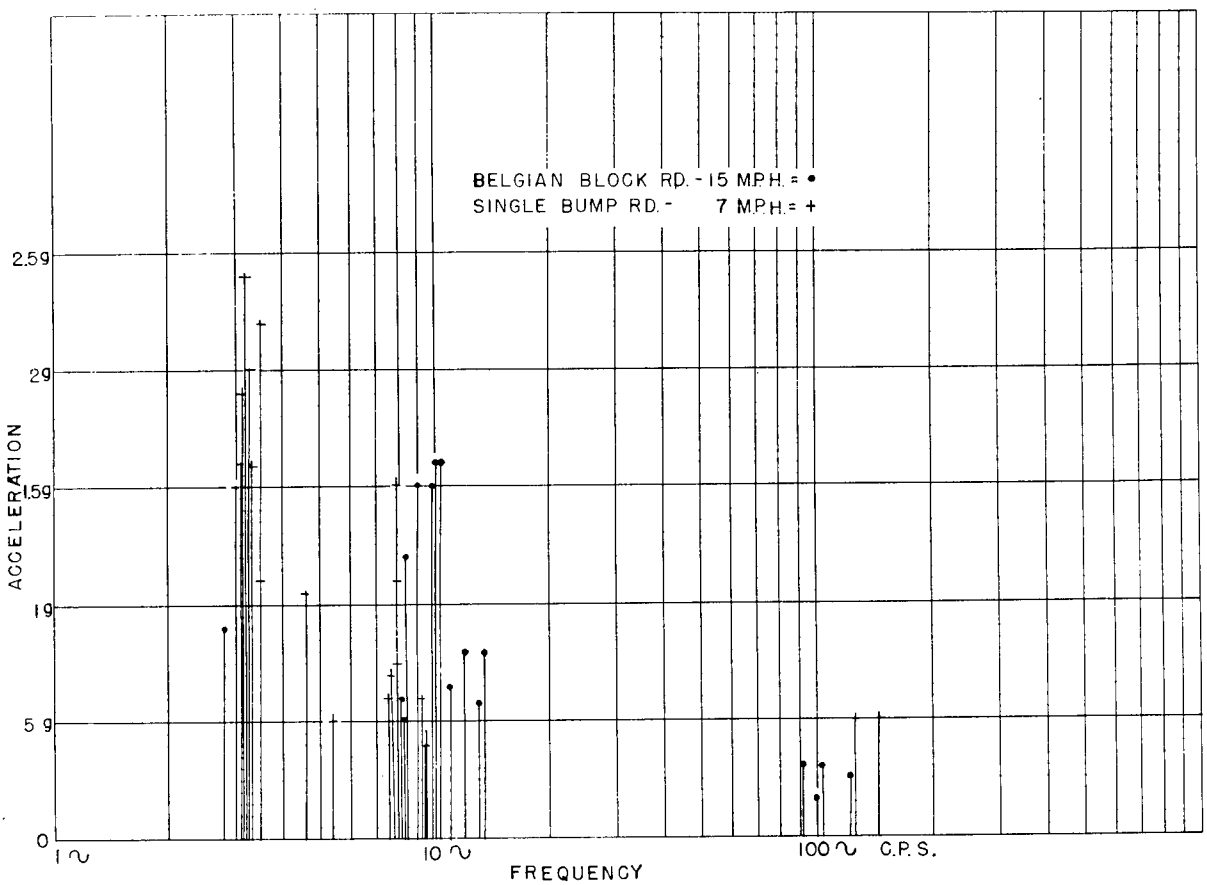


Figure 2

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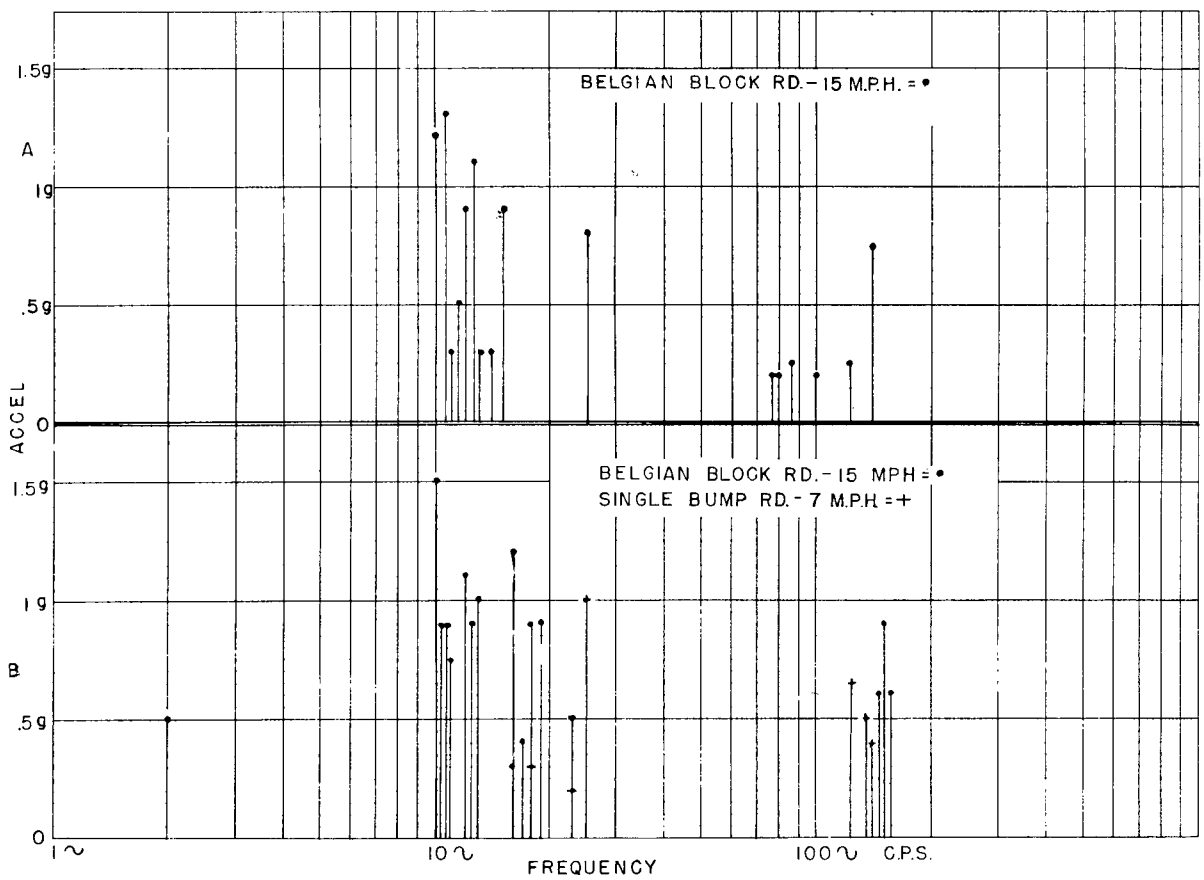

Figure 3

unloaded condition is the worst. It should be remembered, however, that as the load is increased and the springs become effectively softer, the possibilities of "bottoming" the springs, and thereby obtaining impact, become greater.

(h) Since the vibration of the chassis on the main springs causes the major portion of the chassis acceleration, the most advantageous position for instruments in a wheeled vehicle would be at or near the intersection of the rotational axes of the vehicle chassis. For the 2-1/2 ton truck, this position with reference to the cargo space is along the longitudinal center line, as close to the cab as possible. Figure 4 shows the vibration spectrum for another vehicle, a one-ton cargo trailer with no load. The same three principal frequency bands are indicated.

(i) The maximum instantaneous acceleration of the chassis of a wheeled vehicle is dependent upon the forward speed. Figure 5 is a plot of the average maximum instantaneous accelerations, including both positive and negative values, against forward speed for three vehicles. The curves obtained are open to question since they are based on too little data. They are presented, however, for the value of the further investigation they suggest. These curves are an incidental result of the tests actually carried out, since no tests were planned to determine such a relationship. The curves in Figure 5 indicate that an approximate equation relating maximum acceleration, forward speed and the spring stiffness might possibly be obtained. If such a relationship could be established, it would be a great aid in the design of shock-mounted installations.

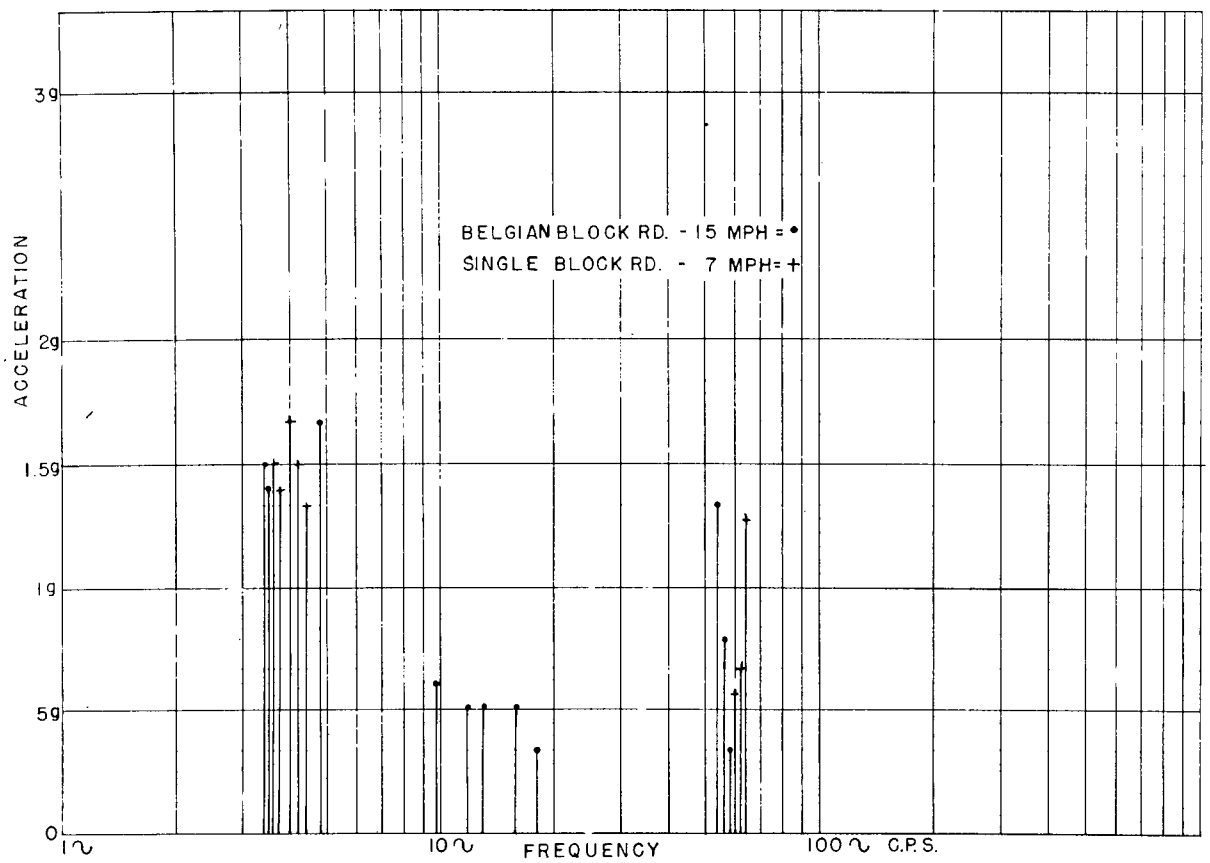

Figure 4

3. Tabular Summary - A tabular summary of the results for wheeled vehicles are presented in Tables I to IV on page 42.

These tables show the maximum accelerations recorded for the predominant frequency bands which can be seen upon inspection of the vibration spectra. The tabular summaries are intended for quick comparison of results on different vehicles or on different parts of one vehicle. For specific information on any vehicle, the corresponding vibration spectrum should be consulted.

RECOMMENDATIONS

The results of this work disclosed that the problem of vibrations and impacts transmitted to fire control instruments in military vehicles is of such magnitude and complexity that a general solution could not possibly be obtained in four months of work represented by this report. It is recommended, therefore, that a project be initiated to carry out a complete research on the subject. The work should include the following phases:

(a) An extensive survey of work already done in this field by various industrial and governmental agencies.

(b) Laboratory investigations to correlate impact and vibration measurements with "strength of materials" theory. This phase of the work would necessarily develop instrumentation and measuring techniques essential for the following phases.

(c) Measurements of the actual impacts occurring in military vehicles due to road irregularities, recoil, and projectile impact.

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(d) Further investigation of vibrations on the basis of the results of this report. Tests should be designed to establish the indicated relationships between acceleration and the variables determining the motions of the vehicles.

(e) The design, development, or purchase of vibration and impact testing machines to fit the requirements dictated by the previous phases.

RECAPITULATION

Accelerations were recorded for each vehicle in three directions-vertical, crosswise and longitudinal. The results of these tests, which did not include spring bottoming, indicate that road irregularities cause wheeled vehicles to vibrate at the natural frequencies of the spring systems of the vehicle. The actual motion was very complex, however, and the predominant frequencies of vibration occurred in the ranges of 2-5 cps, 8-15 cps, and 60-200 cps. These frequency ranges are the natural frequencies respectively of the main springs, the tires, and the body structures. The maximum recorded acceleration was 3.5 g, but the recording pen struck its mechanical stop at that value and a maximum value between 5 and 7 g was extrapolated from the records.

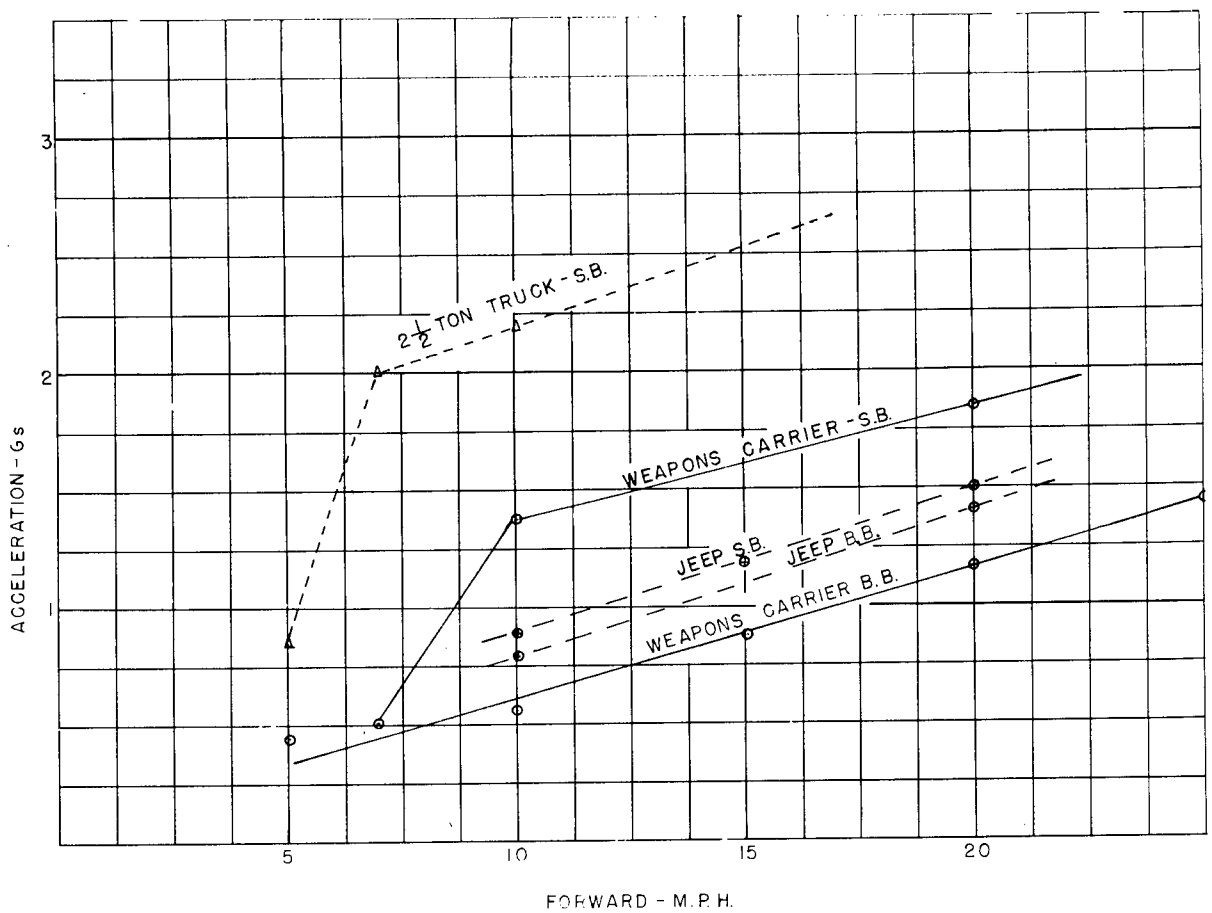

Figure 5

Table I. Truck, 6x6 No Load

POSITION IN VEHICLE	VERTICAL MAX. ACCELERATIONS						MAX. INST. ACCEL.
	LOW		MEDIUM		HIGH		
	FREQ.	G	FREQ.	G	FREQ.	G	
FRONT LEFT	2-4	1.7	3-11	.8	75-100	2.3	1.9
CEN. RIGHT	3-5	3.5	9-15	1.5	50-80		3.5
REAR	3-5	2.4	8-15	1.6	30-150	.3	2.4
	LONGITUDINAL MAX. ACCELERATIONS						
FRONT LEFT	---	--	8-25	1.5	90-110	.3	1.5
REAR	---	--	10-25	1.3	70-140	.75	1.3
	CROSSWISE MAX. ACCELERATIONS						
FRONT LEFT	2	1.1	10-16	1.1	80-110	.1	1.1
REAR	---	--	10-25	1.5	120-150	.9	1.5

Table II. Truck, 6x6 1700# Load

POSITION IN VEHICLE	VERTICAL MAX. ACCELERATIONS						MAX. INST. ACCEL.
	LOW		MEDIUM		HIGH		
	FREQ.	G	FREQ.	G	FREQ.	G	
FRONT LEFT	2-3	1.1	7-18	.7	70-86	.7	1.7
CEN. RIGHT	2-4	1.2	6-40	.7	110-140	.3	1.4
REAR LEFT	2-4	1.5	7-30	.8	110-190	3.1	3.1
	LONGITUDINAL MAX. ACCELERATIONS						
FRONT LEFT	7-8	.8	18-21	.3	75-90	1.4	1.7
CEN. RIGHT	7-12	.4	20-42	.2	110-140	.3	.7
REAR LEFT	---	--	8-36	.4	60-100	.2	.5
	CROSSWISE MAX. ACCELERATIONS						
FRONT LEFT	---	--	10-25	.6	40-130	.2	.95
CEN. RIGHT	1-8	.5	6-20	.4	70-140	.6	.95
REAR LEFT	2-5	1.4	10-22	1.2	60-190	1.1	1.6

Table III.

	WEAPONS CARRIER						
POSITION IN VEHICLE	VERTICAL MAX ACCELERATIONS						MAX INST. ACCEL.
	LOW		MEDIUM		HIGH		
	FREQ.	G	FREQ.	G	FREQ.	G	
	2-3	1.5	5-10	1.7	60-100	2	2
	LONGITUDINAL MAX. ACCELERATIONS						
	---	--	---	--	60-90	.8	.8
	CROSSWISE MAX. ACCELERATIONS						
	3-4	.6	13-15	.6	40-80	1.5	1.5
	CARGO TRAILER-1 TON-NO LOAD						
	VERTICAL MAX ACCELERATIONS						
CENTER	3-5	1.7	10-18	.5	50-70	1.3	2.2
	LONGITUDINAL MAX. ACCELERATIONS						
OF	---	--	8-20	.9	70-110	1.5	1.6
	CROSSWISE MAX. ACCELERATIONS						
FLOOR	2-4	.9	22-32	.6	110-130	.4	1.2
	CARGO TRAILER-1 TON- 800# LD						
	VERTICAL MAX. ACCELERATIONS						
CENTER	2-3	1.0	8-11	1.1	60-110	2.8	2.8
	LONGITUDINAL MAX. ACCELERATIONS						
OF	---	--	20-40	1.3	75-130	.5	1.6
	CROSSWISE MAX. ACCELERATIONS						
FLOOR	---	--	35-42	1.3	65-110	1.7	2.1

Table IV.

	M-14 TRAILER-NO LOAD						
POSITION IN VEHICLE	VERTICAL MAX. ACCELERATIONS						MAX. INST. ACCEL.
	LOW		MEDIUM		HIGH		
	FREQ.	G	FREQ.	G	FREQ.	G	
RIGHT	1.8-4	1.5	14-34	.3	50-70	1.0	1.5
	LONGITUDINAL MAX. ACCELERATIONS						
REAR OF	3.6-20	.7	---	--	110-130	.2	1.2
	CROSSWISE MAX. ACCELERATIONS						
FLOOR	1.8-8	.6	20-70	.2	---	--	.6
	40mm AA GUN CARRIAGE						
	VERTICAL MAX. ACCELERATIONS						
GUN	2-5	1.1	9-30	.3	110-160	.3	1.3
	CROSSWISE MAX. ACCELERATIONS						
CRADLE	1.8-4	1.0	10-30	.4	---	--	1.0
	VERTICAL MAX. ACCELERATIONS						
SIGHT-	2-3	1.7	18-33	.7	80-110	1.1	1.7
	LONGITUDINAL MAX. ACCELERATIONS						
BAR	3-4	1.0	---	--	30-40	1.5	1.5
	VERTICAL MAX. ACCELERATIONS						
TELE-	2-4	1.8	27-36	2	90-110	1.7	1.7
	CROSSWISE MAX. ACCELERATIONS						
SCOPE	1.8-4	1.2	20-25	1.8	85-100	.7	1.8

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DISCUSSION OF THE THREE PRECEDING PAPERS

MR. B. J. W. GRIER, TMB: A few supplementary remarks may be in order. Road shocks normally are not confined to a single pulse, but may consist of many pulses. The initial displacement may be amplified in succeeding violent vibrations or may be diminished, depending upon the shock pulse and its frequency, the natural frequency of the suspensions, and their phase relationships. Considered as a rigid body on a single suspension, the vehicle itself has six degrees of freedom; however, this is not the actual case. In fact, the vehicle has more than one suspension with several component parts attached, which have their own modes of vibration.

An excellent example of this complexity is the Radar Set AN/TPL-1, which has not been mentioned. This set is a transportable unit mounted in a trailer body, which is supported by a set of vehicle tires. When transported from one field of operations to another, it is, in turn, mounted in trailer M-1. This equipment was field tested at the Aberdeen Proving Ground, and measurements were made at various locations. The records obtained indicated the complexity of the problem; however, after careful analysis, the principal modes of vibration and necessary data were obtained.

It can be emphasized here that the chief modes of vibration in military vehicles are bounce and pitch, while roll is much less severe. This was borne out in the motion pictures of field tests of military vehicles shown by the reproductions in the paper from Aberdeen Proving Ground. Bounce, as referred to here, is the vertical displacement of the vehicle body, and pitch is the rotation about the transverse axis of the vehicle body. As was shown in the film, pitch action encountered in tanks often causes bottoming of the vehicle suspensions and results in severe damage to equipment, as well as to the vehicle and its personnel.

Records show that pitch frequencies and bounce frequencies differ but slightly from each other. This is due to the inherent design characteristics, where the radius of gyration of the vehicle is approximately one-half the wheel base.

I shall mention, briefly, the following phases of the problem which have been completed or are underway at the Signal Corps Engineering Laboratories:

1. Simulation of actual road shock and vibration conditions in the laboratory.
2. Establishment of adequate performance specifications for equipment which is installed in vehicles and equipment which is transported in vehicles.
3. Revision of equipment design requirements in the light of the above.
4. Compilation and correlation of data on rail transportation with other modes of transportation, to arrive at complete statements on over-all requirements.
5. Measurements to determine the nature of shock and vibration encountered by all types of military vehicles when subject to Arctic conditions.

In connection with the simulation of actual road shocks and vibrations, many devices have been designed and tested - some of which are the Bump Rig, Tumbling Test Machine and modified vibration

machines. The Bump Rig consists of equipment designed to be attached to the rear suspension of the Army 2-1/2-ton truck 6x6 for the purpose of shock exciting the system. The Tumbling Test Machine was designed to simulate conditions encountered by equipment being transported loosely stored in military vehicles. Modifications of existing vibration machines were necessary to obtain increased amplitudes and accelerations at low frequencies, which are predominant in military vehicles.

The subject of vehicular vibrations and shocks encountered by equipment while mounted in military vehicles subject to Arctic conditions is of great interest to the Army. No technical data is available on this subject. Army engineers who have participated in Arctic operations report various opinions of damage due to vehicular shocks and vibrations. It is assumed that these conditions are different, due to varying types of road conditions and the types of vehicles normally used in Arctic transportation. Vibration measuring equipments are being assembled by the Signal Corps Engineering Laboratories for field tests, which are soon to be underway.

DR. GOLDMAN, NMRI: Personnel and comfort problems have not been mentioned. While the matter of comfort may not seem very military, have any measurements been made of vibration and shock in vehicles to which personnel would be subjected?

MR. GRIER: There is no specific data at present regarding personnel comfort. I think that this topic has been neglected. We have felt that personnel is an expendable item.

MR. PRIEBE, SQUIER SIGNAL LABORATORY: Generally, the driver protects himself. If he feels that the vibrations are becoming too severe, he slows down. I drove a Jeep over the washboard course, accidentally, at 25 miles per hour and stopped very suddenly. You can't go very far under those conditions. Your foot gets off the gas pedal, the machine slows down, or something happens that the vehicle breaks down before any damage occurs to personnel.

MR. MONTGOMERY, ABERDEEN PROVING GROUND: What Mr. Priebe says is generally true - we do make light of the driver; but when the driver operated for long distances, everybody had to give a little thought to his comfort. A special seat was made which consisted of soft springs plus shock absorbers. Some commercial companies are further developing it. Some changes have been made in the tank seats. Even though we make fun of it, a little work is being done in that direction.

MR. ARMSTRONG, NOL: Everyone seems to shy away from the subject of spring bottoming. Is any work being done on investigating peak accelerations or shock patterns?

MR. PRIEBE: Work is underway in the development of instrumentation for that particular problem. The problem of bottoming is not as bad as generally stated. We have some rubber bump snubbers on which bottoming takes place. At present we cannot very well use instruments.

MR. GRIER: In addition, quite a bit of work has been done on torsional suspensions. In that direction, inertial controlled dampers have been used, with quite good results, in preventing bottoming. The shortcoming was to get the proper amount of instrumentation to measure the condition where various types of materials come in contact with each other.

MR. PRIEBE: Difficulty was encountered with using instruments. This cannot be overemphasized. No illustrations have been shown of the instrumentation (or at least part of it) being carried on a truck which travels beside the vehicle at the same speed as the vehicle being tested. In this case cable effects are present, and the long cables are very difficult to

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handle. Investigation of the suggestion of using telemetering would be very much appreciated.

MR. RICH, TMB: It becomes evident that the whole group of laboratories have incidental problems at some time. It would be of value if some meeting of the entire group, or just those laboratories concerned, could be devoted to instrumentation problems. All could greatly benefit from a meeting of that kind.

DR. KLEIN, NRL: In answer to that, you will find a summary in Bulletin No. 5, Page 61, encouraging each of you to express his needs in terms of instrumentation. We are planning, for the benefit of all, to hold a symposium in which the subject of instrumentation will be emphasized. Everyone realizes the great need for precise measurements and for instrumentations to get these measurements. Don't fail to read the article and to give the Committee an idea of your needs in instrumentation and how urgent it is to set this meeting, or symposium, or colloquium, for the discussion of instrumentation.

MR. CUNNINGHAM, NRL: Mr. Montgomery, would you discuss some of the actual difficulties with the trailer recording system?

MR. MONTGOMERY: The arrangement is six accelerometers on a light tank, which we run over the washboard course. We have 200-foot, wire-covered cables on stands. the cables are looped back. As the vehicle progresses, it will pull the cables, allowing the tank to go only the 200 feet. This is well in theory, but when pulling cables across the ground and across the vehicle, with the probability of residual magnetism in the tank, other signals will be induced in the circuits.

DR. BALL, NOL: I am interested in the economics of the Army programs. One of the incentives which we have for laboratory simulation is the matter of cost. Most field work requires a lot of personnel and a lot of expenditure. Does the Army have a similar driving force?

MR. PRIEBE: The joint program was undertaken during the later phases of the war when money and personnel in the Armed Forces were still available. Now the economic situation forces us to reduce the number of field tests and go to laboratory tests. We feel that field tests in the Artic region are necessary. Nothing has been said about conditions in the Arctic region. There, sleds are pulled behind vehicles which are traversed over rough terrain - principally ice. It has been found, for example, that oil tanks split open with a few miles of driving. These field tests are necessary, and after that we can continue in the laboratory.

MR. MONTGOMERY: Maybe I should have started with an introduction of what we have at Aberdeen. We have every type of standard military vehicle - not only one, but several of a given type - trailers, tanks, gun motor carriers, etc., and the Ordnance Department would be willing to co-operate with anyone who wants to use the facilities. A great deal of work has been done with the Signal Corps using the Belgian Block course. The facilities for tests are available to anyone who makes the right contacts. If you are interested in running over prepared courses, we have prepared courses plus any variety of cross-country conditions - slope operation, etc. We can simulate anything - closely, if not accurately.

DR. KLEIN: The subject of simulation in the laboratory versus full-scale tests is being studied by all activities. The British can give us a lesson in the subject of simulation, because their funds are more drastically cut than ours.

COL. BRIGHT, BRITISH SUPPLY OFFICE: Yes, in England we are running on a smaller scale now because of the economic crisis.

We are forced to laboratory simulation to a greater extent than you are. In England we have a fighting vehicle proving establishment which, to us, is identical with what the proving ground at Aberdeen is over here. We have been working on much the same lines as regards shock and vibration, but probably more from the laboratory point of view, to cut cost. Against that, we had a longer period during the war when we were interested in these problems. One of the things which has not been stressed in these papers but which we came up against is that although it is perfectly true that the driver of the vehicle will protect himself from excessive shock, still the fact remains that you must consider the fatigue of the driver. He must be able to fight efficiently, even at the end of a long day. We must consider shock and vibration and how it affects the driver if we expect him to drive in convoy, for example, where he cannot regulate the speed of the vehicle. Possibly on that side we have done rather more, whereas on the full-scale side we have probably done less. We are now emphasizing the laboratory side more in England than you are here.

DR. KLEIN: Before we leave the subject of the driver, we should be thinking further along these lines. With the prospects of higher speeds in the future, the driver must be considered if effectiveness in operation is to be achieved. Since Dr. Goldman is the lone man representing the interests of the driver, it will be encouraging to his group to know that several activities are thinking along these lines. Each activity is considering what might be done to help personnel as well as machines. Here at NRL, at TMB, and at several other organizations programs are underway, in co-operation with medical shock and vibration groups, for the benefit of personnel.

MR. JASPER, TMB: I notice that in summarizing the data on road vehicles, the information has been based mainly on frequencies and peak accelerations. I was wondering if any effort has been made to summarize the number of cycles at each frequency and, also, the number of cycles during individual shocks?

MR. PRIEBE: We obtained some instruments from Chrysler on which we recorded the number of times acceleration was exceeded at a certain frequency. These were peak reading instruments which had magnetic release. They did not prove too useful, and we have not made exhaustive studies, as yet, but contemplate doing so. We also used the Biot pendulum to analyze the records we have accumulated. We have some of these records now and must find out what to do about them.

MR. McGOLDRICK, TMB: In the same connection is there any definite reason for the measurement of acceleration rather than velocity or displacement?

MR. PRIEBE: Acceleration was decided upon in a conference with Dr. Mindlin because acceleration represents force. Some displacement as well as velocity measurements were made. Velocity measurements are discussed in the paper by the Frankford Arsenal, and displacement measurements are discussed in earlier papers which covered the displacement measurements made with General Electric instruments. However, in working at high frequencies, attendant forces are concealed in displacement measurement.

DR. TRENT, NRL: I should like to see if we could tie in some of the results with information gathered in a recent visit to England, pertaining to shock in ships. In that case, in their studies of underwater shock, the frequencies involved lay in three bands: below 5 cycles, or the low-frequency band; those from 30 to 100 or 200 cycles, the medium-frequency band; and the high-frequency band, of the order of kilocycles - 1 or above. The high-frequency band is characterized by high peak acceleration. Unless the equipment

is made of brittle castings, nothing happens. In the low-frequency band, large displacements can cause some trouble from misalignment of shafts. The medium-frequency band is the one which damages the equipment mounted on the ship. The point to emphasize, I think, is that what tears up the equipment is not necessarily acceleration, but how much strain is imparted to the equipment. Mr. Struthers, here at NRL, does non-destructive testing which involves acceleration of about 1000 g. If these three frequency bands are found in military vehicles, is there one band you can put your finger on as ruining equipment?

MR. PRIEBE: Yes, the paper presented by Mr. Lee gives some results of destructive tests on electronic equipment carried in military vehicles. The bands which are most damaging to equipment have frequencies which lie between 8 and 60 cycles per second. The two frequency bands are due to tires and to resonant vibrations of the components and equipments carried in vehicles. Repeated low-frequency accelerations in the neighborhood of 4 g have components which continuously excite resonant frequencies of the destructive variety.

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DAMAGE TO COMPONENTS AND EQUIPMENTS; EFFECTIVENESS OF SHOCK MOUNTS

By

Norman Lee
Coles Signal Laboratory

The types and extent of damage to electronic components, when used in vehicular installations, are described. The relative effectiveness of various types of shock mounts, in terms of damage to equipments, are indicated. The consequences of using wrong shock mounts for vehicular applications are demonstrated.

The material presented so far has dealt with instrumentation for measuring vehicular vibration and the results obtained on many vehicles. Just how these vibrations and incidental shocks will affect equipment is of considerable interest. This is an account of an experimental attempt to obtain data on this problem, principally with regard to electronic components.

In general, earlier papers showed that vehicular vibration is characterized by a very complex wave form, but with characteristic periodicities which can be traced directly to the different elastic systems of the vehicle. It was shown that these periodicities may be grouped into distinct and different frequency sectors. The absolute accelerations are relatively small - a maximum of about 3 g in most cases. Offhand, it might be assumed that, since the accelerations are so low, there need be little concern for this type of problem, but I will attempt to show that a long time application of vehicular vibration can cause considerable damage to the equipment.

In the previous discussions, the question arose as to which was the more significant aspect - the characteristic frequencies of the vehicle or the magnitudes of the acceleration. Both of these factors are involved and, in addition, the time of application is also extremely important. As pointed out in the preceding papers, the vehicle movement is extremely complex and, despite the nominal periodicity mentioned, there are many discontinuities in the wave forms. Since the excitation is continuous while the vehicle is in motion and this wave form is capable of exciting and producing stress repititions in almost any elastic element, many of the failures are due to fatigue and wear. The total number of stress repititions, which is the fatigue criterion, depends on the length of time the vehicular vibration is applied.

We concern ourselves with three problems:

1. The types and extent of the damage to electronic components when used in vehicular installations.

2. The relative effectiveness of various types of shock mounts in terms, again, of damage to the equipment.

3. As corollary to (2), the emphatic consequences of using the wrong shock mount for vehicular applications.

These tests involved large numbers of components and a long time excitation in a typical vehicular installation; therefore, they are considered quite representative of the effects of vehicular applications in the field. Although the results of these tests were originally to be interpreted in terms of damage, I shall attempt to correlate some of the results with vehicular vibration data.

These tests were initiated to determine the effectiveness of various types of shock mounts. Although there had been many efforts and much speculation on vehicular shock mounting problems toward more satisfactory results, it had been difficult to translate them into actual practices because of the difficulty in resolving most vibration and shock considerations in terms of damage to equipment.

At the outset, to determine the magnitude of the problem and the need for further work, a preliminary test was run involving a newly developed shock mount, a widely used typical commercial shock mount and the technique of bolting the equipment down solidly to the vehicle. Four chassis were fabricated; each one incorporated four typical oscillator circuits, eight miniature type tubes, sixteen ceramic capacitors and a number of resistors. One of the chassis was of 1/32 inch steel and was bolted solidly to the equipment. The second chassis was of 1/8 inch steel and was supported on experimental shock mounts. The third chassis was of 1/8 inch steel and was supported on a representative commercial shock mount in accordance with common practice at the time. The fourth chassis was also of 1/8 inch steel and was bolted solidly to the vehicle. These chassis were placed on a frame in a panel truck (see Figure 1) and were driven over representative roads for 1,000 miles.

The damage after this test was principally in the oscillator assemblies. It was noted that one of the chassis using shock mounts was damaged considerably more than even the chassis which were bolted down solidly. In the light of what has been disclosed in preceding papers, however, the reasons for this are apparent. The resonant frequency of this chassis on its shock mount was about 12 cycles per second. If you remember, one of the strong frequency sectors in wheeled vehicles was from 8 to 12 cycles per second, and it was due to the axle system. Consequently, the shock mount amplified the vehicle vibration in this range and, in combination with the fact that it had a relatively sharp discontinuity in its load deflection curve, the impulses passed on to the components had an extremely high harmonic content. This could excite the components more than they would have been if they were subjected just to the vehicle vibration.

In most cases, the failures of the oscillators were due to parted turns in the choke coils. Although the choke coils were placed with a 1/8 inch spacing to nearest surface, there was sufficient excitation so they vibrated against the surface and finally wore through some of the turns. This is a wear phenomena and, of course, the damage is dependent on both the stimuli applied to it and the length of time they are applied. The importance of providing sufficient clearance is evident.

The frame was then given 40 drops on a hard surface by lifting one corner of the frame 10 times and dropping it from a height of 12 inches. The results are shown in Figure 2.

The purpose of this test was to get some indication of the consequences of rough handling. It had been our experience that this was a particularly important factor in the life of the equipment. Many of our equipments, which were satisfactory once they were installed, failed in transportation due solely to the dropping and

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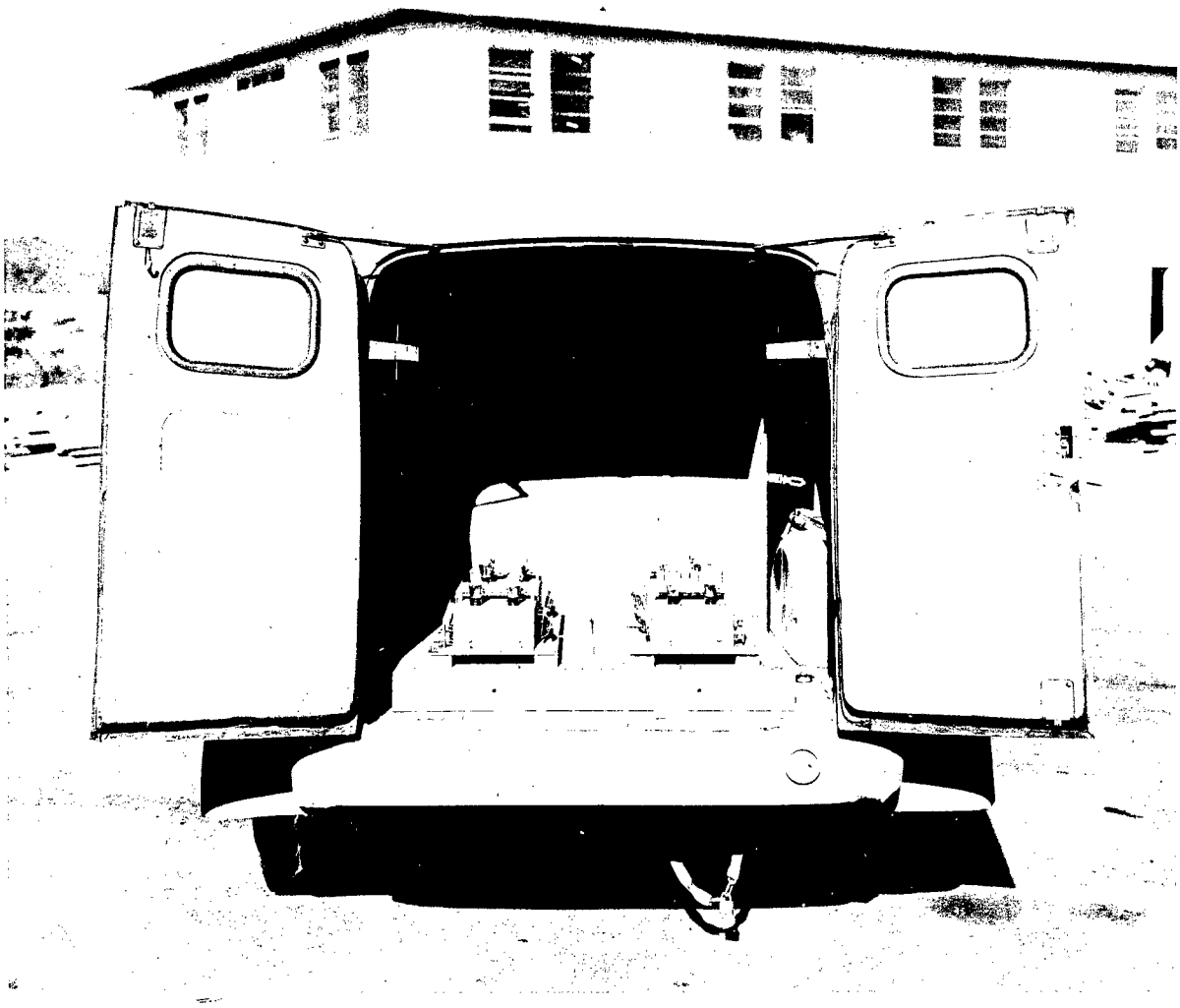

Figure 1

tumbling. This test was somewhat extreme, however, because of the large number of drops, and the results should be interpreted in that light. The reason for the large number of drops was that, since it was not a carefully controlled test, dependence was put on the averaging possibilities of a multiplicity of drops to eliminate random effects.

This test indicated that a more exhaustive investigation was required, because of the emphatic differences in damage in the various chassis. In particular, the shock mount which represented a widely used practice produced an excessive amount of failures.

Having shown, then, that this problem was a serious one, a similar test was devised to determine the best shock mounting solution. Six large chassis were devised, each one incorporating a larger quantity of components: twenty-two miniature type tubes, eight metal 6-volt tubes, sixteen ceramic capacitors, twenty-four tubular capacitors, sixteen plate chokes and eight vitreous enamel resistors (see Figure 3). The six chassis were identical in construction. Chassis No. 1 and No. 2

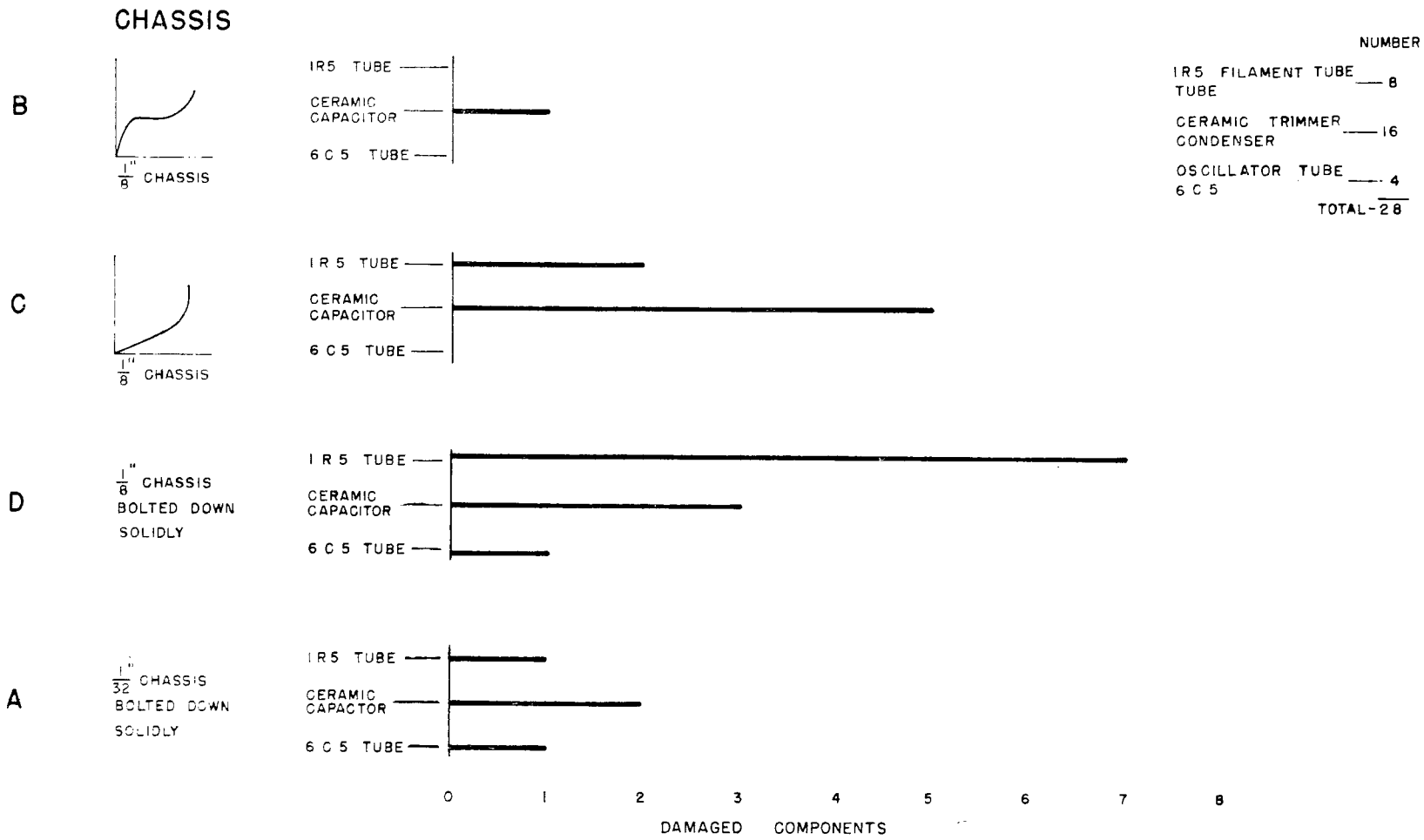

Figure 2 Damage Due to 1000 Mile Ride and 40 - 12" Drops

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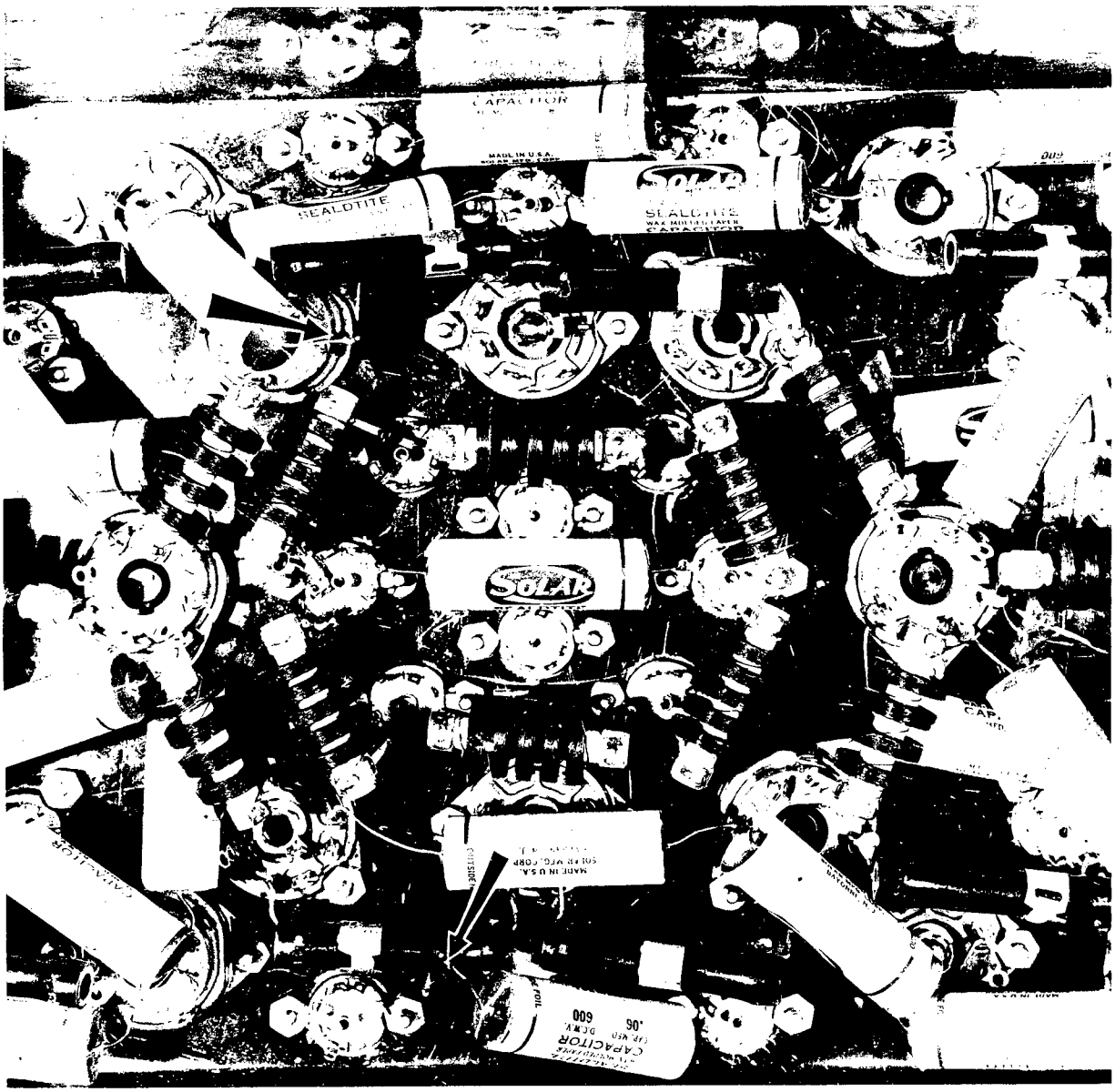

Figure 3

were supported on shock mounts developed at the Signal Corps Engineering Laboratories, of two different load ratings. Chassis No. 3 and No. 4 were mounted on essentially compression mounts of another manufacturer; the difference between the two being a different rubber compound. Chassis No. 5 was mounted on a shock mount almost identical to No. 6 but of greater stiffness and more damping. Chassis No. 6 was mounted on a commercial shock mount of the type used in the previous test and was used as a control.

These chassis were mounted on a frame placed in the same vehicle and driven 2,200 miles over representative roads. The frames were systematically shifted after every 400 miles to eliminate the variables introduced by different locations of the chassis in the vehicle. After the 2,200-mile ride, the six chassis

were inspected and the damage noted (see Figure 4). The frame on which the chassis were mounted was then again given drops by lifting the corner of the frame, and again the damage was noted (see Figure 5).

In this test, no effort was made to use chassis bolted solidly to the vehicle. Because of a number of considerations not pertinent to the vibration problem, shock mounts are desirable on military equipment when it is possible to use them, and the prime emphasis in these tests was to determine the adequacy of the alternatives which were available at the time. The choice of the operating load on the shock mounts represented either a prevailing practice or the manufacturer's recommendation.

The damage produced by the vehicle's motion depended on whether the component was vulnerable to fatigue and wear. In general, with the exception of those components which failed due to the high accelerations in the dropping test, components failed due to the consequences of the long time of application of the vibration. Leads failed in fatigue; components wore, as in the choke coils; soldered joints parted; and, on inspection of some of the inoperative tubes, it was noted that even these failures were due to fatigue and wear. The problem of fatigue is susceptible to a straightforward solution, since its parameters are known. It appears that the failures due to internal wear occur when an excited element vibrates and its constraints are not true nodes. There is relative movement between the element and its constraints. This relative movement produces a large amount of wear in a comparatively short time and is particularly true in the case of vacuum tubes. The extent of damage to individual components is best seen by examination of the graphs.

As far as military vehicular vibration itself is concerned, it appears that a considerable improvement can be made over some of the present techniques, by the use of shock mounts properly selected. In general, the characteristics required for an adequate vehicular shock mount are a relatively high resonant frequency from 25 to 35 cycles per second (the vehicle spectra included in the paper on "Transmission of Road Impulses to Fire Control Instruments" show that this is a fairly vibration-free sector), a relatively long displacement possibility (about 3/16 inch), a smooth load deflection curve and as much damping as it is possible to incorporate in the mount. At the present stage of the art, however, it appears that, despite this, some failures will always occur after a long vehicular ride.

The results of the test indicate that the designer has an enormous responsibility in using shock mounts, because, through improper selection, he may increase the damage involved in the equipment by a factor of six; or he may decrease the life of the equipment by almost the same factor.

While this test was an expedient at the time it was performed, equipment designed for vehicular installations, despite the fact that the art has progressed somewhat, must still be verified by means of such an empirical test. Although the designer now knows what is going to be applied to the equipment, he still must have information on the capabilities of the components in terms of this type of excitation. However, this information is non-existent, particularly for electronic components.

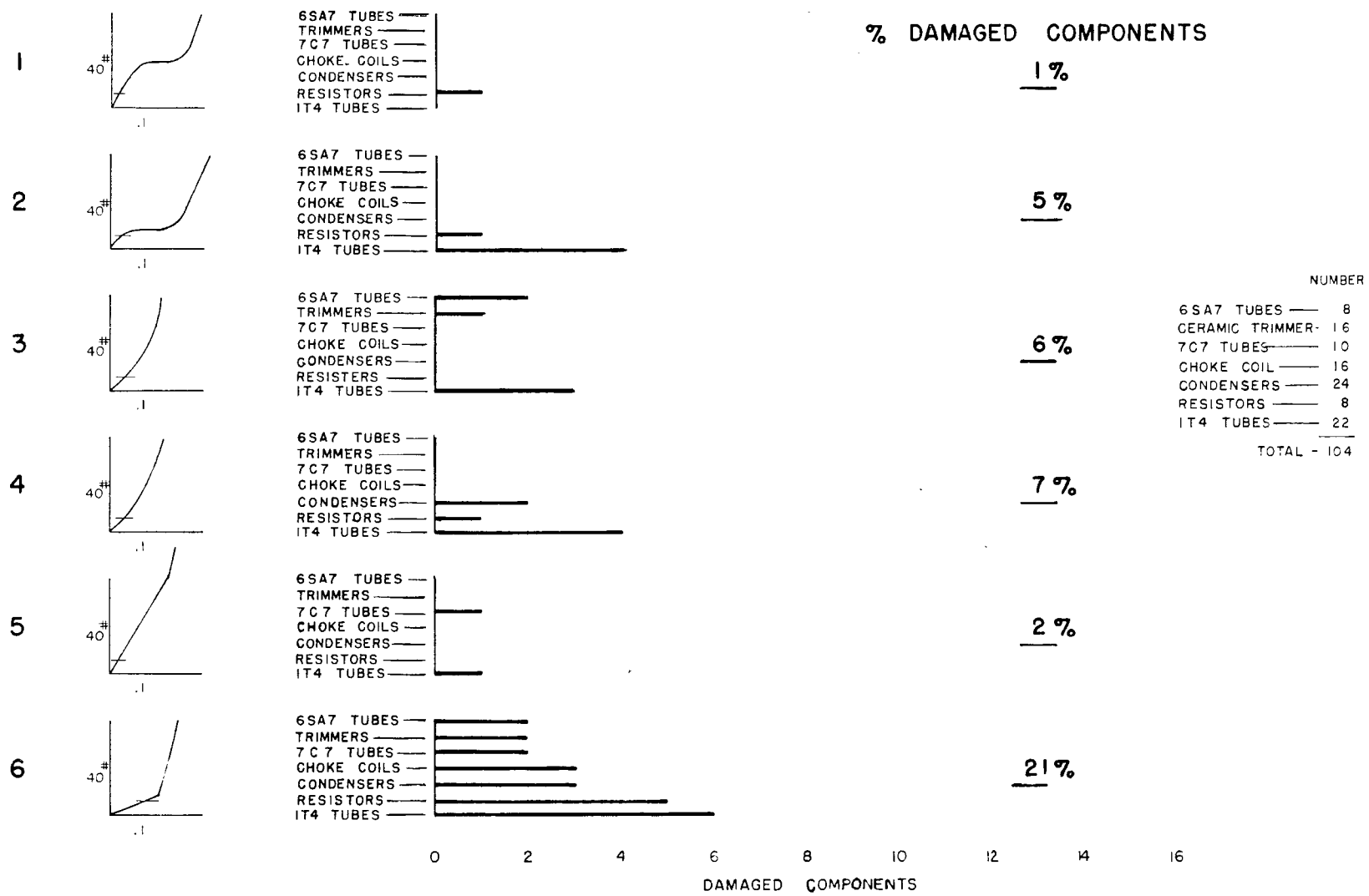

Figure 4 Damage to Components Due to 2200 Mile Ride

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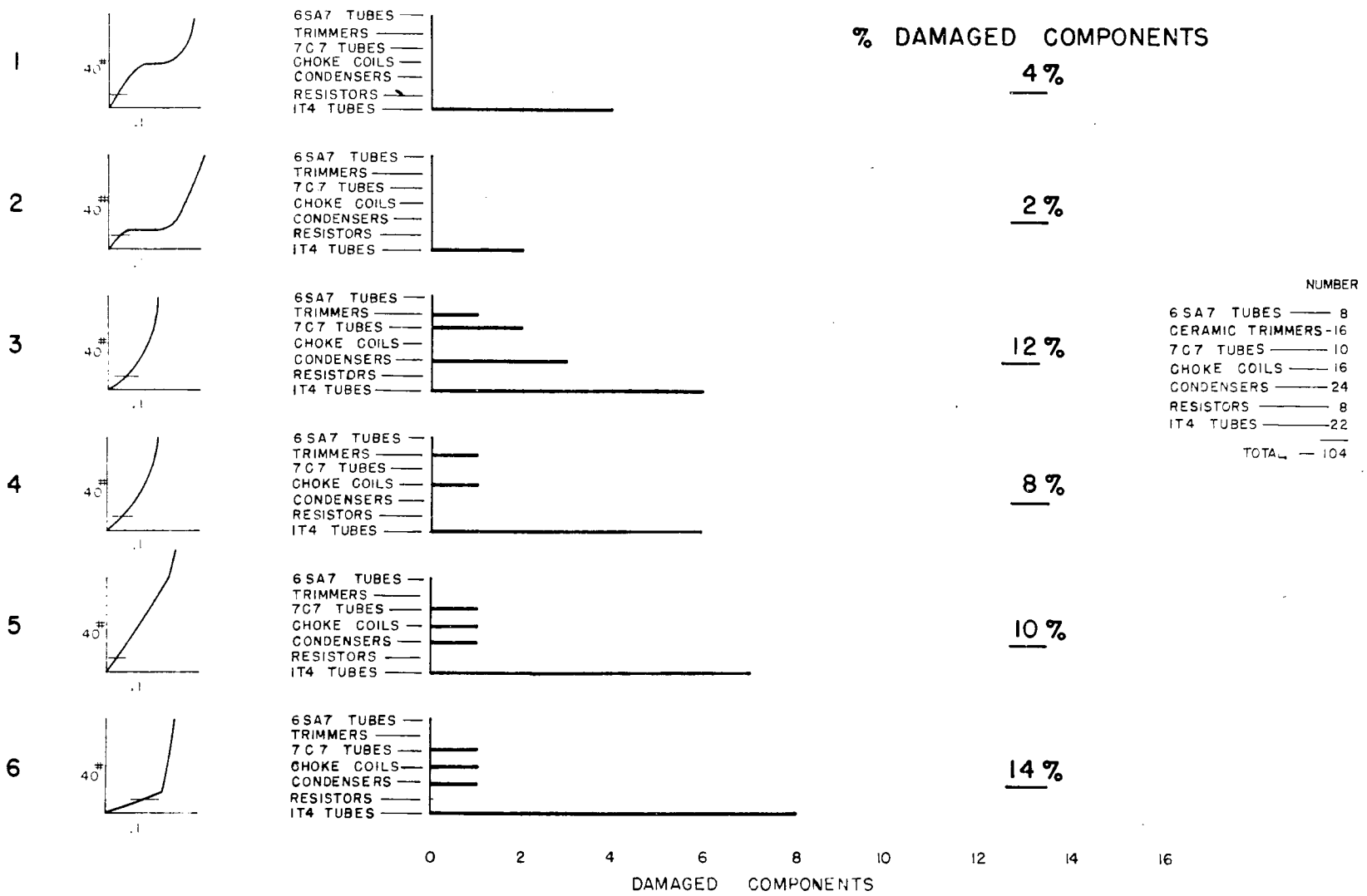

Figure 5 Damage to Components Due to 40 -12" Drops

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DISCUSSION

MR. TREITEL, NRL: Mr. Lee, could you give more detail about the tube failures? For instance, it was stated that during the test the components were not energized. Then what constituted a tube failure?

MR. LEE, COLES SIGNAL LABORATORY: After completion of the tests, the tubes were put into a tube checker. In some cases inter-element shorts were indicated, and in others opens were indicated.

MR. TREITEL: Do you think there would have been a higher percentage of tube failures, especially in the miniature series, if the tubes had been energized?

MR. LEE: Probably so in the filament type tubes, but quite a few failures were due to the time element involved. With all the energy coming up from the base and causing the elements to vibrate, since there wasn't any support at the top, there probably was some relative movement between the grid support and the filament. In some of the tubes the grid wire had actually worn through the mica spacers. It was a good deal of trouble to energize them, and we felt that quite a bit of the equipment is actually driven around cold and that the tests were enough to give us the answer we wanted; but you are right.

MR. HENOCH, BuSHIPS: In using different mounts, with the same rubber elements in them, were any of these elements adjustable as far as physical properties and geometry are concerned?

MR. LEE: The mounts were the same as far as geometry is concerned. In most cases natural rubber compounds were used. We have, possibly, not as valid a story as we should have on quantitative data, but we know the right direction to go. If we can, we work out something that resonates between 25 and 35 cycles per second. As for the equipment, the resonant frequencies are relatively high -- possibly 60 cycles per second, 60 to 100. We use wrought materials with the best physical properties which can stand a good deal of elongation and deformation. As far as we can, we are also trying to make use of strain-energy phenomena to get as much yielding as we can without permanent damage.

MR. WALSH, NRL: A word of caution should be expressed about forming any conclusions on the relative merits of natural rubber or synthetic rubber based on these tests. When the compound was changed in the mount under discussion, the characteristics of load deflection curve were changed. The initial portion of the load deflection curve was stiffened, which got the natural frequency of the load on the mounts out of the driving frequency range of the truck. This could have been accomplished by changing the durometer of the natural rubber mount. Therefore, no conclusions should be drawn on natural rubber versus synthetic rubber mounts. Initial slope of the load deflection curve of the mounting must be considered in the case of steady state vibration, and the slope should be chosen so that the load on the mount is not resonant with any of the exciting frequencies of the truck. It is not a question of the natural rubber mount being a good or a bad mount. It is clearly a case of a good mount being misapplied.

MR. LEE: I did not intend to imply that.

DR. KLEIN, NRL: Mr. Lee mentioned Navy type shock mounts as being different. Does the theory of shock mounts discriminate -- are there other distinctions?

MR. WALSH: In answer to Dr. Klein's question, the drop test which Mr. Lee performed comes closer to the Navy high-impact shock test than the tests of the equipment in the truck. This drop test subjects the equipment to a velocity change, and the impact characteristic of

the high-impact shock test is a velocity change.

MR. JASPER, TMB: You spoke about several mounts having the same geometry. Isn't it the load deflection curve which indicates the suitability of the mounting, and the geometry doesn't enter into it?

MR. LEE: Yes, you can't select shock mounting on the basis of how it looks.

MR. PRIEBE, SQUIER SIGNAL LABORATORY: I want to point out that in the Army the space requirement and time are moving factors. We have a given space in a tank where we have to put electronic equipment, and it has to fill certain performance requirements. As to shock protection devices, they are a necessary evil. Either we don't think about the matter at all, or it's the last thing we think about. Hence, we had to make this study on the basis of geometry.

MR. JASPER: Regarding shock mounting of equipment in trucks, have you found any definite frequency for which you design the mountings for road vehicles?

MR. LEE: As a general practice, we try to keep the frequency between 25 and 30 cycles per second.

MR. JASPER: Do you use Barry mounts?

MR. LEE: We have used them on some of our equipment.

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SOME GERMAN METHODS OF INVESTIGATING FORCES IN VEHICLES

By

Frances F. Vane, TMB

This paper concerns dynamic motions and forces observed in tanks, trucks and railroad equipment during U. S. Naval Technical Mission to the MAN Research Institute in Augsburg, Germany in October 1946.*

In October 1946, a U. S. Naval Technical Mission was sent to Augsburg, Germany to bring to the United States the laboratory equipment in the MAN* Research Institute. The information presented was obtained from engineers at the Research Institute. Although the MAN industry is known in this country principally for the manufacture of diesel engines, the industry also made locomotives, railroad cars, armored vehicles, and trucks. It was logical, therefore, that the staff of the Research Institute should become interested in the investigations of motions and forces in vehicles. We will describe some of their methods in measuring forces that might be applied to problems in this country.

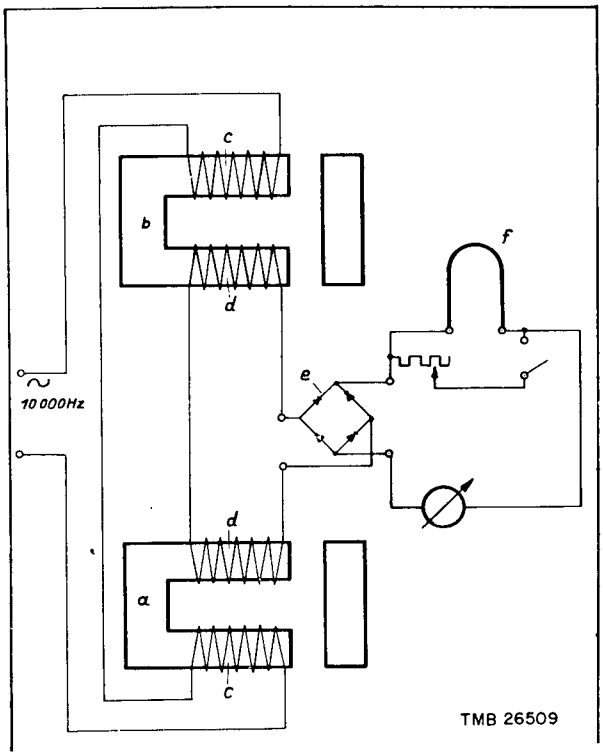
INSTRUMENTATION

Instruments used in these investigations warrant consideration first, since the validity of results and the trials and tribulations in obtaining results are dependent upon them. Instruments for determining riding qualities of vehicles are identical in principle to many used in the United States. Their American equivalents have been or will be described by other papers. Discussion of this topic is therefore omitted here. The Lehr inductive strain gage however, was used with unique applications in many field investigations and will be described.

THE LEHR INDUCTIVE STRAIN GAGE

The Lehr inductive strain gage depends on the relative motion between a magnet and its armature. As shown in the wiring diagram in Figure 1 the gage magnet and its comparison magnet each have two coils. The primary coils of each magnet are in series with a 10,000 CPS carrier frequency generator. The secondary coils of the two magnets are connected oppositely, so that the coil of the comparison magnet is always at higher voltage than that expected in the coil of the gage magnet. This is accomplished by suitable adjustment of the magnet gaps. Then, as the gap of the strain gage magnet changes, a current flows and there is a change in the amplitude of the carrier wave in the secondary coils. A copper-oxide rectifier cuts off all signals below the zero line and the string galvanometer, with a natural frequency of 2,000 CPS, filters the carrier frequency and the higher strain frequencies. The gage is satisfactory for a range of frequencies from

* *MAN – Machinenfabrik-Augsburg-Nurnburg*



a. Gage magnet and armature
b. Comparison magnet and armature
c. Primary coils
d. Secondary coils
f. Galvanometer string

Figure 1 Wiring Diagram of the Lehr Inductive Strain Gage.

1/60 to 1,000 CPS, and the accuracy is plus or minus one percent.

A typical installation for the operation of the gage and for recording strains is shown in Figure 2. The storage batteries are for the d-c motor which drives the 10,000 CPS carrier generator. The "measuring" boxes contain comparison magnets, milliammeters for checking primary and secondary coil currents, and connections to the gages and to the string oscillographs. The oscillograph was specially rigged; the galvanometers were moved back about 30 inches to obtain greater magnification. In this way a gage installation was obtained which was sensitive enough for the purposes of the Research Institute without incurring electronic difficulties inherent in amplifier design. The Germans admitted that the complete equipment was bulky but contended that they were able to install the entire outfit in comparatively small vehicles.

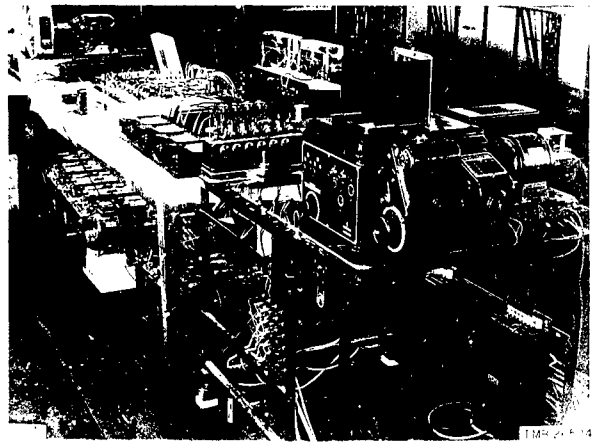


Figure 2 Typical Installation for Operating and Recording with the Lehr Inductive Strain Gage

The component parts of the strain gage itself are shown in the drawing, Figure 3. Each half of the gage was mounted on a conical pin soldered to the member whose strain was to be measured. The magnet, with primary and secondary coils, was fastened to one-half the gage and the armature to the other half.

The gages were made in 50 mm and 20 mm gage lengths. The newest gages were temperature-compensated by the judicious use of material so that the gage expanded at the same rate as the steel member to which it was fastened. The centers of gravity of the two halves were arranged so that accelerations up to 2,000 g affected both portions equally, leaving the strain values unaffected.

The inductive strain gage was adapted for use in loading rings, pressure gages, and pressure boxes, and was used in measuring accelerations, torsions and deflections. The gage installation is useful for three reasons: the gage operates satisfactorily up to fairly high temperatures, it can be calibrated after in-

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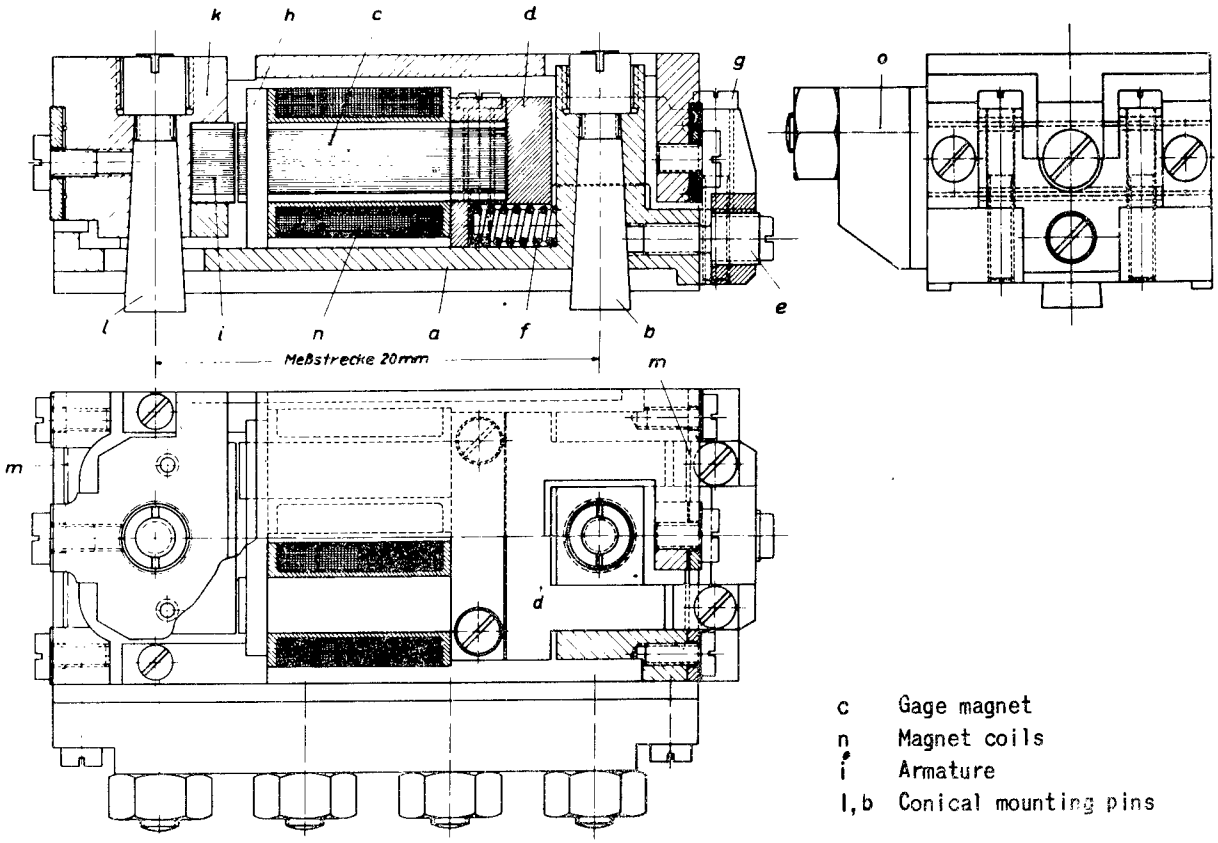


Figure 3 Diagram of the Lehr Inductive Strain Gage

stallation, and no electronic amplifiers are necessary. The disadvantages are the bulkiness of the installation, the elaborate equipment needed for soldering gage pins in place, and uncertainty about the true gage length.

FIELD INVESTIGATIONS

We are now ready to see how the inductive strain gage and its adaptations were used for investigations on a variety of vehicles.

Locomotives

The inductive strain gage was used to measure strains in the connecting rod of a locomotive for the German Railroad (*deutsches Reichsbahngeseilschaft*) during regular runs. As shown in Figure 4, gages were connected to the web and flanges of the connecting rod and the leads were conducted through protective tubing to the recording equipment. There are four points of bending in the leads, but the amount of bending at each point is small. This can be seen more clearly in Figure 5 for three of these points of bending. The results of this particular test indicated that the stresses in the connecting rod were not excessive.

Railroad Cars

Another problem the German Railroad had was in regard to car axles. The data for the calculation of loads to which a railroad car axle is subjected during service was inadequate. The weight supported by the axle is only a fraction of the real load. Lateral forces on the

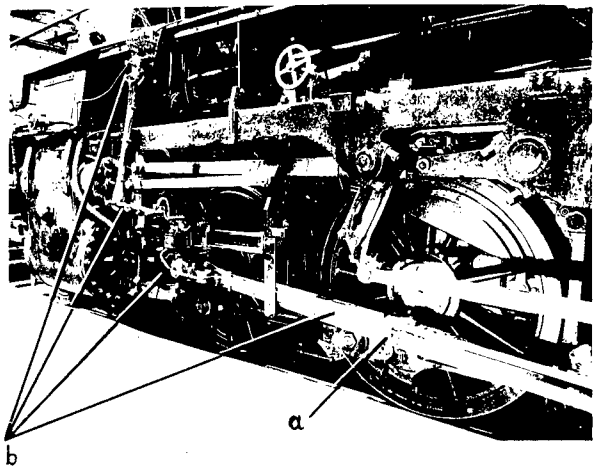


Figure 4 Gage Installation and Gage Leads on the Connecting Rod of a Locomotive

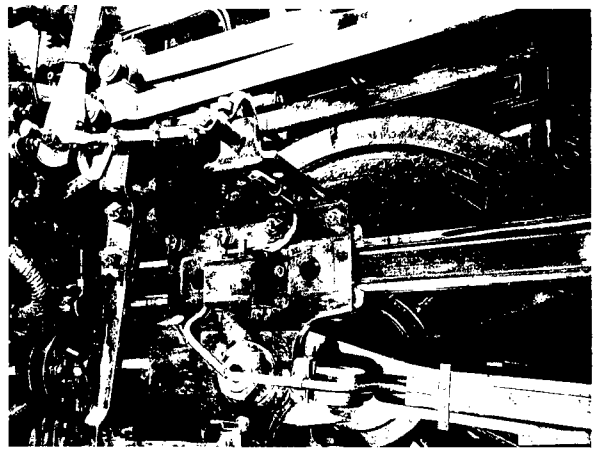


Figure 5 Details of Gage Leads on the Connecting Rod of a Locomotive

wheels and on the rims are not amenable to calculation. When the train is moving, the wheel flanges press alternately against the right and left track, resulting in a weaving motion of the car. Strain gages were installed as indicated in Figure 6. Pairs of gages were installed on opposite sides of the axle so that bending strains, as well as axial strains, could be measured. The gages were connected to the recording system through slip rings. Service loads were measured on passenger car axles up to 88 mph (140 km/hr) and on freight car axles up to 47 mph (75 km/hr), with the loads greater for the freight car axles than for the passenger car axles. The lateral rail impacts added materially to the loads on the axles; on the other hand, the impact due to rail joints did not add appreciably to the loads. As a result of this investigation, a greater knowledge was obtained of the dimensions necessary to reduce fatigue failures on railroad car axles.

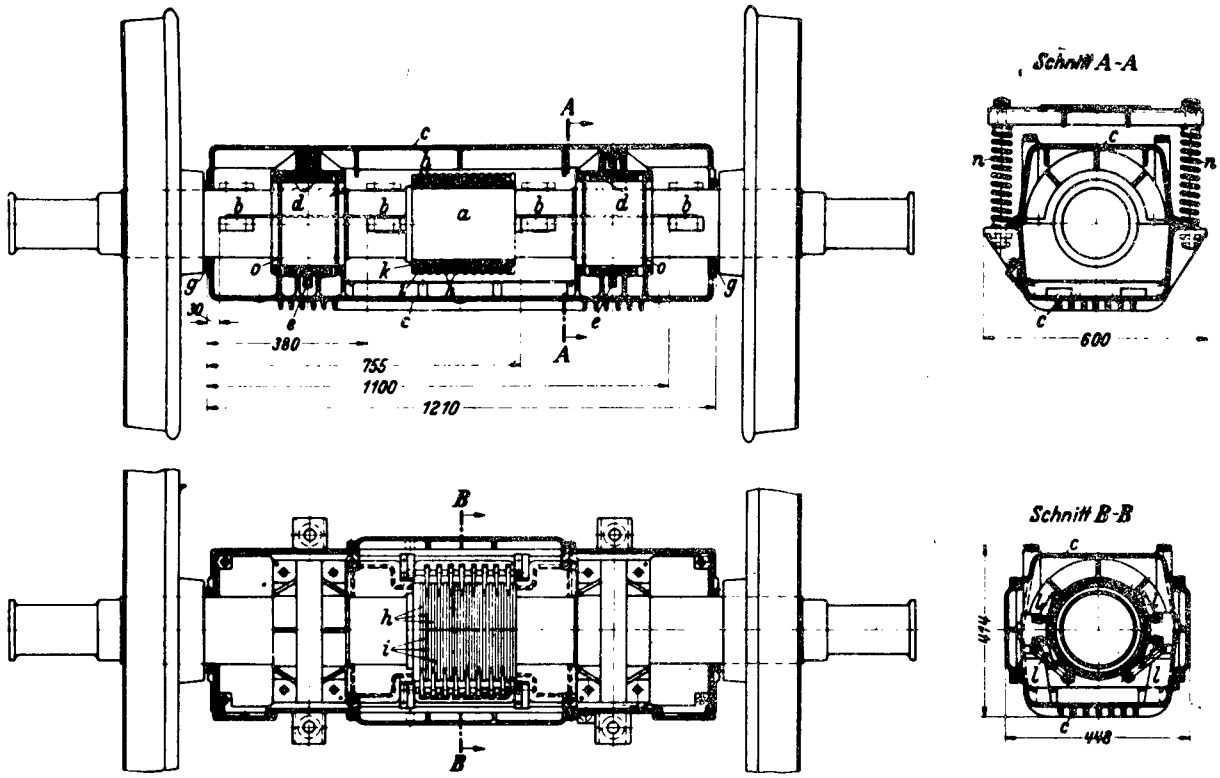
Caterpillar Tread Vehicles

Two investigations will be presented here, one of deflections in link bolts of tractor treads, the other an evaluation of drive shaft forces of a tank during braking.

D 7 Tractor. -To improve the design of tract treads it was deemed necessary to find peak forces in the tread links during operation of the vehicle. The bending of the link bolts was believed to be a good index of the forces involved. A considerable amount of experimentation was necessary before an installation was finally built that would satisfactorily record the bending stresses in a link bolt.

A drawing of the D 7 tractor is shown in Figure 7, and a more detailed view of the track treads in Figure 8. An auxiliary chain track, moving at the same speed as the track tread, is used to transmit the signal from the gage through slip rings to the recording system in the vehicle. A diagram of the slip rings is shown in Figure 9. The gage installation on the link bolt is shown in Figures 10 and 11.

The highest tractive force in the track tread was 9,400 pounds, (4,280 kg) at a speed of 38 mph (60 kg/hr). The tests indicated that speed alone was not decisive in increasing track link tread forces, both other conditions-the nature of the ground, the position of the member in its path, centrifugal forces, and the weight of the vehicle-were of importance. As a consequence of this investigation, weight was saved in new designs thus



b. Inductive strain gages for measuring axial and bending strains.
e. Inductive strain gages for measuring torsion.
a. Slip rings.

Figure 6. Strain Gage Installation on a Railroad Car Axle

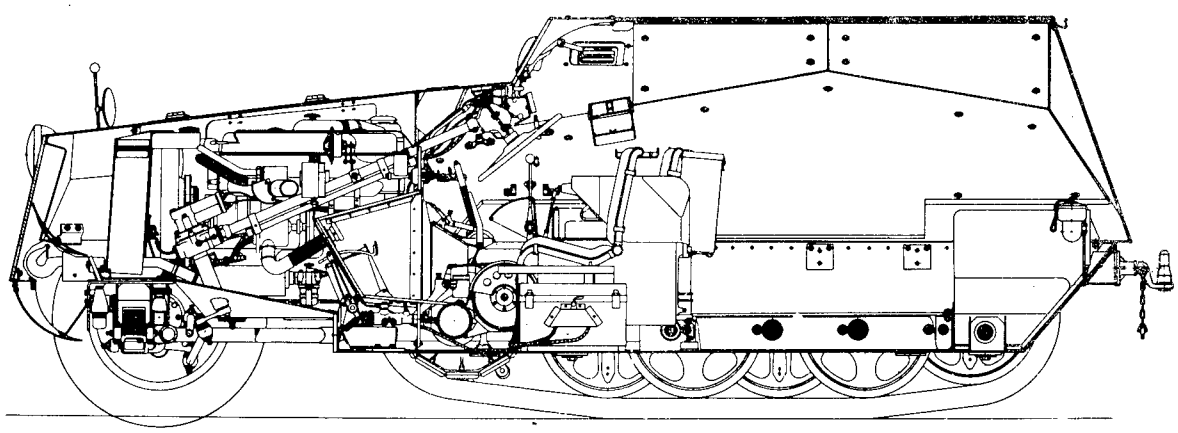


Figure 7. Diagram of a D 7 Tractor

enabling the track tread to be operated at higher speeds.

Tank. -The problem was to measure axial, bending, and torsional forces in the drive shaft of the tank at various speeds and during braking. Exterior views of the vehicle are shown in Figures 12 and 13. At the ends of the drive wheels the housings for the slip ring installations are

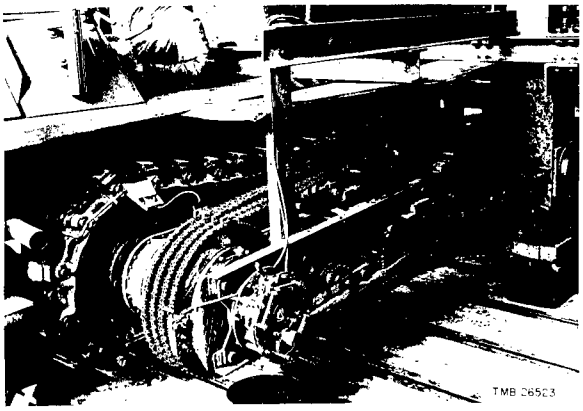
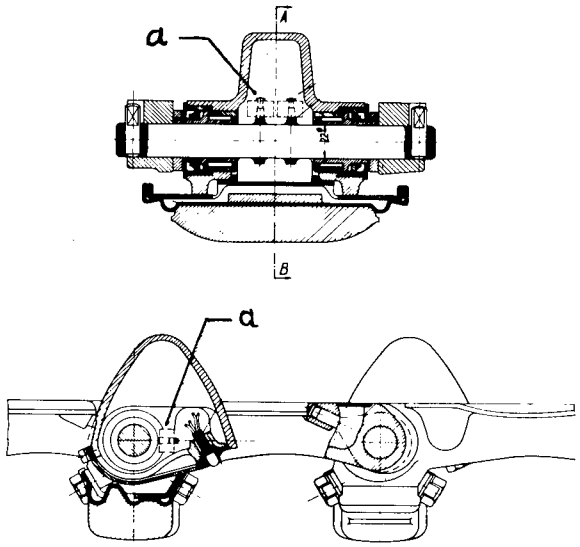


Figure 8 Track Tread and the Signal Chain Track of a D 7 Tractor



a. Location of the inductive strain gage to measure bending.

Figure 10 Diagram of the Gage Installation on a Link Bolt of a D 7 Tractor

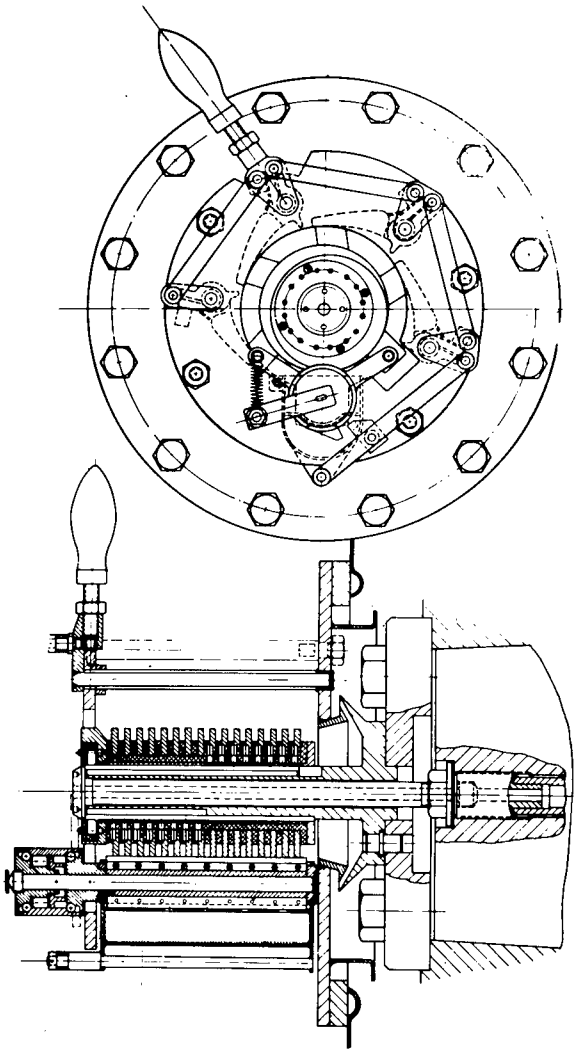


Figure 9 Slip Rings on the D 7 Tractor

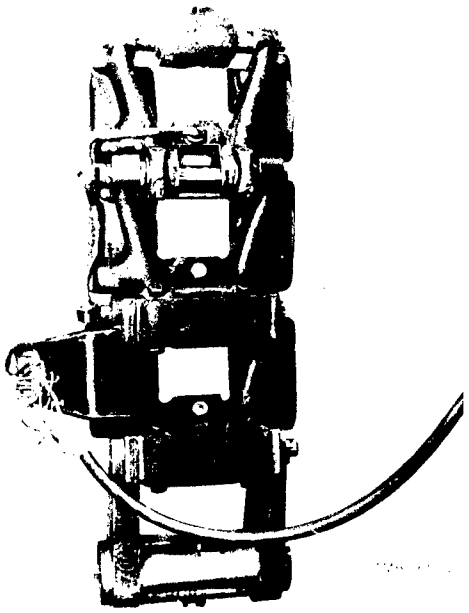


Figure 11 General Arrangement of Gage and Leads on the Track of a D 7 Tractor

Figure 12 Panther Tank - Front View

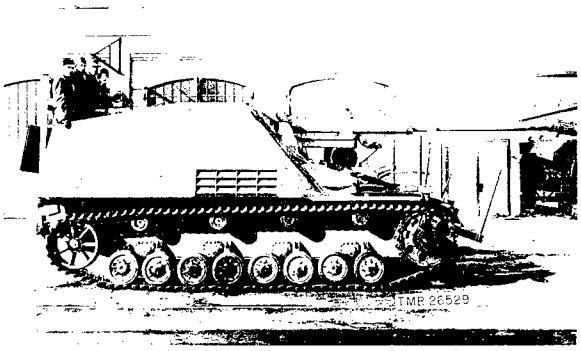


Figure 13 Panther Tank - Side View

shown. The operating and recording equipment fitted into the vehicle is shown in Figure 14.

Trucks

It was desirable in this instance to measure bending and torsional strains which occur in the rear axle of a truck during starting and braking. The gages for measuring axial and bending strains are shown at the right of the differential housing, and those for measuring torsional loads are shown at the left of the differential housing, in Figure 15. A more detailed view of the gage installation is shown in Figure 16.

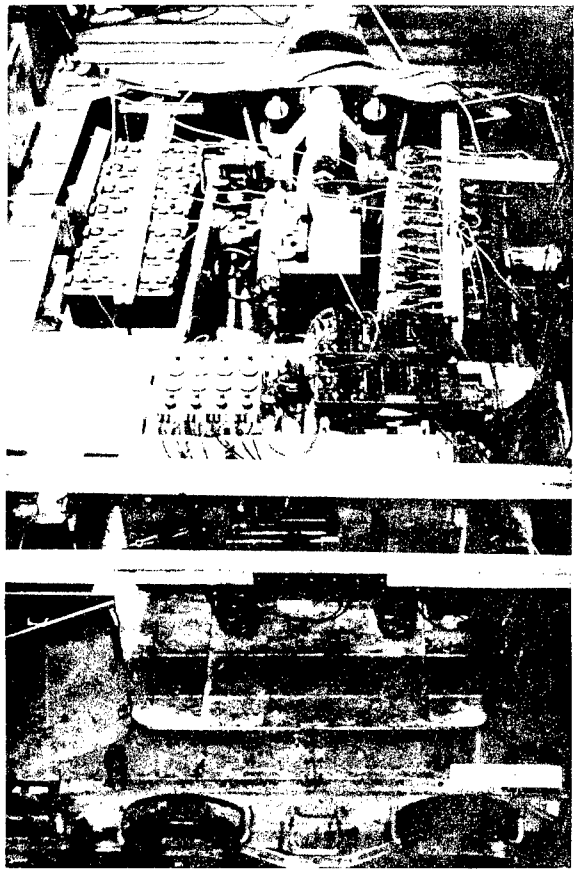


Figure 14 Installation of Operating and Recording Equipment in the Panther Tank

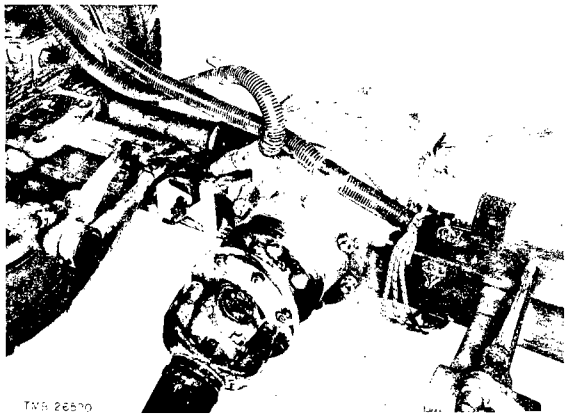


Figure 15 Strain Gage Installation on the Rear Axle of a Type LI Truck

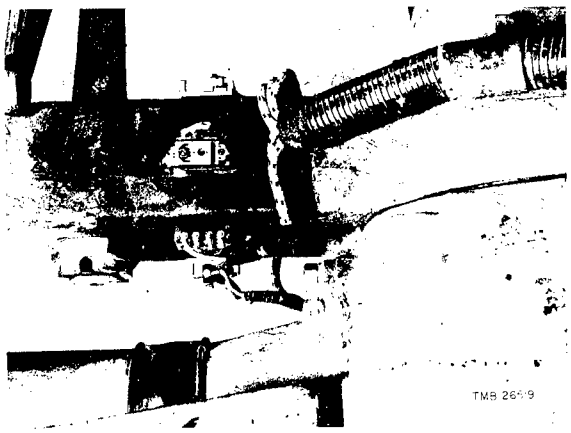


Figure 16 Details of Gage Installation on the Axle

Engines

Tests have been made to measure strains during operation in diesel engine components such as connecting rods, engine frames, cylinder heads, and crankshafts. A typical example is that of the crankshaft, shown in Figure 17, where gages

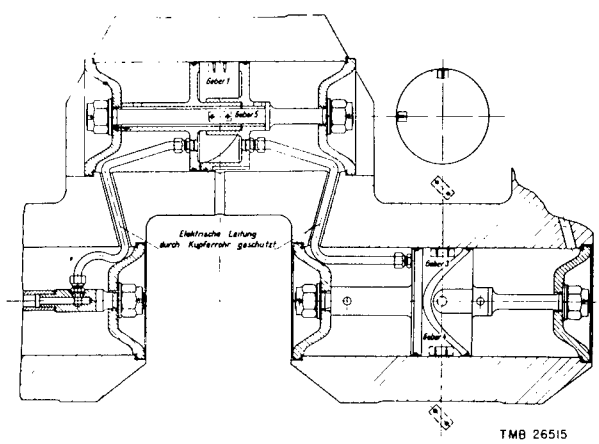


Figure 17 Gage Installation on a Crankshaft

were installed inside the hollow crank pins to measure torsion. Leads were conducted through copper tubes, along the lubrication borings to slip rings at the end of the crankshaft.

The director of the MAN Research Institute, Dr. Lehr, believed that tests under actual operating conditions were the source of real design information. Strenuous efforts were made to obtain this information and were exemplified by the tests briefly described.

DISCUSSION

Question: Were the Germans able to reduce all the data that they accumulated on these tests to an analytical method, to enable engineers to design proper crankshafts from them?

Answer: Perhaps not completely, but they have always made efforts to get to that point.

Question: Has anyone here been able to do this?

Answer: Not to my knowledge. As far as the Germans are concerned, I know very little about what was done in this direction at other laboratories. The laboratory we are discussing was really under Dr. Lehr, who was the Director. He is the man you know as the writer of the book on Vibration Techniques. Dr. Lehr believed in giving equal emphasis to all parts of the work. He was the real guiding spirit; all others were helpers. He was a man of great energy, and wide interests in development of instruments, in analysis and in dissemination of information. He even lectured the Army on how to deflect spring suspensions. Even after taking all these measurements, they still have to cut and try to design a suitable crankshaft.

Question: Do they know precisely which way to go as a result of these measurements?

Answer: Previous experience showed them the general direction to follow. In that way there was a choice of two or three possibilities. Without any experience and with random guessing, you would have to eliminate 90 percent of the tried factors before you could arrive at a solution.

MR. PRIEBE, of Squier Signal Laboratory asked, "Have you any records which were taken from this instrumentation?"

Answer: I saved about 90 reports which include results on some of these tests and reports on laboratory tests - both statistical and dynamic laboratory tests on components. Those can be made available by Commander Radford. We have finally come to the point where we are finished with them, and Commander Radford is the repository.

MR. TREITEL, NRL - The instrument problem comes up again. Can you tell us a little about German slip rings - the materials used, the speeds at which they operated, and the magnitude of the signal from the pick-up device?

Answer: I'm going to stick my neck out and hope my memory is correct. The particular method of wiping they had made uniform resistance. The total order of resistance was a tenth of an ohm. The slip ring noise was less than one percent for normal signals. They used silver-alloy rings. I don't remember the composition of the alloy, but it was very similar to the last alloy used by the Model Basin. The magnitude of the signal from the pick-up device was of the order of millivolts.

DR. TRENT, NRL - Did you manage to bring any of this equipment to the States?

Answer: Yes, we brought over quite a lot of equipment and set it up in this country.

MR. CUNNINGHAM, NRL - Why did the Germans avoid the use of the vacuum tube?

Answer: Because Dr. Lehr, of this laboratory, wanted an instrument that could be used under all conditions and had probably had trouble with vacuum tubes. Before the laboratory was established in 1937, when he was still teaching in Berlin, he did work for a time trying to develop the capacitance type of strain gage. He gave up after a while and started using the inductance strain gage.

DR. TRENT, NRL - In evaluating this tube situation, we must remember that the Germans had practically only one tube - the RV12P2000. It was the only tube in Germany. They were not in as good shape as either the British or ourselves in the development of tubes.

MR. THOMPSON, NRL - Since this strain gage is a magnetic device, was the sensitivity of the gage a function of attachment?

Answer: It was felt that magnetic leakage was no cause for concern. However, the first question we raised with the engineers was, "What is the gage length as influenced by the attachment to the device?" They thought that heat treatment prevented changes in the length of the gage. For soldering they had a special alloy which melted at 450 degrees F., but this alloy was almost as good as silver solder. They felt that it did not cause enough effect to change materials in the vicinity of the gage and thought the heat was small enough to matter. We asked how the conical mounting pins in a circular seat influenced the length of the gage. They did not have much of an answer. At that time Dr. Lehr was not living.

EMPIRICAL TESTS ON UNITS OF PORTABLE ELECTRONIC EQUIPMENTS

By

F. F. Koptonak

Watson Laboratories

A description of a series of empirical tests on units of land-based, portable electronic equipments and a discussion of the correlation between field and laboratory test failures are presented. The relative merits of rigid and flexible chassis construction are discussed. Types of equipments used and the effects of field and laboratory tests are illustrated.

Almost all Army and Navy contracts for electronic equipment contain a clause requiring that the gear be capable of withstanding the mechanical shocks and vibration encountered in field service. To measure the conditions which equipments transported over the land must withstand and to tend towards the establishment of uniform test requirements, the Army and Navy entered into a joint research program. The data here presented covers a phase of the program. It describes a series of empirical tests made on units of land-based, portable electronic equipment to determine the correlation between field test failures and laboratory test failures and to determine the relative merits of rigid and flexible chassis construction.

To make the empirical information gained as useful and as representative as possible, rigid and flexible units were tested, with and without transit cases, in various types of vehicles. Unfortunately, it was generally impossible to test more than one equipment of a given type. Therefore, the results indicate broad trends and do not provide detailed data on any given equipment.

Laboratory tests conducted on various units were those employed by many agencies to indicate a check on the ability of equipments of this type to meet shock and vibration requirements. Those for shock were the standard Navy shock tests; 1, 2, and 3 foot blows (per spec. RE9284B given on the HI shock machine for lightweight equipment). For vibration, on Army test procedures the equipment was vibrated in three mutually perpendicular directions, with the frequency cycling from 10 to 55 cycles per second and vertical amplitude being 1/64 inch at 15 cps and 3/128 inch at 55 cps. Crosswise amplitude was 1/64 inch at 10 cps and 1/32 inch at 55 cps. Vibration continued for two hours in each of the three directions.

On Navy vibration test procedures, the equipment was secured to a table and vibrated for one minute at each discrete frequency between 23 cps and 55 cps, with an amplitude of 0.005 inch. Also, the equipment was secured to a table and vibrated for one minute on each separate frequency between 10 cps and 55 cps, with 0.03-inch amplitude from 10 to 33 cps and 0.008-inch amplitude from 33 to 55 cps.

Field tests conducted were of the following nature:

1. Unit placed loosely in an unloaded 1-1/2 ton 6x6 Army truck which was driven over rough roads in the vicinity of Shark River Hills, near Belmar, New Jersey, the duration of the drive was 1-1/2 hours.

2. Unit was placed loosely in its transit case on the floor of a 1-1/2 ton Army truck and first driven over an old golf course for about 30 minutes at a speed of about 5 miles per hour. The most severe part of this run was when the truck drove over a two-foot-high bunker four times, causing front wheels and then rear wheels to leave the ground. The truck was then driven over a corduroy road at about 25 miles per hour for 30 minutes. This produced a discernible high-frequency vibration on the floor of the truck. Then the truck was driven over a road that was particularly rough, containing several cross gulleys made by rains. During this run (lasting 1-1/2 hours), the passenger in the front seat had considerable difficulty "holding on" and felt very much jounced at the end of 30 minutes. The equipment bounced into the air about six inches and fell back to the floor of the truck about eight times for each round of the course. It took about 10 minutes for each round.

3. Unit without case was placed in a cardboard carton without packing and shipped via Naval Air Cargo Express from Washington, D. C. to Oakland, California and back to Washington, D. C.

4. Unit in transit case was pushed off the 45-inch tailboard of a truck to a hard gravel road three times. Cathode ray tubes were removed.

5. Unit not in its transit case was dropped three times from a height of 55 inches to a hard gravel road. Cathode ray tubes were removed.

6. Equipments were placed loosely on the back of an unloaded 3/4-ton 4x4 truck, weapons carrier, and driven several times over the Belgium Block road at 18 to 20 miles per hour, then over the occasional bumps at 15 miles per hour, and then over the washboard road at a speed regulated to coincide with the vehicle spring resonance.

7. Again, equipment placed loosely in the back of a 1-1/2 ton 6x6 truck, and the truck then was driven several times over the Belgium Block road at 22 to 26 miles per hour. Under these conditions, the equipment bounced into the air about six inches, although the truck wheels did not leave the ground. (The acceleration of the floor of the truck exceeded one g, because the weight of the wheels caused the springs to pull the chassis down whenever it bounced resonantly.) The truck was then driven over the occasional bumps at about 23 miles per hour. On this run, the rear four wheels of the truck left the ground when going over the bumps, and the equipment bounced about 10 inches in the air. The passenger found it was much more comfortable to stand, during these tests, than to sit on the car seat mounted in the back of the truck. At one point, one unit (BC-606D Radio Receiver) bounced 18 inches into the air.

8. Then equipment was placed loosely in the cargo space of various Army vehicles and driven 1624 miles over paved roads, 720 miles over secondary roads, and 824 miles cross country over the Wirsing Area of the proving grounds.

The various units tested included the TS-34/AP Oscilloscope, the TS-34A/AP Oscilloscope, the TS-35A/AP Signal Generator, the BC-1306 Radio Transmitter-Receiver, the BC-603D Radio Receiver, and the BC-221AE Frequency Meter. Figures 1 through 13 present more detailed accounts of the types of equipments used in the various tests and indicate the types of failures which occurred under field and laboratory tests.

An analysis of the data accumulated on these tests indicates that:

1. Some of the damage caused by shocks received in the field is duplicated by laboratory tests on the HI shock machine and by low-frequency vibration tests, 10-33 cps.

2. Almost none of the damage caused by field tests is duplicated by high-frequency vibration tests, 10-55 cps, which have the additional disadvantage of causing many failures that do not occur in the field.

3. The units with strong, rigid chassis were damaged less than the units with flimsier, flexible chassis.

Most notable about all the tests is the fact that very few vacuum tubes failed (ruggedized tubes were not used), and usually the units would operate electrically, either in spite of the mechanical damage or, at least, as soon as simple repairs were made. Electron tubes which were not held in place with retainers tended to come out of their sockets during both field and laboratory shock tests. The electron tubes that broke generally did so because they collided with other items.

Neither the truck nor the airplane vibrations caused nearly as much damage to to the equipment as the shocks caused. The only damage sustained by the unit shipped across the continent and back again via air cargo express was shock damage. On all equipments tested, only one pigtail-mounted piece of apparatus broke loose on the field tests, whereas many of these items broke loose on high-frequency tests.

Units receiving both HI shock tests and low-frequency vibration tests exhibited a total of 23 field failures. Thirteen of these were duplicated by the laboratory HI shock and lower frequency vibration tests. Units which were subjected to the high-frequency, large-amplitude vibration tests exhibited 26 field failures. Only two of these were duplicated by the high-frequency vibration tests, which also caused 28 failures that were not duplicated in the field. This data seems to indicate that the high-frequency vibration test is of no value as a means of detecting weaknesses in the equipment which might result in field failures. The HI shock test and the low-frequency vibration test do not seem to provide an entirely satisfactory screening test. The tumbling that the equipment receives when transported for long distances subjects it to shocks from every direction as it bounces into the air and falls in almost every conceivable position.

Further analysis revealed that the amount of damage which occurred in the field increased as the duration or severity of the field test increased. For example, the TS-34/AP Oscilloscope had one failure when it was dropped three times from a height of 48 inches. On the other hand, two failures occurred when it was transported for a short time over reasonably rough roads. Five failures occurred on the two short trips over very rough roads. Finally, eight failures occurred on the long, 3000-mile test at Aberdeen. The degree and amount of tumbling which the equipment received was an important factor governing the number of failures that occurred. It is apparent, then, that a few controlled drops do not make a satisfactory screening test any more than a high-frequency vibration test makes a satisfactory screening test.

The strong, rigid equipments withstood field tests better than the flexible, weak equipments did. This occurred, principally, because the designers did not make proper allowances for the large excursions which flexible elements have under shock excitation. None of the structural deformations interfered with the electrical operations of the equipments.

The data covered indicate the desirability of subjecting units of portable electronic equipment to laboratory tests which more nearly simulate field conditions. The ideal laboratory machine, for simulating vehicular transport, would provide vibrations of about five inches amplitude at about three cycles per second. Superimposed on these would be high-frequency vibrations of only a few mils amplitude. These would approximate the high-frequency, transient chassis vibrations superimposed on the fundamental spring vibration of Army vehicles. Equipment placed loosely in the cargo spaces of land vehicles would be placed unfastened on the platform of this vibration table. It would then receive a tumbling test that simulates field conditions quite closely. Equipment ordinarily secured rigidly to the vehicle would be secured rigidly to the vibration machine. It would be subjected to rather slowly applied accelerations of the type that occur in the field when vehicles go over rough roads.

Since it is advisable, wherever possible, to utilize existing testing facilities instead of specifying new ones, a departure from this rule appears warranted in the case under discussion. Both the field measurements described in Squier Laboratory reports and the empirical tests described here indicate that existing vibration and shock test requirements for land-based, portable electronic equipments require some consideration for change.

DISCUSSION

MR. CUNNINGHAM, NRL: I feel that the type of shock tests described were not realistic. I doubt the validity of such tests. For example, in field tests, would a service man be justified in allowing the equipment to bounce six to ten inches -- wouldn't he lash it down?

MR. KOPTONAK, WATSON LABORATORIES: The field abuse ordinarily encountered will not generally approach that which I have described. In view of these conditions, we realized that some of the tests amplified failures. The intent of this test was not overlooked, and the actual field conditions may reach the state outlined.

MR. PRIEBE, SQUIER SIGNAL LABORATORY: These are actual field conditions. Under combat conditions, the operator does not have much time to secure all the gear. In combat areas you have those wide terrains with shell holes over which you have to drive with equipment in a hurry.

DR. VIGNESS, NRL: In testing on a light-weight machine, it seems surprising that anyone would expect a piece of equipment to pass this sort of test where you have multiple repetitions of small shocks. The light-weight machine and the shock tests which you have seen in Building 10 are intended for a small number of shocks -- not a large number of small shocks, but a small number of rather large magnitude. To screen out the equipments that failed due to a very large number of multiple shocks in the back of a truck requires another type of test differing from that given in the Navy.

MR. LEE, COLES SIGNAL LABORATORY: Let me comment on the light-weight impact machine. We have been using it mostly as a matter of correlation with the dropping test. You know how difficult it is to get reproducible drop tests. We have used it on about 15 tests. It is an excellent quality controlled shock. In one particular case we detected a number of tubes badly cemented into sockets. This could not have been found unless the impact test machine were used. Of course, we must use judgement in interpreting re-

sults, but it does point out things that are bad practice.

DR. VIGNESS: Any way in which equipment is treated in a rough manner is generally a screening test. The impact type test which we have in the high-impact type machines here is pretty representative of rough treatment that equipment might get in service. It just does not do everything, but it does very well in certain cases. It is something that we can repeat -- something we can put into specifications. It might be well to use that, even though it doesn't simulate all types of practice.

MR. WALSH, NRL: I think that the Army people should be congratulated for their work and their very forthright way of getting at the problem. I should like to ask the author to review relative failures they got on field tests versus laboratory tests. Did any failures show up in the laboratory which did not show up in the field?

MR. KOPTONAK: As pointed out, we listed the comparative numbers of failures obtained through field conditions and then resorted to an indication of the numbers of failures on Army or Navy vibration high-impact shock tests. When we compared them, shocks obtained through simulated or laboratory methods exceeded those in the field. Speaking of numbers, I could compare one or two equipments. The Oscillograph showed 23 failures in laboratory tests, while in field tests it ran to 16. Further analysis is required for classification of types. We did not obtain detailed descriptions of the types. Only overall consideration of failures was evaluated.

MR. WALSH: It would seem that the type of failure is the important point, not the numbers of failures. If in the laboratory tests the breakage of components is different than in the field, then the type of component that fails is important.

MR. KOPTONAK: It would be so considered. That was taken into account in comparing relative numbers of failures due to field and laboratory conditions.

On pages 74 through 79 appear illustrations which were not referred to directly in the text of this article, but which supplement the material contained therein.

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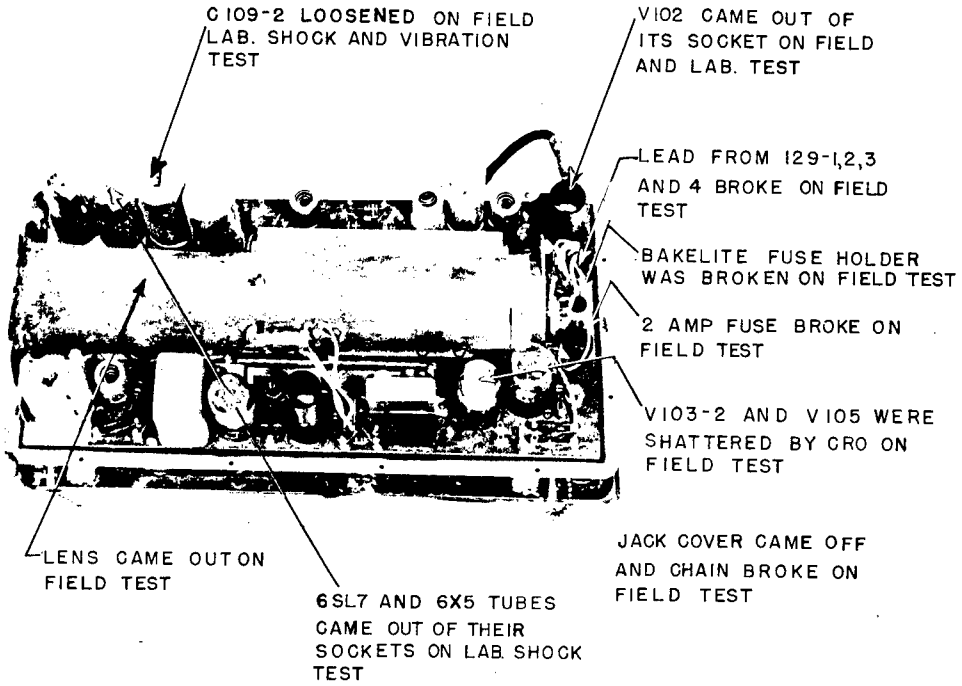

Figure 1a TS-34/AP Oscilloscope
This is a rather flexible .050 inch thick mild steel chassis containing many large cut-outs and having rather flexible component mounting panels attached to it. The apparatus side of the chassis is shown.

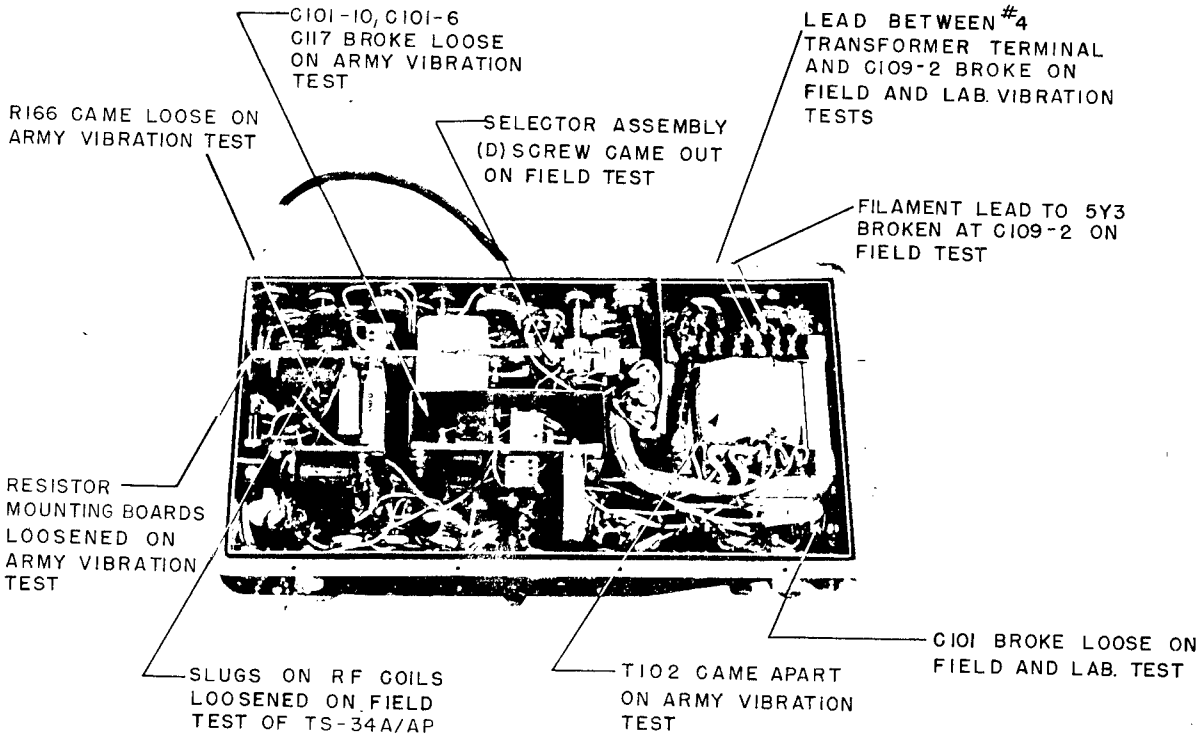

Figure 1b Oscilloscope Wiring Side of Chassis with Cover Removed.

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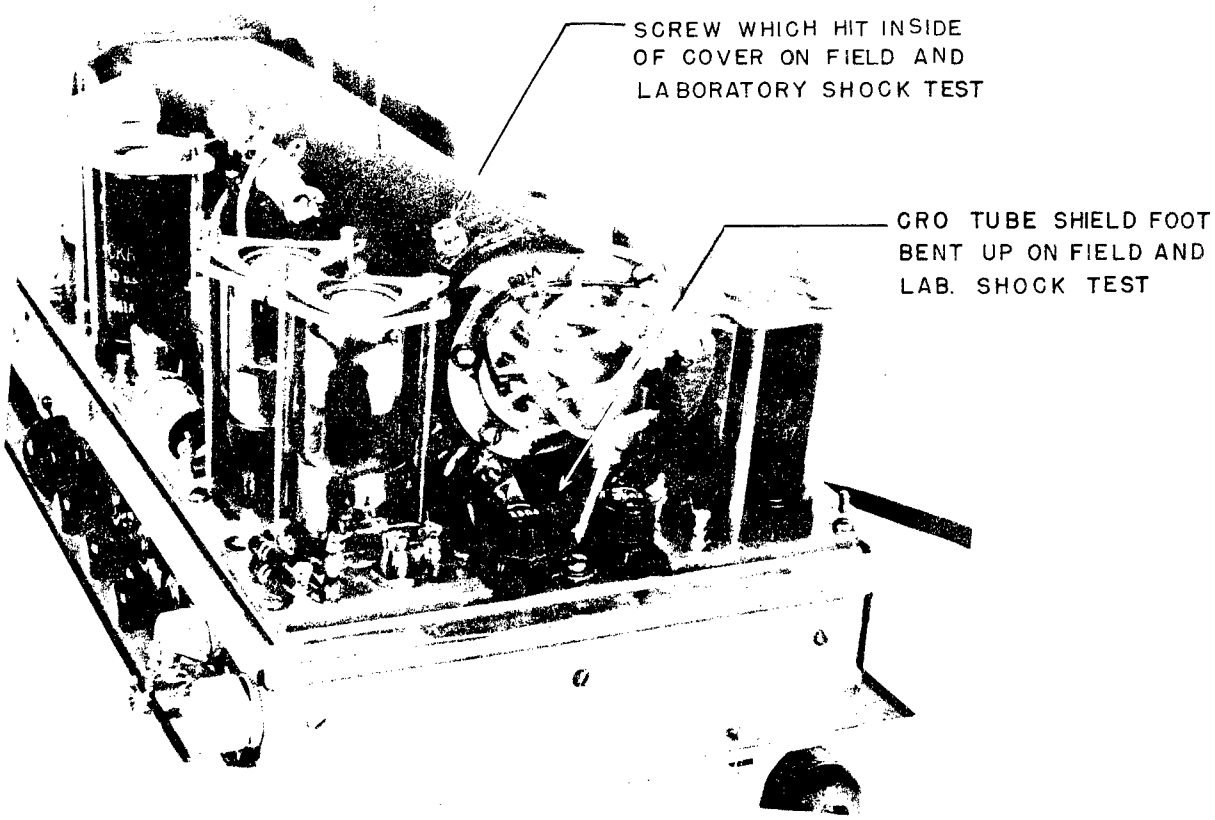


Figure 2a TS-34A/AP Oscilloscope
This is a ruggedized version of the TS-34. The chassis is reinforced with flanges and around the outside of the flimsy shielding cover is a strong louvered cover. This unit was tested in the transit case (Fig. 2c) shown on the next slide. This case is made of .040 inch aluminum and has no padding in it. Retainers are provided for all tubes in this unit.

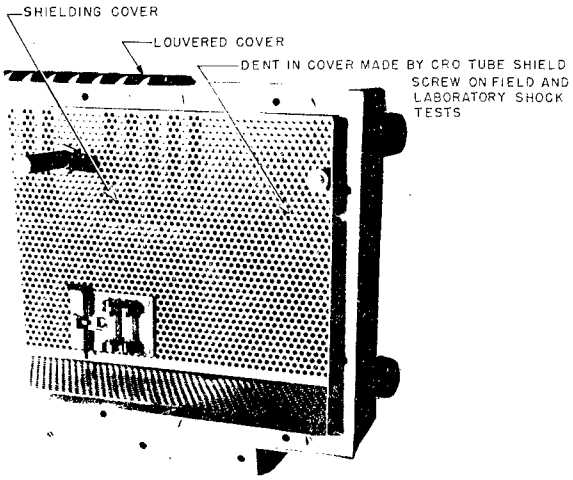


Figure 2b TS-34A/AP Oscilloscope Cover

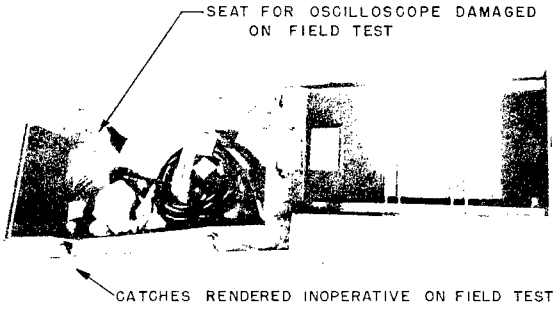


Figure 2c TS-34A/AP Oscilloscope Transit Case.

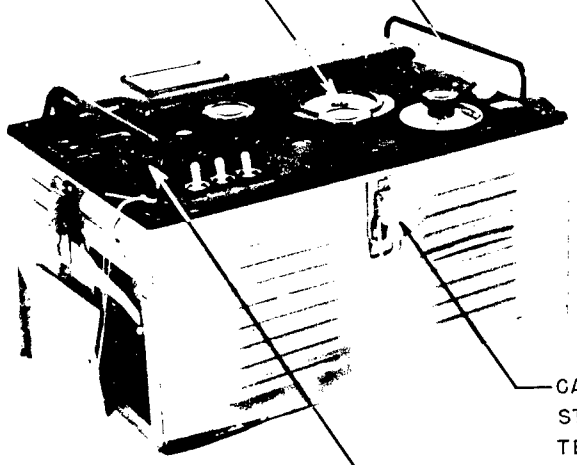


Figure 3a This unit has a rather flexible 1/16 inch aluminum chassis which is spot welded together. It also includes a section of wave guide.

Figure 3b The cover of this unit is its carrying case. Retainers are provided for all tubes.

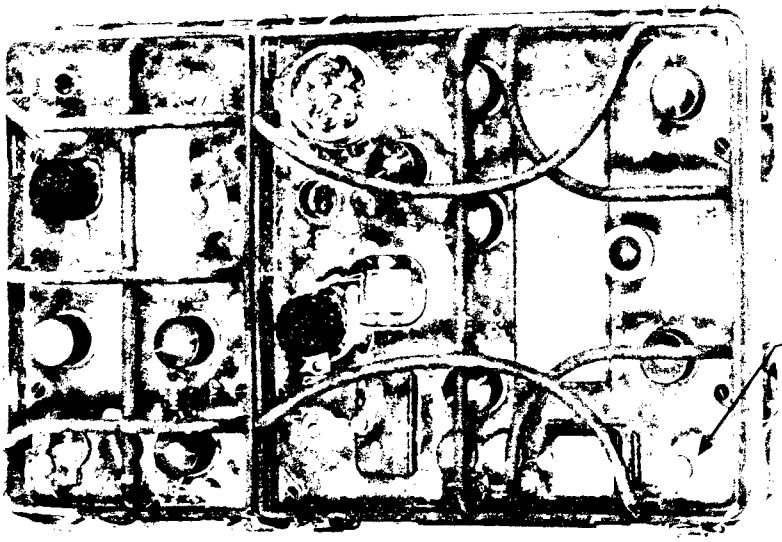


Figure 4a BC-1306 Radio Transmitter Receiver.
This unit has a thin aluminum chassis which is very rigid because it is small in size and because the mounted components are light in weight.

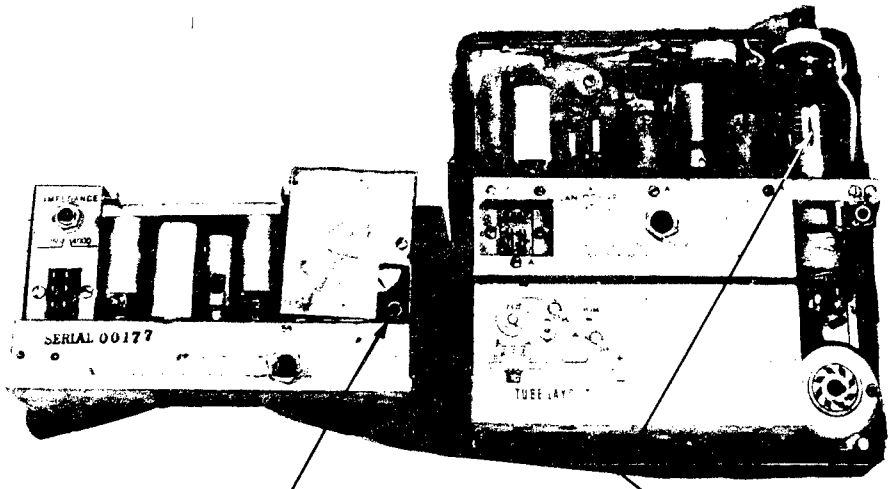


Figure 4b The cover for this unit forms the transit case. Retainers are provided for all tubes.

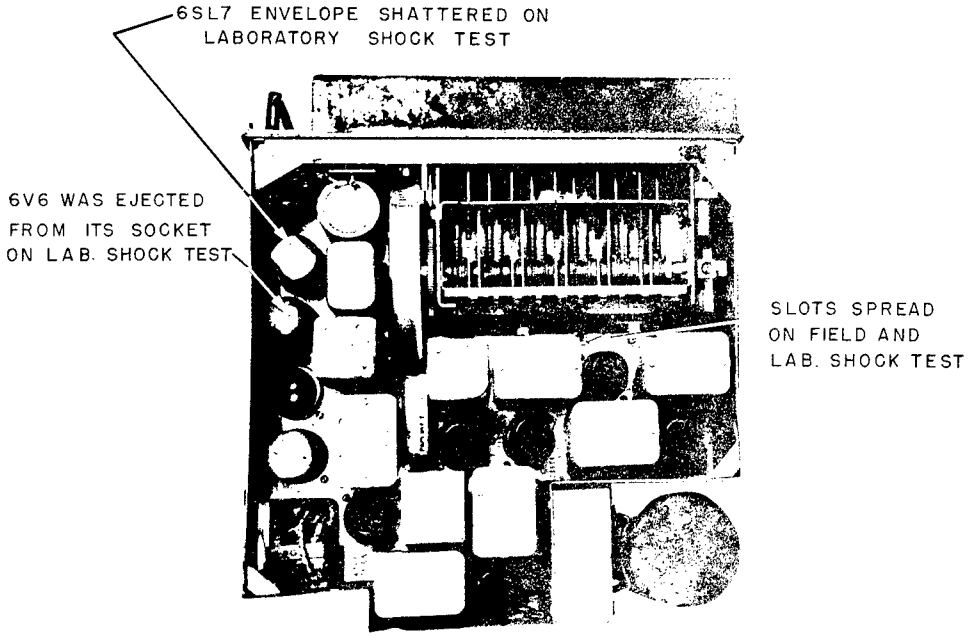
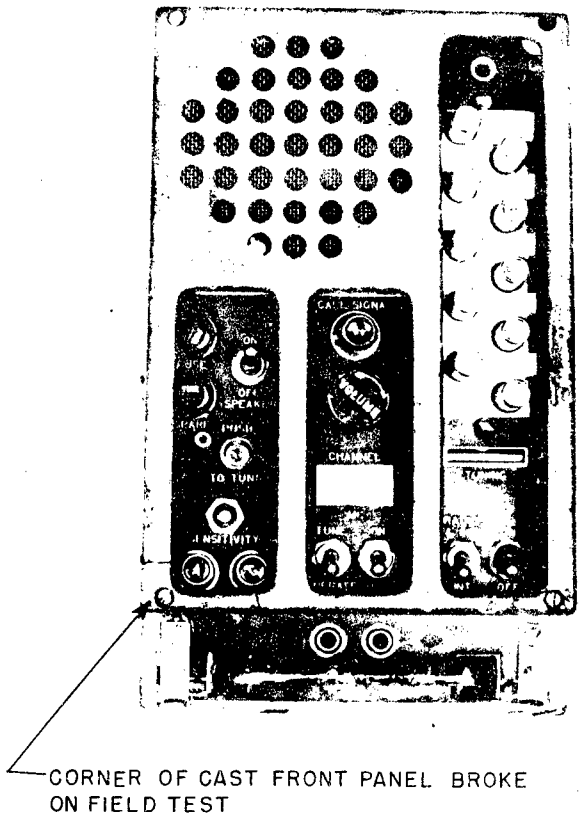


Figure 5a This is one unit of a tank set. It is rigid with a casting for its front panel and a 3/64 inch thick sheet steel cover and chassis.



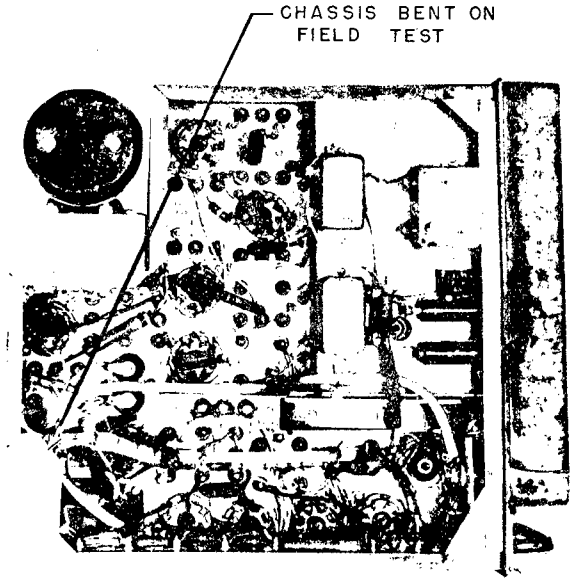


Figure 5b BC-603-D Tank Set, Wiring side with cover removed.

← Figure 5c Although it is ordinarily supported on shock mounts, it was tested as a portable box of electronic gear without any mounts. No electron tube retainers are provided in this unit.

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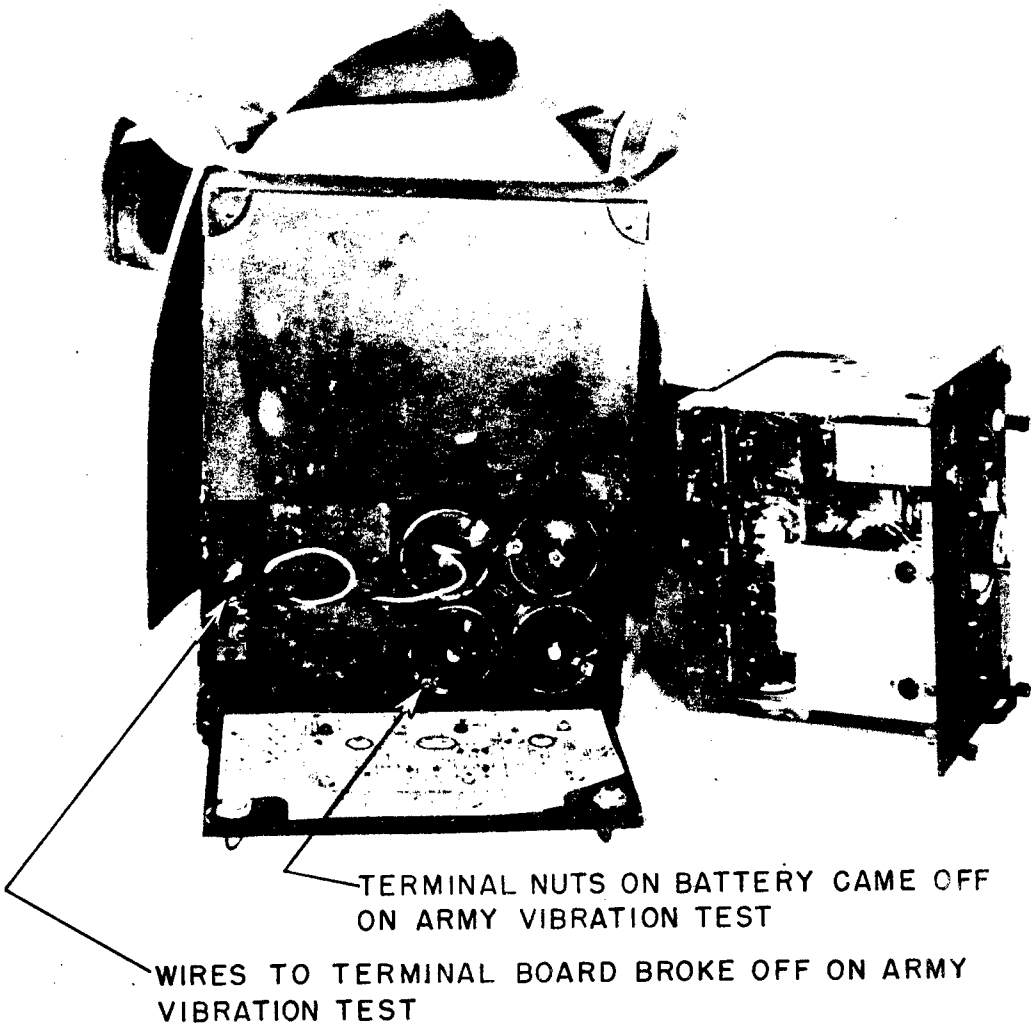

Figure 6 Dry cell batteries are kept in the bottom compartment of the case. Retainers are provided for all tubes.

VIBRATION AND SHOCKS ENCOUNTERED BY MINES DURING SHIPMENT IN A RAILROAD FREIGHT CAR

by

R. H. Suessle, NOL

In order to prevent detonation of mines during shipment in railroad cars, test instruments were placed in some of the mine cases to study the shocks occurring in transit. These data are discussed.

One phase of the Mine and Depth Charge Program at NOL during the war was to make certain that any weapon released for use by the fleet would be undamaged during its shipment to the fleet. Primarily, the mine or depth charge had to be perfectly safe at all times. Secondarily, it had to be in a working condition when it reached its destination.

The quickest and most practical way of assuring that a weapon was rugged enough to withstand field vibrations was to subject it to a laboratory vibration or shock test which was more severe than any vibration or shock to be expected in the field. Obviously, to devise adequate laboratory tests, vibration conditions aboard various transportation carriers had to be determined. Data of this type were sparse and in most cases non-existent. The laboratory had to take it upon itself to measure field conditions.

Early in 1943 the Naval Ordnance Laboratory continuous vibration recording equipment was assembled. There were several factors which guided the selection of the type of equipment employed. First, the equipment had to be assembled rapidly because of wartime urgencies. Second, it had to be versatile, since measurements were contemplated aboard trucks, trains, planes, naval vessels, etc. Third, a permanent record of the measurements was desirable. And fourth, it had to be portable.

The equipment consisted of the following units, as shown in Figure 1: three Shure Type 61B Rochelle salt pickups; one 60-cycle calibrator; three General Radio Vibration Meters, Type 761-A; one three-channel oscilloscope; one General Radio moving picture camera; and a power supply. The principle of operation of the equipment was to detect and amplify the output of the three Rochelle salt crystal pickups and photograph the cathode ray traces of these outputs.

The three crystal pickups were mounted rigidly to the object on which measurements were to be taken, so that vibrations along the three major axes were recorded. The output of the crystal pickups was fed into the three General Radio Vibration Meters. Since most vibration signals encountered had complex wave forms, the Vibration Meters were used only as amplifiers and integrators and never as direct-reading instruments. The use of the General Radio Vibration Meters saved valuable time which would have been spent on the construction of amplifiers to match the crystal pickups.

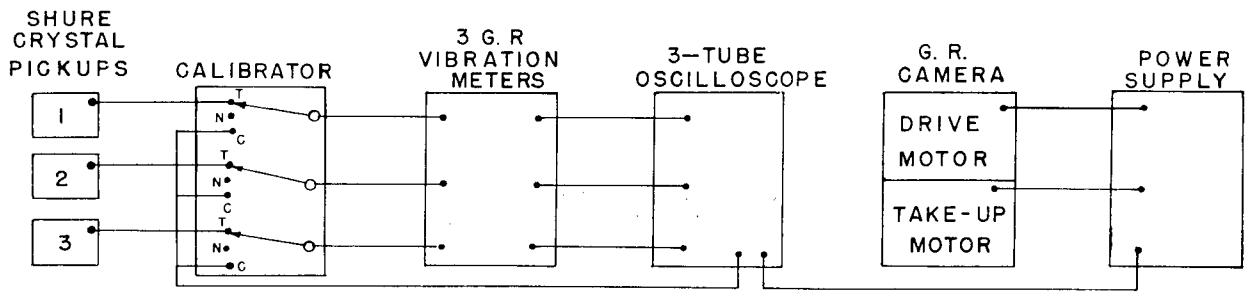

Figure 1 Block Diagram of Continuous Vibration Recording Equipment

Calibration was provided by a 3-volt, 60-cycle signal taken from the filament transformer of the oscilloscope. The calibrator had a three-point selector switch. Position T completed the circuits from all three pickup units to the oscilloscope for signal recording; position N left all three circuits open so that a microphonics reading could be taken; and position C completed the circuit from transformer to the oscilloscope for a calibration trace.

The three-channel oscilloscope was designed and built at NOL. It contained three 2" AP-5 Cathode Ray Tubes and a direct current amplifier for each. This oscilloscope had no sweep circuit, and any signal appeared on the screen as a horizontal line. The oscilloscope also contained a relaxation oscillator which flashed a 1/25-watt argon tube. This provided the time base for frequency measurement.

The shutterless camera was a continuous-recording, motor-driven type using 35 millimeter film, on which were focused the three oscillograph traces and timing signal.

In May 1945, the vibrations encountered by mines transported by railroad freight cars were measured during a trip on a standard freight train from Washington, D. C. to Rouses Point, N. Y. The recording equipment was installed in a Pennsylvania Railroad caboose, and the mines were placed in a standard box car directly ahead of the caboose. The necessary leads and cables were connected to the instruments in the caboose through a hole in the box car. The freight car and caboose used for the test were just ahead of the caboose of the train crew during the entire trip.

The five mines were blocked in place along one side of the car from the door to the rear, with their longitudinal axes parallel to the transverse axis of the car. The three crystal pickups were mounted in the mechanism compartments of the mines. From Washington, D. C. to Carbondale, Pennsylvania records were taken with the pickups mounted in the end mine, which was located over the wheel. From Carbondale, Pennsylvania to Rouses Point, N. Y. records were taken with the pickups mounted in the mine nearest the door of the car.

A Cambridge Accelerometer was also attached to the tail plate of one of the mines. The two directions of shock recorded with the accelerometer were the vertical and fore and aft directions with respect to the train. The maximum acceleration recorded in the vertical direction was 4.8 g, while the average acceleration ran about 3.2 g. The maximum acceleration recorded in the fore and aft direction was 14.6 g, and the average was about 5.6 g.

The predominant frequencies recorded in the vertical direction were 2400 cpm and 236 cpm. The former was undoubtedly

the natural frequency of the freight car bed, and the latter was the natural frequency of the spring suspension of the car. The springs of the freight car were designed to have a natural frequency of 125 cpm when fully loaded. Assuming this value, the natural frequency of the car as loaded for this test would be approximately 200 cpm. This compares favorably with the 236-cycle average frequency recorded. The maximum acceleration recorded at this low frequency was 1.4 g peak to peak, and the average was .74 g.

Table I

VIBRATION OF MINES IN FREIGHT CAR

	FORE & AFT	CROSSWISE	VERTICAL
AVERAGE FREQUENCY	2630	1170	2400
AVERAGE ACCELERATION	.26	.35	.70
MAXIMUM ACCELERATION	.73	.80	2.04
FREQUENCY	3500	1700	2230
MAXIMUM FREQUENCY	3900	1770	3300
ACCELERATION	.39	.46	1.50

Frequencies - Cycles per minute
Accelerations - G's (peak to peak)

The directions of vibration referred to are with respect to the train. In almost all cases, the vibrations measured in the mine located over the wheels were more severe than those measured near the center of the car. All accelerations listed in Table I are peak to peak.

Measurements similar to those taken on the freight car have been made on some truck types, destroyers, PT boats, and various other transportation carriers. The results obtained are by no means comprehensive but have been valuable in laboratory vibration tests.

DISCUSSION

MR. N. H. JASPER, DTMB, asked: "On the General Radio Vibration Meters, did you have any trouble with drifting of calibration with change of sensitivity?"

Answer: A calibration trace was taken directly before each reading. That was the reason for installing a calibrator in there.

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GENERAL DISCUSSION

DR. KLEIN: Are there any points to be brought up for the general agenda?

DR. TRENT: I would like to make some remarks concerning the Standards Committee. The work this committee has done so far is published in the Bulletin, and we have asked for constructive criticism. I would like to report that the number of comments which have come in so far is extremely small. We thought we would really get in your hair and you would be mad at us. Please do look these recommendations over carefully and if you have any criticisms, we should like to have them as a basis of deliberation before the next meeting.

DR. KLEIN: The Agenda Committee would like suggestions from you regarding the next meeting. Tentatively, it is to be a discussion of shock and vibration pertaining to aircraft and land structures--shore buildings nearby. The next meeting would, in a way, conclude our survey work. We have been reviewing and summarizing the field of shock and vibration in general. With this review, our respective programs should be oriented more effectively.

Are there any suggestions or comments about the symposia--past or future?

MR. JASPER: I should like to inquire about further discussion on shipboard vibrations.

DR. KLEIN: According to the Definitions and Standards Committee, shock is a form of vibration and both were included in the earlier reviews. To be sure, the subject of shipboard vibration was not discussed at length; all we are trying to do here in these symposia is to summarize what we know now and where we are going--by merely touching these various fields on the surface.

MR. LEE: This may be out of order, but do you have any policy regarding having representatives of commercial firms attend the symposia?

DR. KLEIN: So far, our invitations are limited to the government agencies and to those organizations which have contracts with government agencies in the field of shock and vibration. However, we are trying to work out procedures whereby certain concerns in this field may be invited to our meetings to the mutual benefit of the government agencies as well as the industrial firms.

The Centralizing Group has been asked by several panels of the RDB Committees to state our function. They seem to be satisfied with the progress we have made during our existence so far. Perhaps Dr. Stephenson would like to say a few words about the RDB organization to acquaint you with this new setup.

DR. STEPHENSON: I think you are all familiar with the fact that there is a research and development board which reports directly to the Secretary of National Defense. This board is relatively new and the various committees are working out a system to make it effective. In general, the committees are broken down into panels and these, in turn, may be divided into subpanels or sub-committees. There is a continuous problem as to the assignment of a particular line of work under a committee or panel. At present, shock and vibrations is in the program of the Acoustics Panel of the Electronics Committee.

The question may be asked, should shock and vibration logically be placed under acoustics or a mechanics committee? Unfortunately, none of the present committees specifically cover mechanics so the immediate question is, shall shock and vibration be continued under the Acoustics Panel until a Mechanics Panel or Committee is established?

Such a mechanics panel or committee may well be established if there is sufficient demand for it. From my personal experience, I would say that it is desirable for any group to be definitely assigned to a panel, and that one that may not be considered ideal is still better than none. The reason for this statement is that RDB will soon have a strong influence on the budget for all the armed services and that it is highly desirable to be sponsored by some committee. As a member of the Acoustics Panel, I think I can assure you that your interests will be represented.

Mr. Walsh stated that he felt shock and vibration was definitely applied mechanics and should be soon assigned.

Dr. Stephenson suggested that some educational work might well be done to convince RDB that the research and development programs in the field of mechanics should be recognized by the establishment of an RDB committee for that field.

THE MULTIFREQUENCY REED GAGE

Vibration Section Staff
Taylor Model Basin

The multifrequency reed gage is an instrument whose function is to make possible the analysis of a given shock motion from the deflection of a series of reeds of different natural frequencies.

The gage consists of a set of reeds of different natural frequencies rigidly fastened as cantilevers in a heavy frame. The frame, in turn, is fastened rigidly to the structure subjected to shock motions. The reeds have masses at their extremities to which styluses are fastened to record the motion of the reeds on waxed paper. In the Type I multifrequency reed gage, (see Figure 1) the waxed paper remains stationary. The Type 2 multifrequency reed gage (see Figures 2 and 3) has an inertia-actuated platen to which the waxed paper is fastened. In the latter

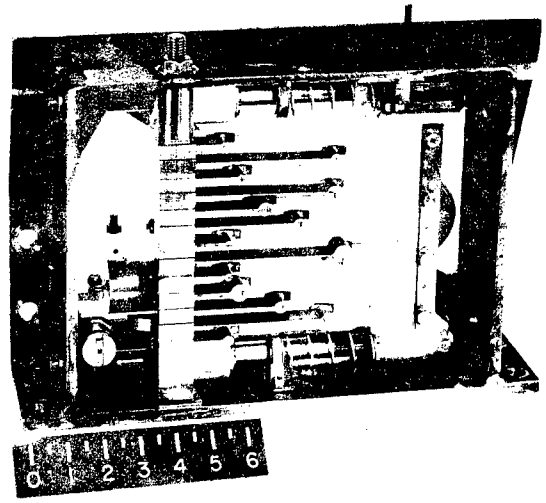

Figure 2 Type 2 Multifrequency Reed Gage - Front Showing Reeds

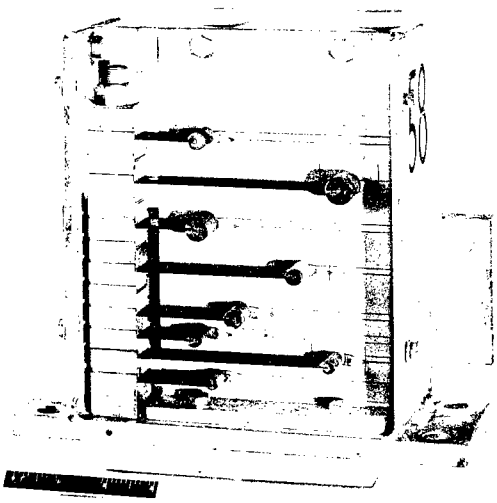

Figure 1 Type I Multifrequency Reed Gage

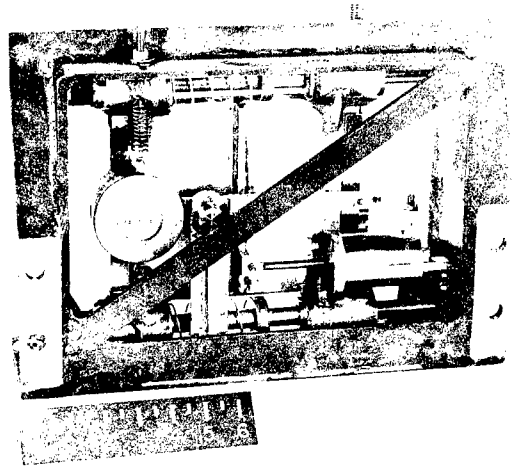

Figure 3 Type 2 Multifrequency Reed Gage - Back Showing Mechanism for Inertia Actuation of the Platen

gage the number of oscillations of the individual reeds can be observed.

Since the instrument was devised to measure shock motions, a discussion of the general characteristics of shock motions and of the adequacy of the reed gage in recording them is in order.

The Standards Committee has tentatively promulgated the following definitions:

Shock Motion: Mechanical shock motions exist in a system if the transient which occurs between two steady states produces significant relative displacements between points in the system.

Applied Shock: Any excitation which, if applied to a system, produces shock motions within the system is classified as applied shock.

It is generally agreed that the time duration of the specific motion considered is an important factor, as far as shock intensity is concerned, because:

1. A high instantaneous acceleration does not necessarily indicate high shock since high accelerations which last for very short periods can have little shock effect.

2. A high velocity does not necessarily indicate high shock; a high velocity must be acquired in a relatively short time to produce a shock effect.

3. A large displacement does not necessarily indicate high shock. The large displacement must take place in a relatively short time to have shock effect.

It is apparent that the reed gage cannot reveal all the components of a complicated shock motion. It appears upon analysis however, that at least for the types of shock motion encountered on shipboard, the reed gage records sufficient information for deducing the maximum velocity and the type of motion describing the shock motion. For most practical purposes this information is sufficient for determining the intensity of "shock".

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 with one degree of freedom. This assumption seems reasonable, since the second mode of a cantilever reed has six times the frequency of the fundamental and, except for the case of resonance, amplitudes in the higher modes should be small.

Under steady state conditions, it is known that the amplitude of a system of one degree of freedom depends on the ratio of the "frequency" of the impressed force to the natural frequency of the system. It is to be expected that the deflections of the various reeds will be unequal upon the application of an arbitrary force to the support of the reed gage. For the Type I reed gage the only data recorded are the maximum reed deflections. Thus, the problem becomes that of deducing the nature of the impressed force or applied shock from the magnitude of the reed deflections.

The mathematical equations for the theoretical values of the reed deflections have been derived for the following types of impressed motions frequently encountered in the study of shock motions:

1. Velocity
   a) Step velocity
   b) Rectangular pulse of velocity
2. Acceleration
   a) Step acceleration
   b) Rectangular pulse of acceleration
3. Simple Harmonic Motion
   a) $X = X_0 \sin \omega t$
   b) $X = X_0 \cos \omega t$
   c) $X = X_0 (1 - \cos \omega t)$

These equations may be used to evaluate the shock excitation imparted to the single-degree-of-freedom system. In the case of the Type II gage the user must be determined whether the deflection traces of the various reeds simulate any of the theoretical deflection-time curves computed for one degree systems of corresponding natural frequencies. If the recorded curves correspond to the standard types of impressed shock motions except for a scale factor, then for all practical purposes the shock motion may be defined by multiplying the ideal displacement-time, velocity-time, or acceleration-time function by the appropriate scale factor.

With the Type I gage the process is more complicated. Since only the peak deflections of the reeds are known, the nature of the impressed motion is not revealed by the deflection of a single reed. In this case it is necessary to tabulate the reed deflections. These tabulated values are then compared with the peak values tabulated from the theoretical curves corresponding to various types of impressed motions.

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INTRODUCTORY REMARKS ON SHOCK INSTRUMENTATION

By

I. Vigness, NRL

PROLOGUE

This discussion will attempt to answer partially some common questions that are often in the minds of those involved in problems relating to mechanical shock measurements. In a qualitative manner, it will cover the following:

1. What must be measured in order to determine shock motions adequately?
2. What are the usual types of measurements?
3. Of what use are the measurements?

The paper is written in the hope that it may provoke thought and discussion that can be aired in a forthcoming Symposium on these problems, at which time these and other problems will be considered in detail.

SHOCK MOTION DETERMINATION

To determine completely the shock motions of a point, all that is required is that these motions be recorded accurately as a function of time. It is immaterial whether they be acceleration, velocity, or displacement. Generally, rotational components are not considered, so that the records involve only the translational motions along the three principal directions. Distortions and strains that occur in a region can be determined by shock measurements at several locations or by determining the relative motions between points of interest. As will be more apparent later, the type of instrument to be used depends upon the frequency range of interest, the amplitudes involved, the possible elaborateness of the setup, the other economic and special considerations.

AMPLITUDE AND FREQUENCY REQUIREMENTS

As a result of experience, it has been found that the mechanical vibration frequency range of most interest to the Navy with respect to shock damage covers, roughly, from 5 cps to 5000 cps. The most damaging part of this range is probably between 10 and 500 cps for most types of equipment. Also as a result of experience, it is known that when serious (but not disabling) structural damage occurs on board a ship (caused by non-contact underwater explosions), accelerations up to 5000 g (g = unit of gravity), velocities up to 40 feet per second, and displacements up to several inches may occur in the vicinity. It has been observed that the maximum accelerations expected vary about linearly with the frequency, up to about 5000 g at 5000 cps; the maximum probable velocity is nearly independent of frequency in the above range; the value for any one component does not generally exceed 10 feet per second; and the displacements vary inversely with the frequency, attaining a magnitude of several inches only at the lowest end of the above frequency range.

INSTRUMENTS AND THEIR SPECIAL CHARACTERISTICS

The instruments considered are those in the present state of development. They include only the types suitable for "high-impact" shock measurements and are grouped as: accelerometers, velocity meters, displacement meters, impact gauges, and tuned reed gauges.

Accelerometers - It can be stated with safety that an accelerometer is the most accurate and dependable type of instrument for high-frequency (above 500 cps) components of shock motion. Accelerometers can easily be constructed that have no natural frequencies below 20,000 cps and that give a good signal output, if one is satisfied with only high-frequency response. They have the following advantages:

1. They are light, small, rigid, and rugged.
2. They have no displacement limitations (such limitations are generally caused by low-frequency seismic masses bumping their stops).

They have the following disadvantages:

1. The crystal types have very high impedance.
2. As a result of the high impedance, spurious signals are often introduced by mechanical deformation of the cable connector.
3. Cumulative errors occur in integrated records that make derived velocity and displacement curves much less accurate than the acceleration curve for integrations over more than a short period of time.
4. It is not reasonable to expect an accelerometer to measure accurately both the low-frequency and high-frequency components of shock, when on the same record the high-frequency amplitudes may be 10 to 100 times as great as important low-frequency amplitudes.

Acceleration records are usually very complicated and, to the uninitiated, they often appear unreasonable, but when careful work is done, greater experience leads to greater confidence in their results. However, until more suitable connecting cables are obtained, the quartz crystal accelerometer should be regarded as unsuitable for normal field work. Rochelle salt type accelerometers, low-frequency strain gauge types, and many others designed for relatively low frequencies and amplitudes, have proven satisfactory for field work for their limited range.

Velocity Meters - Velocity meters operate on an electromagnetic principle; generally a voltage is induced in a coil of wire by the rate of change of magnetic flux. So, contrary to the accelerometer where there is nearly no relative motion between parts, the velocity meter generally requires considerable motion. This relative motion is between a seismic mass of long period and the main frame of the meter. The disadvantages of the available types suitable for shipboard use are:

1. They are heavy and bulky.
2. Their seismic element is usually limited to two inches displacement overall.
3. Their frequency range is limited from about twice their natural frequency (about 7 cps = natural frequency) to some indefinite value, generally somewhat greater than 1000 cps.
4. They possess a damping factor of unknown magnitude and nature.

For shipboard field work, however, the advantages of the velocity meter at present are outstanding - namely:

1. The impedance of the velocity meter is very low, and the output signal is very high.
2. It covers the most important frequency range between about 15 and 1000 cps without accentuating either high or low frequencies. (This is because the range of amplitudes of probable velocities is independent of the frequency.)
3. Accelerations or displacements, within the above frequency range, can be derived from the velocity records with a single differentiation or integration. This provides better accuracy for the derived records than would be possible if double differentiation or integration were required.

As a consequence, for field work, where weight is not of too great importance and where conditions for measurements are poor, the velocity measurements have predominated. For measurements on machines and equipment in the laboratory, acceleration measurements have often been made to cover the high-frequency range and displacement measurements to cover the low-frequency range.

Displacement Meters - There are a great variety of types of displacement measuring pickups. They extend from the techniques involving high-speed movies, streak photography, and photoelectric methods to simple slide wire voltage dividers. For field work, where no stationary reference is normally available, the displacements measured are usually relative, although a few seismic type displacement instruments have been built. In the laboratory, some of the displacement measuring techniques, particularly the streak photography and photoelectric methods, are capable of very great accuracy. However, for field work, they excel only for low-frequency work (up to a few hundred cps). Of course, the high-speed movie records show such a wealth of information besides the displacement of a single point, that this type deserves special consideration.

Impact Gauges - Impact gauges consist of various devices that (when subjected to an acceleration) crush, break, deform, or indent mechanical parts, or they may make or break electrical contacts. The contact-break types will respond to very short duration acceleration pulses (less than 0.0001 second) and, consequently, generally indicate higher values of peak acceleration than are obtained by other methods. The other types of these gauges respond best to accelerations of longer duration than 0.0001 second. The time is dependent upon the displacement required for their record and, generally, should give lower values than recorded by an instrument capable of recording to 5000 cps. In addition, the impact gauges are complicated by dynamic overshooting and by rattling after a deformation is made; however, dependent on the instrument, when an impact gauge is used, it definitely sets an upper limit, which value the acceleration does not exceed for a given frequency. Of course, the limit may be too high by several hundred per cent; nevertheless, the results may be of considerable value. A moral of this story is not to expect (maximum) values of acceleration as recorded by different impact gauges or accelerometers, under identical conditions, to be the same unless the frequency response of the instrument is known to be the same; and even when the frequency response is the same, the dynamic overshooting may cause considerable variation.

Tuned Reed Gauges - A reed gauge consists of a number of reeds, each of a definite frequency, and chosen to cover the frequency spectrum of most interest. It is usually assumed that a reed approximates a single-degree-of-freedom system and that the deflection of the reed by any shock disturbance is a measure of the "response" of a single-degree-of-freedom system to that shock motion. Thus, it may be said that the reed gauge analyzes the shock motion by measuring directly the probability of damage to simple systems as caused by the shock motion. The reed gauge has the following advantages:

1. It is simple, direct-reading, easy to operate, and requires no synchronization or timing problems and no auxiliary equipment.
2. It gives a directly useful answer.

The disadvantages are:

1. It is heavy and bulky.
2. Because of the small displacements of the high-frequency reeds, measurements of response for reeds having natural frequencies greater than about 200 cps are very inaccurate and may well be disregarded.
3. While some information can be interpreted from the reed gauge records as

to the nature of the shock function, it is inadequate for most purposes.

4. The reeds possess a large amount of friction damping of indeterminate nature.

OF WHAT USE ARE SHOCK MEASUREMENTS?

There has been criticism that many shock measurements have been made promptly stored away, with no study or analysis. What has happened is that the records were examined, sometimes rather superficially, and then put away. A large number of complicated curves are very unsatisfactory for discussion, so obvious amplitudes, frequencies, and other characteristics of the curves have been selected and reported. Exclusive of Crossroads, the amount of field shock measurement has not been large, but much useful information has been obtained.

Shock measurements are made for three principal reasons:

1. To determine magnitudes and other characteristics of shock motion so that machines can be built that will simulate these general magnitudes and characteristics.

2. To make possible the use of simple shock motions that can be devised and delivered by machines that can be made to simulate certain components of the field shock or be equivalent to certain phases of shock. This, of course, requires a knowledge of the general functions. Simple shock motions make shock possible an analytical study of shock effects.

3. To provide information for the designer which will allow him to intelligently design his equipment.

Thus far, the first factor has had most emphasis, the second factor has had a small amount of consideration, and the third factor has been largely neglected. Undoubtedly, the first two factors will continue to be of most importance and the "develop-and-test" method of making shock-resistant material will continue to be necessary; but the process can be shortened into a smaller number of cycles of "develop and test" if the designer is given good simple information that is obtainable from a more detailed study and a more careful and systematic analysis of the shock records. These methods should include not only the superficial method previously employed but, also, a study of the response of structures to the measured shock functions. This should be done primarily by electrical analogues of a relatively small number of types of mechanical systems and should aim at telling the designer, by only a few numerical values, what strength, rigidity, and clearances are required for any mechanical system under the most severe condition of shock for which protection is required.

EPILOGUE

In answer to the three questions originally raised, the following are proposed:

1. Adequate measurement of a shock motion requires a time function of either displacement velocity or acceleration. This gives only the translational motion of a point and will describe only that. Additional information concerning distortion and relative motion between parts may be essential in many cases.

2. Because of the wide amplitude and frequency ranges that are required for coverage for the measurement of shock motions, it is not practically possible for one type of instrument to suffice throughout. Consequently, accelerometers are used to bring out the high-frequency end of the spectrum, velocity meters are used to cover the middle (and by far the most important) region of this spectrum, and displacement meters are best over the low-frequency end. Special consideration is given to high-speed moving pictures, which can provide such a wealth of information concerning distortion and relative motion as well as making possible simple displacement recordings. Impact and reed type gauges are

used to provide, quickly and easily, information concerning special properties of the shock motion.

3. The information obtained from shock measurements is used to construct machines to simulate the shock and to provide designers with information for more intelligent design of shock-resistant equipment.

The greatest need at the present time is the extension of our knowledge of the nature of shock in the field and the reduction of measurements to such a form that they will be of use to the equipment designer.

SAVIBULL

During its last two conventions, the Society for Experimental Stress Analysis has emphasized mechanical shock and its related problems. At the Chicago meeting in May, 1947, as well as at the New York meeting in December, 1947, symposia were conducted on shock and impact. Most of the papers and discussions will be published in the Society's journal some time late in 1947 and in 1948. However, it seems that a few of the titles of the papers presented and some abstracts are of interest to the personnel associated with Savibull. From the May meeting:

1. Lateral Vibration and Stress in a Beam Under Shock Machine Loading - E. G. Fischer, Westinghouse Research Laboratories, East Pittsburgh, Pa. This paper solved the Bernouilli-Euler beam equation for the lateral vibration of a uniform bar on the assumptions that its ends are simply supported and subjected to a sudden velocity. It also described tests on the bar mounted to a shock machine of the medium size used by the Navy. Fair agreement was obtained between theoretical and experimental results, and it should be noted that this is a fruitful approach to a study of shock machine performance. Fischer pointed out that the accelerations determined from his theoretical solution were unbounded, but Professor Mindlin, of Columbia University, observed that the differential equation of motion of the beam was not complete as used and, in particular, took no account of internal damping. In the very interesting and important paper presented by Professor Mindlin (No. 6 below), allowance was made for internal damping and fair agreement was obtained between theoretical and experimental values of acceleration which are finite.

2. Determination of the Effect of Ground Impact Forces in Drop Testing of Airplanes - A. H. Pendersen and J. G. McCarthy, The Glenn L. Martin Company, Baltimore, Md.

3. The Experimental Determination of the Dynamic Structural Response of an Airplane to Impact Loadings - W. M. Claflin, Grumman Aircraft Engineering Corporation, Bethpage, N. Y.

These two papers on airplanes were devoted to impacts induced by landing forces. In particular, the paper by Mr. Claflin described the use of a structural model dynamically similar to an airplane. This work provides another example of experiments designed to evaluate response of structures to shock. The author stated that despite the relatively small size of the model, useful data were obtained for determining the performance of the prototype.

4. Effects of Impacts on Simple Elastic Structures in Terms of Equivalent Static Loads - J. M. Frankland and G. E. Hudson, David Taylor Model Basin, Navy Department, Washington, D. C.

5. Determining the Effects of Impacts on a Mechanical System by Tests Using Electro-Mechanical Equivalents - R. O. Fehr, General Electric Company, Schenectady, N. Y.

This paper described an important example of the use of the electrical

analogy for studying characteristics of mechanical shock. The impact acceleration was detected by an accelerometer whose output was recorded on a sound track of a motion picture film. The record on the sound track could be played back via a reproducing unit into a capacity-inductance-resistance circuit representing the analogue of the mechanical system under study. The author estimated that errors in the analysis are less than 15% for the overall system. He considers the results to be of practical significance.

6. Influence of Damping on the Response to Shock of Distributed Elastic Systems - R. D. Mindlin, Columbia University, New York, N. Y.

7. Some Characteristics of Navy "High Impact" Type Shock Machines - I. Vigness, Naval Research Laboratory, Washington, D.C.

8. High Speed Impact Testing Machines at the University of Illinois - N. M. Newmark, University of Illinois, Urbana, Ill.

A paper presented on moderately high velocity impact testing machines. The machines were shown by means of lantern slides. One was of the rotary type and the other a short guillotine type with a pneumatic cylinder and piston on top of it. The rotary type was somewhat like that used by Nadai at Westinghouse some years ago. Rotating discs carry near the periphery a movable tup which can be sprung out and caused to strike one end of a tensile specimen, the other end of which is fixed to ground. The guillotine type is used primarily for thrusting a steel capped piston transversely against a beam which has its ends supported on the ground. An adaptation to tensile testing is made by having the piston strike an auxilliary hammer which, in turn, strikes one end of a tensile specimen fixed to ground. The latter method is not satisfactory and illustrates vividly one of the difficulties overcome by use of the Guillotine Impact Testing Machine in the Material Laboratory, New York.

9. A Proposed New Type of Shock Instrument - W. P. Welch, Westinghouse Research Laboratories, East Pittsburgh, Pa.

This paper described preliminary design plans for a shock measuring and recording instrument. The instrument consists essentially of damped mass-spring system with stylus and recording drum. The mass is supported from flexure plates and and has the stylus affixed to one end and wire coil for electromagnetic damping affixed to the other end. It is proposed to use a permanent magnet as part of a magnetic circuit in order to provide the necessary damping. The effect of the damping was studied on a Westinghouse Transient Analyzer which schematically is a resistance-inductance-capacitance analogue of the mechanical system. A tentative conclusion is that 0.5 of critical is the most practicable value of the fractional damping factor. While the principle of the instrument is satisfactory, it is considered that much development is needed to produce a useful device for field tests.

The following papers were presented at the December meeting, in New York:

1. Studies of Transient Stresses in an Airplane Model Wing During Drop Tests - O. N. Leal, R. L. Bisplinghoff and T. H. H. Pian, Massachusetts Institute of Technology, Cambridge, Mass.

2. A Direct-Coupled Amplifier for Recording Dynamic Strain - W. R. Mehaffey, J. N. Van Scoyoc and D. S. Schover, Armour Research Foundation, Chicago, Illinois.

There are many applications of strain gauges in which the presence of high frequencies makes the use of the carrier systems impractical for field measure-

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ments. The need for measuring static as well as dynamic strains makes a direct-coupled amplifier a logical choice. This paper described the design and construction of a direct-coupled amplifier to meet the severe problems of stability and frequency response essential to strain gauge measurements over a wide range of frequencies.

The amplifier consists of four balanced push-pull stages and is stabilized by means of current and voltage feedback. A novel input circuit is used to provide push-pull input directly from the bridge. The amplifier has a flat frequency response with 500-foot cables from zero to 20 kilocycles. This response is obtained by a compensating feedback circuit which is adjusted to correct for the frequency response of the cable system.

The amplifier power supplies are of the regulated type and adjusted to hold a constant output voltage for large changes in line voltages. The filament and plate voltages are regulated as well as the d-c bridge excitation voltage.

Each amplifier channel is divided into a bridge section, amplifier section, and a power supply section. These sections are constructed on separate chassis units which are arranged to plug into a compartment of an amplifier cabinet.

3. Transients in Mechanical Systems - J. T. Muller, Bell Telephone Laboratories, Inc., Whippany, N. J.

This was a review of the shock work associated with shock mount design as performed at the Whippany location of the Bell Telephone Laboratories. The data and theories presented are the result of studies of shock mounts which have successfully withstood the action of shock machines. A definition of shock and the magnitude of the physical constants associated with it were discussed as well as the machines which produce shock or mechanical transients. Since we are dealing with transients, the mathematical analysis was performed by means of the Laplace transforms. Four impulses were shown, together with their transforms, and the result of these impulses on a solid body were discussed, particularly the motion which results from the application of a mechanical transient.

Next, a complete system, consisting of a base, a resilient medium or shock mount, and a solid mass and its mathematical analysis, was considered. Slides of a series of curves illustrated the interesting points of the above-mentioned theories. The transient analysis also gives solutions for the motions which occur after the pulse has disappeared. The results of this analysis can be used to great advantage in the proper design of shock mounts. With the aid of a model, some of those interesting features were demonstrated. Viscous damping and frictional damping were discussed and the results given in a family of curves. These theoretical considerations have led to a method of selecting the best natural frequency for a system so that it has maximum resistance against shock.

4. Theoretical and Experimental Investigation of Buckling Shock Mount - C. E. Crede and S. E. Young, The Barry Corporation, Cambridge, Mass.

This paper considered the buckling type of shock mount as a protective medium. A theoretical investigation and experimental verification of the acceleration transmitted by a buckling shock mount to the mounted equipment were included. The effect upon flexible components of the equipment was analyzed. Equations and curves were provided for selecting characteristics of the mounts required for optimum performance. An account of the design and performance of a shock machine, constructed to verify the theoretical investigation, was included.

The following are comments by Dr. Vigness, NRL, on linear versus "buckling" type shock mounts in relation to the papers by Mr. Muller and by Messrs. Crede and Young:

Mr. Muller has considered shock mounts with the usual linear force-deflection curve over a large part of the action; and Messrs. Crede and Young have considered a shock mount in which the force is nearly constant, regardless of the deflection -- except for very small or very large deflections. It can be easily shown that the latter, buckling type of mount will absorb more energy, for a given displacement and for a given force transmitted to the equipment it supports, than can any other type of mount. This also means that this type of mount will get the equipment it supports to move along with the shock motions with the least maximum force on this equipment for a given displacement available across the mount.

At first glance, this buckling type mount appears the better type to use for shock installations. However, it presents unknown factors and additional problems that require considerable study before it can be said safely that it is better than the linear mount, under general conditions. As with most controversial methods, it is probable that as our knowledge increases, we will find that some conditions dictate one type of method and other conditions require the other method.

As compared with the linear mount, the most obvious disadvantages of the buckling type mount are:

(a) When displacement occurs across the buckling mount, forces are suddenly applied to the supported equipment. This "shock excites" most of the supported parts into transients at their natural frequencies. Practically, as this force is relatively small and does take an appreciable time for its attainment, it is not likely that its effect will be serious under conditions of shock. It is an unknown factor, however, and requires further study. It is to be noted that these transients will be excited each time the mount passes through zero deflection.

(b) Under excitation of steady state vibration, the linear mount is resonant at a definite frequency, but the nonlinear mount has its resonant frequency a function of amplitude of vibration. If one assumes a design condition of identical maximum displacement (D) across the two types of mounts and the same energy absorbed in the mounts under the given maximum displacement, there results, as shown in Figure I, a restoring force (F) for the buckling mount and a restoring force (2F at D) for the linear mount. It can be shown that, for an amplitude of vibration (X_0), the period of the buckling type mount will be shorter than that of the linear mount until X_0 is slightly greater than 0.6 D. For greater amplitudes, the period of the buckling mount is greater than that of the linear until for $X_0 = D$, we have a ratio of $\frac{4}{\pi}$.

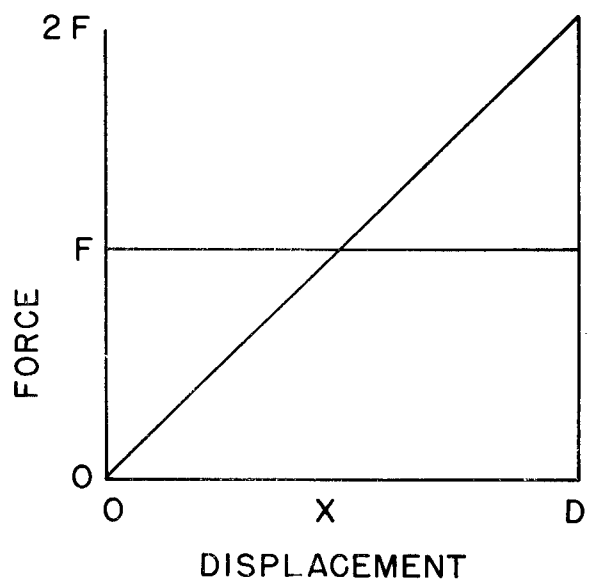

Figure I

It appears that, with normal damping, the effective stiffness of the buckling mount is greater than that of the linear

mount for frequencies and amplitudes probable on shipboard. However, we are still worried about the sudden change of force as the displacement goes through zero.

5. A pendulum Analyser for Mechanical Transients - J. N. Macduff, Polytechnic Institute of Brooklyn, Brooklyn, N. Y.

This article described the Pendulum Analyser constructed at the Polytechnic Institute of Brooklyn in conjunction with the Signal Corps Engineering Laboratories. Its basic purpose is to furnish design information for equipment subjected to transient shock and vibration. It accomplishes this end by means of a model analogy between a simple torsional pendulum and the full-scale degree of freedom system. The basic theory of the analyser was developed, and response curves for various simple pulses were presented. Dr. Vigness presents these comments on the paper by Mr. Macduff:

A first step in determining the probable damaging effect that a shock has on a structure is to study simple structures. The simplest structure of interest is that possessing but a single degree of freedom, such as a mass on a spring. If the base of the spring is subjected to a shock motion, it may be stated that the probability of damage is greater, the greater the spring deflection. If one is concerned about the spring, it is more likely to fail at large deflections. If one is worried about the force on the mass, it is also greater at larger deflections.

The deflection across the spring is commonly called the "response" of the mass to the shock motion. The magnitude of the response is often expressed in terms of non-dimensional units, such as "static deflection" which is the deflection of the spring caused by the weight of the mass it is supporting. This is, therefore, the deflection caused by one g, or one unit of gravity.

A first step in designing simple shock-resistant equipment -- say, a single-degree-of-freedom system -- would be to determine the forces acting on the equipment. These forces may be expressed in terms of the number of units of gravity acting on the system. The study of this response of single-degree-of-freedom systems to typical shock motions will give the designer the required information for this simple system.

A torsion pendulum analyser is a convenient single-degree-of-freedom system useful for the analysis of shock motions -- i.e., the determination of the response of the system to the shock motions. The top of the pendulum wire is twisted by an amount proportional to the acceleration of the shock motion by tracing an acceleration curve with a pointer which is at the end of an arm attached to the pendulum wire. This is done at a uniform rate along the time axis. The rotation of the mass supported by the wire will measure the response. By varying the moment of inertia of this mass about the wire axis, the frequency of the system can be changed. The speed of tracing can be conveniently slow, if the natural frequency of the pendulum is made low, for if one changes the speed of tracing and the natural frequency by the same amount, the response curves are similar. Therefore, one can use the response curve made by a slow transient on a low-frequency system to determine the response of the same transient on a faster time scale on a higher frequency system. By these methods, curves of response, or equivalent "g", versus the frequency of the system can be plotted for all types of shock under consideration. From these curves the designer determines the forces on his equipment and the deflection across his mounting system for any shock considered and for any natural frequency single-degree-of-freedom system.

Mechanical methods, such as the pendulum analyser, are slow and not practically capable of extension to systems of

greater complication than those of a single degree of freedom. Electrical cirucits usually can be devised that have differential equations which are the same in form as any given mechanical circuit. If proper electrical functions are impressed at proper points on an electrical analogue of a mechanical system, then voltages or currents, at some locations, have the same form, as mechanical motions or forces on the mechanical system. The mechanical response curve for a system can be determined quite readily, therefore, by electrical measurements only.

The electrical method has its disadvantages, of course; otherwise its use would be much more general. Chiefly, these are:

(a) Records of shock motions are usually obtained as curves, on paper or film, which are difficult to transfer back to the necessary electrical signal to actuate the electrical system.

(b) The transient must be fed to the electrical analyser at a fairly high speed if circuit components of inductances and capacities are to be kept at reasonable values.

(c) The damping in a mechanical circuit is often less than that of the electrical circuit unless inductances of very low resistance are used.

It is probable that the mechanical analyser will continue to be used for simple systems but that the electrical analyser will be more generally used. After the response, or number of units of static deflection, or equivalent "g", effective for a number of different mechanical systems of reasonable simplicity has been determined for typical shocks, much work needs to be done to find out to what degree these simple systems approximate practical systems, so that a designer with normal experience can select merely an effective "g", or some simple equivalent term, around which his equipment may be designed.

6. Vehicular Shock and Vibration Instrumentation and Measurements with Special Consideration of Military Vehicles - F. K. Priebe, Signal Corps Engineering Laboratories, Fort Monmouth, N. J.

7. Shock Testing Technology at the Naval Ordnance Laboratory - J. H. Armstrong, Naval Ordnance Laboratory, Washington, D. C.

8. The Photo-Grid Process for Measuring Strain Caused by Underwater Explosions - D. D. MacLaren, Columbia, University, New York, N. Y.

9. Displacement Measurement by Mechanical Interferometry - R. Weller and B. M. Shepard, Naval Ordnance Laboratory, Washington, D. C.

* * *

"The Editor, Shock and Vibration Bulletin,
Naval Research Laboratory,
Washington, D. C.

"During the war, a new type of shock-wave material was developed by Dr. Clark W. Chamberlain. This may be of interest to the Shock and Vibration Panel. It is described on following page.

"Very truly yours,
Stuart H. Chamberlain,
Associate Professor of
Engineering Research.

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"THE PRINCIPLE OF THE CHAMBERLAIN SHOCK-WAVE ABSORBER"

"A shock wave consists of a sudden pulse of acoustic energy, which rises rapidly to a maximum amplitude and falls off less rapidly thereafter. This pulse is made up of a fundamental frequency, and many strong harmonics or multiples of this frequency. The steeper, or more abrupt, the pulse is, the higher is the dominant frequency of the shock wave.

"The Chamberlain shock-wave absorber employs a series of mismatched acoustic impedances to diminish the amplitude of the shock wave passing through it. These mismatched impedances are provided by an assembly of copper or steel sheets which are subjected to heavy pressure to remove all the air between the metallic layers, except for an irreducible residual air film of a few molecules thickness. This assembly is placed in a tight fitting, welded steel cover and evacuated so that atmospheric pressure helps to keep the sheets pressed tightly together.

"When a shock wave enters the absorber, a portion of the acoustic energy is reflected back at each metal-air boundary, some energy is absorbed, and some energy is transmitted to the next metal-air boundary, etc., so that the amplitude of the transmitted shock wave tends to decrease exponentially as it progresses through the absorber. Higher frequencies are attenuated more than lower frequencies, tending to round off the transmitted pulse and to reduce the accelerations and, therefore, the forces on the body receiving the transmitted pulse. The energy reflected at each metal-air boundary undergoes multiple reflection between the various surfaces, and before this energy can emerge from the absorber as a series of delayed acoustic pulses, absorption at reflection degrades this energy into heat. This heat, tiny in amount, appearing in the air films and in the metallic sheets comprising the absorber, is presumably radiated away from the absorber as radiant heat."

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