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DESIGN AND DEVELOPMENT OF A HEAVY WEIGHT HIGH IMPACT SHOCK MACHINE

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FOREWORD

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This report covers the period of work from January 1952 to December 1954.

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ABSTRACT

The construction and operation of a pneumatically driven impact tester designed to evaluate the dynamic performance characteristics of cushioning materials is described. The 577-pound flat impacting element is capable of being projected downward with a velocity of from 20 ft/second to over 50 ft/second. An analog computer is used in a new application to record and analyze the impact.

PUBLICATION REVIEW

This report has been reviewed and is approved.

FOR THE COMMANDER

M. R. WHITMORE
Technical Director
Materials Laboratory
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INTRODUCTION

The aerial delivery of heavy cargos has required more basic information with respect to the energy absorbing characteristics of the various cushioning materials under high values of dynamic stress.

This work was initiated with the purpose of constructing a test device to determine empirically the performance of these materials under conditions approximating those of actual use.

The problem of properly designing a cushioned impact depends on several related factors. Those which concern the characteristics of the item to be protected, such as the mechanical strength of the components, their natural frequencies of vibration, specific and local conditions of the impact, etc., have been subordinated to the problem of evaluating the characteristics of the cushioning materials themselves.

Packaging engineers have often used the criterion of maximum acceleration in an impact as an indication of cushioning material performance. This has been used as a starting point in this work. Also, in order to conform to the end-use of the materials, these conditions have been formulated to define the area of investigation to which the impact tester shall be applied:

1. The static load of the impact hammer shall be in the order of 1 psi.

2. The entry velocities, i.e., the velocity of the flat impacting element at the time when it strikes the test sample, are to be controllable between 20 ft/second and 50 ft/second.

3. Provisions are to be made for testing samples of thicknesses up to 6 inches and diameters up to 24 inches.

4. Instrumentation shall be capable of providing a suitable means of measuring displacement, velocity, and acceleration of the impact hammer.

5. The conditions of temperature and humidity shall be controllable.

Although many types of shock machines have been built and used for different applications, there is no knowledge of any device operating under most of the specifications listed above. Consequently, a new and different design was considered necessary.

BASIC DESIGN CONSIDERATIONS

An air-conditioned testing laboratory was necessary to provide a reference level of temperature and humidity for testing of cushioning materials. The resulting space limitation precluded the practicability of using gravitational forces to accelerate the hammer; a minimum free-fall distance of 40 feet would have been necessary to achieve a terminal velocity of 50 ft/second.

Two possible methods of obtaining this velocity in a distance of a few feet were considered. One method used the energy stored in a compressed spring, while the other used the energy stored in a compressed air chamber. Either one could drive a hammer down against the material to be tested.

Comparison of the operational requirements for the two methods indicated that the compressed air system was more practical. In the available space (approximately 6 feet), a spring could be compressed through about 4 feet if guides were used to prevent buckling. However, the power to raise the hammer, especially through the last portion of the compression, would be relatively large since the force of the spring increases as it is compressed. Thus, a more powerful motor, as well as more time, would be required for raising the hammer than would be necessary in the pneumatic method wherein the air pressure need not be exerted on the piston until after hoisting. It also followed that with the spring arrangement the restraining force for the hammer in the cocked position would be relatively high and would require a larger triggering force for release. In addition, spring pressure for different velocities is not as easily controllable as air pressure.

The displacement, velocity, and acceleration of the impact hammer through the cushioning material are considered to be those fundamental attributes of its motion which are both necessary and sufficient for an evaluation of cushioning material properties. Velocity and acceleration can be derived from displacement by differentiation. The determination of displacement and velocity from acceleration by the process of integration is not as convenient in a graphical method of analysis; therefore, instrumentation was developed to measure displacement directly.

GENERAL DESCRIPTION OF TEST APPARATUS

The test facility that has been designed and built consists of a main structure supporting a pneumatically driven hammer (impacting element), an anvil (sample supporting element) with the plane of its upper face parallel with the plane of the lower face of the hammer, and a reinforced concrete pyramid foundation to which the anvil is attached.

The unit is adjustable to impart an impact velocity between 15 feet per second to 50 feet per second. The impacting element consists of a 577-pound hammer which is free-falling at the time of impact.

The mechanical and pneumatic systems are shown in simplified diagrammatic form in Figure 1, General Scheme of Test Apparatus.

A test sample is placed on the anvil with the hammer in its retracted position (upward), where it is held by two spring-loaded constraining elements. Operation of the firing switch opens Air Valve #1, which is normally closed, and admits high-pressure air into the cylinder head. This overcomes the constraining forces and the hammer-piston assembly is accelerated in the direction of the anvil. After a pre-set time interval, normally open Air Valve #2 closes. As the piston continues its downward travel, the compressed air in the cylinder is released through the exhaust system. The hammer then reaches the surface of the sample, at which time it has become free-falling because of the loss of compressed air. After the hammer has come to rest, it is hoisted back into its retracted position by means of an electric winch.

Just before the hammer strikes the surface of the sample, an electrical contactor attached to the hammer reaches the upper end of a linear resistance strip which has a regulated DC potential across its terminals. The voltage developed between the contactor and the upper end of the strip, as the hammer compresses the sample, is applied to either of two indicating and recording systems.

One system provides a film record of the displacement of the surface of the test sample versus time. Successive graphical differentiations of the recorded curve yield values of velocity and acceleration versus time, from which the energy-absorption characteristics of the test materials can be calculated.

The other system provides, by means of an analog computer and a camera, a simultaneous record of displacement, velocity, acceleration, and energy curves.

Auxiliary equipment consists of a source of compressed air and a control system which provides electrical control over all operations associated with the testing including operational interlocks to insure the safety of personnel and equipment. For additional protection, a diamond mesh steel cage has been placed around the impact tester with a sliding door providing access to the machine.

The air-conditioned laboratory housing the test facility is maintained at 70^{o}F and 65% R. H.

THE IMPACT TESTER

Hammer and Anvil Structure

To insure a steady platform for the test samples to be impacted, a 35-ton reinforced concrete foundation extending from bedrock to the laboratory floor has been installed. It has the form of a truncated pyramid with the bottom doweled to the bedrock while the upper surface is flush with the concrete floor of the laboratory, from which it is isolated by sand fill and a cork buffer strip. Attached to the surface of the pyramid is a 3-section steel anvil which has a total weight of approximately 3 tons.

The massive proportions of these two elements were considered necessary to insure a structure which would remain stable under the extremely high impact forces which would be encountered.

The hammer-anvil structure consists of a 12-foot high reinforced angle steel framework which supports the air cylinder, piston-hammer assembly, air valves, main air reservoir, and exhaust expansion chamber. All elements supported by the main frame are rigidly attached.

The Air Reservoirs

One reservoir is placed at the back of the frame and serves as a source of supply of a large volume of air which feeds the inlet valve (Valve #1) of the impact tester. The storage of a large volume of air is necessary in order to fill the cylinder of the impact tester in a short period of time. This main air reservoir is rated to withstand a pressure of 300 pounds per square inch. It is attached directly to the impact tester frame to eliminate the possibility of vibration from the impact tester loosening couplings between the impact tester and the reservoir, which would occur if the reservoir were hung as a separate unit.

The second air reservoir or exhaust expansion chamber attached to the frame serves as an exhaust tank for the air issuing from the piston when the vents in the piston are in line with the exhaust ports on the cylinder. This second reservoir is necessary to permit a large volume of air to escape from the piston in a short period of time.

Figure 2 shows the impact tester assembly enclosed in the steel protective cage. The large cylinder located to the right serves as the exhaust expansion chamber with a Fiberglas muffler at the top. For an indication of relative size, it may be noticed that on the platen of the impact tester frame is a sample 24 inches in diameter, 6 inches high, in position for testing. The diameter of the hammer at its surface of contact with the specimen is 27 inches.

The Hoist and Holding Mechanisms

The hammer is raised into its fully retracted position by means of an electric hoist, and it is held there by a constraining mechanism or latch. This latch consists of two spring-loaded members that engage with, and hold onto, two vertical members which are mounted on top of the hammer, one on each side. After these latches have operated in the "up" position, the hoist cables are disconnected from the hammer and moved out of the way preparatory to firing. Four pawls act as safety devices to hold up the hammer if the latch mechanism should fail. These pawls can be retracted after the cage door is closed.

Figure 3 shows the mounting arrangements in the upper structure. The main air reservoir may be seen to the left. The hoist motor with cables is connected to the hammer in its fully extended position. The safety pawls are shown as well as the tips of the latch members at the bottom of the picture.

Figure 4 shows the anvil with the hammer partially raised, and with a cushioning sample in place.

Air Valve System

An air valve, actually two valves, that has been designed for this specific purpose is located between the main air reservoir and the cylinder. Its opening and closing time is in the order of a few milliseconds. Valve #1 permits the high pressure air from the reservoir to enter the cylinder through a 3-inch pipe, overcoming the constraint and driving the hammer downward. After the piston has fallen a pre-set distance, Valve #2 is closed and the air in the cylinder is exhausted as the top of the piston falls below the upper edge of the exhaust ports. It then passes via the large capacity exhaust chamber to the outside through a 5-inch diameter exhaust line. The piston is moving essentially as a free-falling body with the removal of the compressed air above the piston by the exhaust system.

The impact velocity is adjustable to all values between 15 ft/second and over 50 ft/second by varying the air pressure in the main reservoir and the time interval during which air is admitted into the cylinder head from the reservoir, i.e., the time interval between opening Air Valve #1 and closing Air Valve #2. The time interval is adjusted with respect to the reservoir air pressure to affect the closure of Valve #2 as the top of the piston reaches the upper edge of the exhaust ports, in order to permit operation at the lowest possible air pressure at each velocity. The air pressure in the reservoir is controlled by adjustable diaphragm regulators, and the time interval is controlled electronically by a mono-stable multivibrator the time constant of which is adjustable.

Air Valve #1 returns to its normally closed position automatically while Air Valve #2 is reset into its normally open position by the manual operation of a reset switch.

Air Valves #1 and #2 are multiple-port slide valves, operated by air motors. By applying very high pressure air to the air motors and by accelerating their piston-shaft assemblies to a high velocity before engaging the valve slides, the valves are shifted from their normal positions into their operating positions in approximately 10 milliseconds.

In the course of the development of the pneumatic system, certain design features have been incorporated, and certain physical requirements have had to be met to achieve high impact velocities. Leakage of air taking place in the air valve assembly was a very important design consideration since it could prevent obtaining a pressure sufficiently high for the higher velocities. Manufacturing the valve sliding assembly to very close tolerances plus the use of Teflon gasketing has solved the problem of air leakage and has permitted build-up of air pressure close to 150 pounds per square inch. Prior to these measures, it was impossible to obtain and hold pressures over 90 pounds per square inch. Fins (or fairings) were also installed in the entrance and exit of the air valve assembly to reduce air turbulence which restricts the air stream flow.

Another factor affecting the impact velocities which can be obtained with the hammer is the impulse (pressure and time) which is given to the hammer. The impulse is controlled in part by the proper operation of the air motors. Reduced hammer impulse also results from friction in the slide type air valve. This friction, caused by the air pressure behind the valve, increases with air pressure and this requires more air motor pressure. More efficient operation of the valve also results if the air motors operating it are supplied by a separate air tank which maintains a higher and more stable supply. By virtue of such an independent supply, the air motor pressure is not affected by the pressure supplied to the hammer piston.

Accordingly, a 0.875 cubic foot air tank was installed in the air line to the air motors. Suitable controls were included so that the air motors can be operated at a pressure limited only by the maximum pressure rating of the air compressor.

Velocities of 52 ft/second have been achieved in some impacts.

CONTROL PANEL

The control panel consists of five parts:

1. An air control system
2. An indicator panel
3. A power supply chassis
4. An air valve delay control unit
5. A time marker generator

Figure 5 presents a view of the control panel on the right with the high speed camera on the left.

Air Control System

The air control panel serves to regulate the air pressure that will act upon the hammer piston. In order to obtain a large range of pressures, two parallel lines operated from a single compressor are used. One line has a control range from 10 to 90 psi and the other can control the pressure in the main air reservoir between the limits of 75 to 225 psi. Each line may be operated independently; both may be used simultaneously. Operating pressure is indicated on an air gage with a range from 0-300 psi in 1 psi increments. With the controls set for the proper range of air pressure, the air line from the compressor is opened by means of a solenoid valve. This valve is held open until the appropriate pressure has been reached in the main air reservoir.

The control valve and gage for the air motor tank is located above the panel. A diagram of the air control system is presented in Figure 6.

Indicator Panel

Figure 7 shows the indicator panel. This panel contains an extensive system of interlocks, consisting of relays and microswitches, which insure the safety of operating personnel by making it impossible to release the hammer until a series of precautionary measures have been taken. (For example, if the cage door to the impact tester is open, the safety pawls under the hammer cannot be retracted and the firing circuit is invalidated.) Interlocks also help to insure the validity of data by making it impossible to fire until appropriate settings of the rather complex controls have been made.

The relays, by means of red and green lights, indicate the positions of various items in the impact tester frame. Proper positions for the hammer hoist cables and pawls, Valves #1 and #2 (Valve #2 should be open and Valve #1 closed), power switch, cage door, and air pressure settings are all necessary before all the lights on the panel are green so that the instrument is ready to fire.

Firing is accomplished by a push-button switch which is located on the front of the control panel. This switch opens Air Valve #1, applying the air pressure on the hammer piston.

Power Supply Chassis

The power supply and voltage regulator are standard circuits which are necessary to supply DC power for the time marker generator, the air valve delay control unit, and the resistance strip. A full-wave rectifier with electronic regulator circuit supplies 250 volts to the two chassis above. A gas regulator tube utilizes this 250-volt source to supply the resistance strip with a voltage which can be varied from about 100 volts down to 10 volts, depending upon the range required. The required voltage on the resistance strip is determined by the material and thickness as well as by the recording equipment that is used, i.e., the Fairchild camera, or the Philbrick computer. Also provided in the power supply chassis is a switch to check the frequency division in the time marker chassis as well as to supply a calibration voltage in the scope when the Fairchild camera is used.

Air Valve Delay Control Unit

The velocity of the piston is a function of the air pressure action on the piston and the time interval during which both high-speed valves to the piston are open. As mentioned previously, air to the piston is controlled by means of two valves operated by air motors. The time delay between the opening of one valve and the closing of the second is obtained by means of the Air Valve Delay Control. The fire switch actuates the opening of Air Valve #1. It simultaneously kicks off a multivibrator circuit which controls a gas tube and a relay after a known delay determined by an RC circuit. When the gas tube fires, it causes Air Valve #2 to close, thus cutting off the air to the piston. Time delays of 0.15 seconds and 0.11 seconds are available by means of a selector switch on this panel. These two time intervals are sufficient to obtain operating velocities of the hammer of 20, 30, 40, and 50 feet per second. A wider range of delays is readily available by the introduction of additional capacitors in the RC circuit, and adjustment of a variable resistor in the same RC circuit.

Time Marker Generator

The time marker generator is an electronic device that produces negative pulses of short duration with definite known time intervals between pulses. A 100 KC crystal-controlled oscillator is used as an accurate source for the pulses. Two stages of frequency division, one being five and one being eight, are used to obtain a frequency of 2500 cps, or pulses which are spaced 0.4 milliseconds apart. These provide a time base for the photographically recorded curve when the Fairchild camera is used, by modulating or blanking the intensity of the electron beam of the oscilloscope to act as time markers.

TEMPERATURE CHAMBER

A dual-purpose temperature chamber was constructed for conditioning samples at high and low temperatures with accommodations for five samples each 6 inches thick and 24 inches in diameter. The walls and the horizontal door at the top are constructed of a double wall of 1/2-inch plywood sheathing containing a 2-inch thickness of Santocel thermal insulation. Both inside and outside surfaces are coated with aluminum paint. When used at the higher temperature of 160°F, the chamber is heated by means of two 1-kilowatt fin-type electric heaters mounted below the specimens with two air blowers providing the forced circulation. For the lower temperature of -67°F, the chamber is cooled by placing about 35 lbs of dry ice in wire caging around the upper inside of the chamber.

Temperature control, at the higher temperature, is effected manually by varying the output voltage to the heaters with an autotransformer. The lowest temperature attainable is between -60 and -70°F. Temperatures are measured via a recording thermometer actuated by a gas-filled capillary tube inserted into the chamber.

Due to the large size of the impact tester structure, the conditioning arrangement described was considered more feasible than attempting to condition both impact tester and the cushions while the latter were placed on the anvil in readiness for the impact at these extreme temperature conditions. Since approximately 30 seconds elapse between the time the samples are removed from the temperature chamber and impacted, a small unavoidable change in the temperature of the specimen does take place.

PRESENTATION AND ANALYSIS OF DATA

As has been previously stated, two systems have been set up whereby the data can be presented and analyzed. While the first method has not been used since installation of an analog computer system, it is described since it is available and may be used for special tests.

Fairchild Camera and Oscilloscope

The measured quantity in the impact tests is the displacement of the impact hammer as a function of time. A 10,000-ohm linear resistance strip, fixed parallel to the motion of the hammer, has an appropriate voltage connected to one end. Thus, the voltage at any point along this strip is a function of the distance from one end, which may be used as a reference. The strip is so positioned that a contact on the hammer will move along this strip during the period of interest - just before, and while the hammer is in contact with the cushioning material. This voltage, an analog of the motion of the hammer, horizontally displaces the spot on an oscillograph screen. The scope circuits have been modified so as to interchange the X and the Y axes. By interrupting the beam at a predetermined frequency, and photographing the position of the resultant dots in a moving film, a photographic record of displacement as a function of time is obtained. The oscilloscope trace is photographed by means of a Fairchild Oscillograph Camera with an f/1.5 lens. This camera, which is operated with an open shutter, is attached directly to the face of the oscilloscope.

The original design included a delay system which would start the camera prior to the release of the hammer in order to have the film reach a steady speed prior to the impacting of a sample. It was found, however, that the camera took a longer time to come up to full operating speed than the time required to move the hammer from the fully retracted position to its extended position. Rather than introduce another complicated system to control the camera automatically, it was decided simply to start the camera an instant before firing the impact tester. This simply involves a manual control on the camera unit located near the fire switch. The camera attains maximum velocity of 5 ft/second after approximately two feet of film have passed by the shutter (indicated by a footage meter on the camera). At this indication the impact tester is fired. After the firing cycle has been completed, the camera motor is manually shut off. It has been found from experience that a minimum of approximately fifteen feet of film is required for each record. Calibration of the resistance strip is obtained by applying a fixed portion of the voltage on the resistance strip to the input of the oscilloscope and photographing the resulting displacement as a part of the data for each run. In addition, a data card containing pertinent information for each run is photographed on the film at the beginning of operations.

Originally test runs were analyzed by measuring the displacement from the base line of dots in the oscilloscope film record by means of a traveling microscope modified to accommodate and hold lengths of 35 mm film. Subsequent numerical differentiation was used to obtain velocity and acceleration.

This method was found wanting in three respects. First, minor motions of the film trace in the winding mechanism on the microscope stage introduced errors which, although small in themselves, increased out of proportion in the process of differentiation to obtain velocity. Secondly, the operator proceeded mechanically with numerical calculations without benefit of visual observation of the displacement-time trace and its relation to the overall objective. Lastly, the method was tedious and did not lend itself readily to checking and examination.

Consequently, the technique of analysis as described in detail in Figures 8 through 12 was evolved.

By means of double graphical differentiation and other calculations made on an enlargement of the film, it was possible to obtain the desired velocity, acceleration, and energy relationships for the impacted cushioning material. This process was necessary for each sample of material that was evaluated. The system was generally successful and was used until the analog computer was installed. Figure 13 shows a sample analysis.

However, there were shortcomings. To obtain stress-strain curves, acceleration-time curves and energy relationships from the displacement-time trace was by necessity a long and time-consuming procedure. Also, the measurement of slopes, double graphical differentiation and measurement of areas with a planimeter was inherently difficult resulting in a compromise of precision.

Analog Computer

Installation of an analog computer has provided a means for obtaining data one minute after the actual test is completed. Figure 14 is an example of these data. This photocopy of an original picture of the oscilloscope trace in the computer system shows displacement, energy, acceleration, and velocity as functions of time.

The analog computer system utilized in this project is manufactured by George A. Philbrick Researches, Inc., Boston, Massachusetts, and uses a standard Dumont type 304-A Cathode-Ray Oscillograph for presentation, and a Dumont type 297 Oscillograph-record Camera, with standard Polaroid Land Camera back, for recording purposes. This arrangement is presented in Figure 15.

The computer uses the same resistance strip arrangement, with its regulated voltage supply, as in the former method of recording data to obtain a voltage analog of the position, or displacement, of the hammer at any instant. The computer itself is made up of a series of "building block" K-3 components, consisting of adders, integrators, coefficient boxes, and multipliers, which are connected together to make an electrical analog of the block diagram. The main computer rack contains as accessory units: a Model CS Central Signal component, a Model CR Central Response Component, and a regulated Power Supply, Model RS.

In the early method of recording displacement for this program, it was necessary to hold the frequency of the blanking pulses rigidly to 2500 cps so that the time interval between successive dots would be maintained at 0.4 milliseconds. In addition, calibration voltage readings had to be measured and recorded to correlate displacement distance on the photographic record and the physical distance through which the hammer actually moved. This procedure is considerably simplified by means of the computer system. The Central Response component, in conjunction with the Central Signal Component which generates a 20 cps signal, produces a calibrated display of voltage and time on any commercial oscilloscope. This "electronic graph paper", with sample test curves superimposed, (Figure 14), is similar in a broad sense to the familiar TV raster presentation, but rotated through 90 degrees. A 20 cps sawtooth is employed for the time-wise scan, so that a 2500 cps scan in the other direction permits 125 vertical lines. The vertical scan, in voltage form, is compared in passing with a set of constant calibrated voltages (-50, -40, -30 up to +30, +40, and +50 volts) and with the instantaneous values of all variables to be displayed. At each crossing a brightened spot is produced on the screen.

Starting at about 10 milliseconds after the horizontal flyback (return of the horizontal scan), the vertical lines are counted in fives and tens so that the decimal time ordinates may be emphasized by intensification for the 40 millisecond computing time. Thus, the calibration in two dimensions presents horizontal voltage references every 10 volts from -50 volts to +50 volts, and vertical lines every 0.4 millisecond with the 2 and 4 millisecond marks emphasized. This presentation is independent of any distortion in the amplifiers of the oscilloscope or in the geometry of the cathode-ray tube, since the signals and the calibration lines are all similarly transformed.

The Central Response display system allows up to four signals to be permanently displayed (inputs 1, 2, 3, and 4) or blinked by pushing an identifying button, as well as one of four other signals which may be selected and have been introduced to inputs A, B, C, or D. The voltage scan output of this panel is connected to the vertical input of an oscilloscope with the scope intensity input (Z input) from the Z connection of the same display panel.

In normal applications which are generally repetitive functions, a fixed amplitude sweep (ramp) from the CS Central Signal Component is used for the horizontal input (sweep) of the scope. The test of a cushioning material, however, is a "one-shot" affair, since it occurs only once. Provision is therefore made to trigger the single sweep as soon as the contact on the hammer enters the resistance strip.

The trigger circuit is shown in Figure 16, along with the complete computer block diagram. The signal from the resistance strip contact is fed to this trigger circuit as well as to the computing loops. A signal of 3 volts will trigger a single sweep across the scope face. Before firing of the hammer the circuit must be armed by pushing the reset button. The sweep time can be adjusted by changing the setting of the A-3 box to permit as much of the 40 millisecond sweep as necessary to cover the motion of the hammer in the material. The portion of the sweep on the oscilloscope screen can also be adjusted by the setting of the horizontal amplifier.

The Central Signal component, in addition to sweep outputs, also has an adjustable step, or square wave, output, and a clamp for the integrators to enforce an initial equilibrium. It also has an accurate source of DC voltage, manually adjustable. These voltages are available at jacks on the CS panel. The regulated power supply provides adequate regulated plus and minus 300 volts DC as well as 115 volts AC for all the computing components.

The equation for the motion of the hammer while in contact with the cushioning material has the form $M\ddot{X} + C\dot{X} + KX = 0$, where $\ddot{X}$, $\dot{X}$, and X are the second derivative with respect to time (acceleration), the first derivative with respect to time (velocity), and displacement, respectively. The electrical analog of displacement-time, e_X, is available so that the process of differentiation, performed twice, is necessary to get the electrical analogs for velocity and acceleration.

While differentiating components could be used, it has been found for reasons of stability that it is more desirable to use integrating components. Figure 17 is a simplified block diagram for obtaining the derivative of a function.

In the feedback loop the output of the integrating component J must be equal to the negative of the input e_X to have negligible input into the C or coefficient box. With the gain of the C box reasonably large, the output from the C box to the J box must be the derivative of e_X or $e_{\dot{X}}$ to complete the loop.

As shown in Figure 16, two loops similar to the above are used to obtain the electrical analogs of $\dot{X}$ and $\ddot{X}$. Coefficient boxes C-2 and C-4 are also included so as to adjust voltage outputs for proper presentation on the screen. A multiplying unit MU is used to obtain the analog of energy by obtaining the square of the velocity analog with the proper scale factors. Coefficient box C-5 acts as an isolation amplifier between the display unit and the computing loops.

The four voltage analogs for displacement, velocity, acceleration, and energy are displayed simultaneously on the scope. From the photographic record, similar to Figure 14, the physical values of these four items may be determined at any time during the impact, using the formulae and scale factors developed below.

Displacement: Since the length of the resistance strip is 10 inches, the displacement of the contacts from the zero voltage end of the resistance strip is given by the following formula:

$$X = \frac{10}{V} e_X \text{ (in.)}$$

where V is voltage applied to the complete resistance strip, and the value of e_X (voltage analog of displacement) is determined from the photograph of the test. Actual travel into the cushioning material is determined by correcting for location of the strip with respect to the top of the material, and the distance of the contact from the face of the hammer.

Velocity: The maximum possible velocity expected is $\dot{X}_{max}$ = 50 (ft/sec) = 600 (in./sec). The voltage analog of this is $\dot{e}_{X\ max} = \frac{V}{10} \times 600 = 60\ V$ (volts/sec). For a setting of 10 on the "J" box or unit time of 4 milliseconds: $e_{\dot{X}\ max}$ = $4 \times 10^{-3} \times 60\ V = 0.24\ V$ (volts).

Then $$\frac{e_{\dot{X}}}{0.24\ V} = \frac{(C\text{-}4) \times (C\text{-}2)_{\dot{X}}}{600}, \text{ and velocity at any point:}$$

$$\dot{X} = \frac{1}{(C\text{-}4) \times (C\text{-}2)} \times \frac{2.5 \times 10^3}{V} e_{\dot{X}} \text{ (in./sec), or}$$

$$\dot{X} = \frac{1}{(C\text{-}4)\ (C\text{-}2)} \times \frac{2.08 \times 10^2}{V} e_{\dot{X}} \text{ (ft/sec).}$$

The settings of Coefficient Units C-2 and C-4 are used to evaluate these formulas. As above V is the voltage applied to the strip and $e_{\dot{X}}$ (voltage analog of velocity) is determined from the photograph.

Acceleration: The maximum acceleration expected is $\ddot{X}_{max} = 1000(g's) = 1000 \times 32.2 \times 12 = 3.85 \times 10^5 (in./sec^2)$. The voltage analog $e_{\ddot{X}\,max} = 3.85 \times 10^5 \times \frac{V}{10} = 3.85 \times 10^4 V\ (volts/sec^2)$. For 4 milliseconds (setting of 10 in "J" box):

$$e_{\ddot{X}\,max} = (4 \times 10^{-3})^2 \times 3.85 \times 10^4\ V = 61.6 \times 10^{-2}\ V\ (volts).$$

Then $\frac{e_{\ddot{X}}}{61.6 \times 10^{-2} V} = \frac{\ddot{X}}{3.85 \times 10^5}$, so that acceleration at any point: $\ddot{X} = \frac{6.26}{V} \times 10^5\ e_{\ddot{X}}\ (in./sec^2)$ or

$$\ddot{X} = \frac{1.62}{V} \times 10^3\ e_{\ddot{X}}\ (g's).$$

As in the above equations V is resistance strip voltage and $e_{\ddot{X}}$ is voltage analog of acceleration from photograph.

Energy: The maximum energy for a hammer weighing 577 lbs at 600 (in./sec) is:

$$E_{max} = \frac{W}{g} \times \frac{\dot{X}^2_{max}}{2} = \frac{577}{32.2 \times 12} \times \frac{(600)^2}{2} = 2.69 \times 10^5 (in.-lbs).$$

Also $\dot{X}^2_{max} = 3.6 \times 10^5 (in.^2/sec^2)$.

Operation in Multiplying Component, Model MU, is such that $e_E = \frac{(e'\dot{X})^2}{25}$

so that $e_{E\,max} = \frac{(e'_{\dot{X}})^2}{25} = \frac{1}{25}\frac{(\ V\ )^2}{(2500)^2} (C-4)^2 \times 3.6 \times 10^5 = 2.3 \times 10^{-3} \times V^2 \times (C-4)^2$.

Then $\frac{e_E}{e_{E\,max}} = \frac{E}{E_{max}} = \frac{E}{2.69 \times 10^5}$, and

$$E = \frac{2.69 \times 10^5}{2.3 \times 10^{-3} \times V^2 \times (C-4)^2}\ e_E.$$

Simplifying:

$$E = \frac{1.17 \times 10^8}{V^2 \times (C-4)^2}\ e_E\ (in.-lbs)$$

or

$$E = \frac{9.75 \times 10^6}{V^2 \times (C-4)^2}\ e_E\ (ft-lbs).$$

V is voltage across resistance strip, C-4 is setting of the coefficient box, and e_E is the energy analog voltage from photograph.

In the process of setting up the analog computer into operation, considerable time was spent in making trial runs under various conditions and materials. Comparison tests were also made with the earlier method, using the Fairchild Motion Picture camera for recording test results.

APPLICATION OF TEST FACILITY

The heavy weight high impact shock machine described has been successfully used in the course of several hundred impacts taken in connection with a test program of cushioning materials.

The results of this test program are presented in WADC TR 55-229, entitled, "Performance Characteristics of Cushioning Materials Impacted Under a Heavy Weight High Impact Shock Machine".

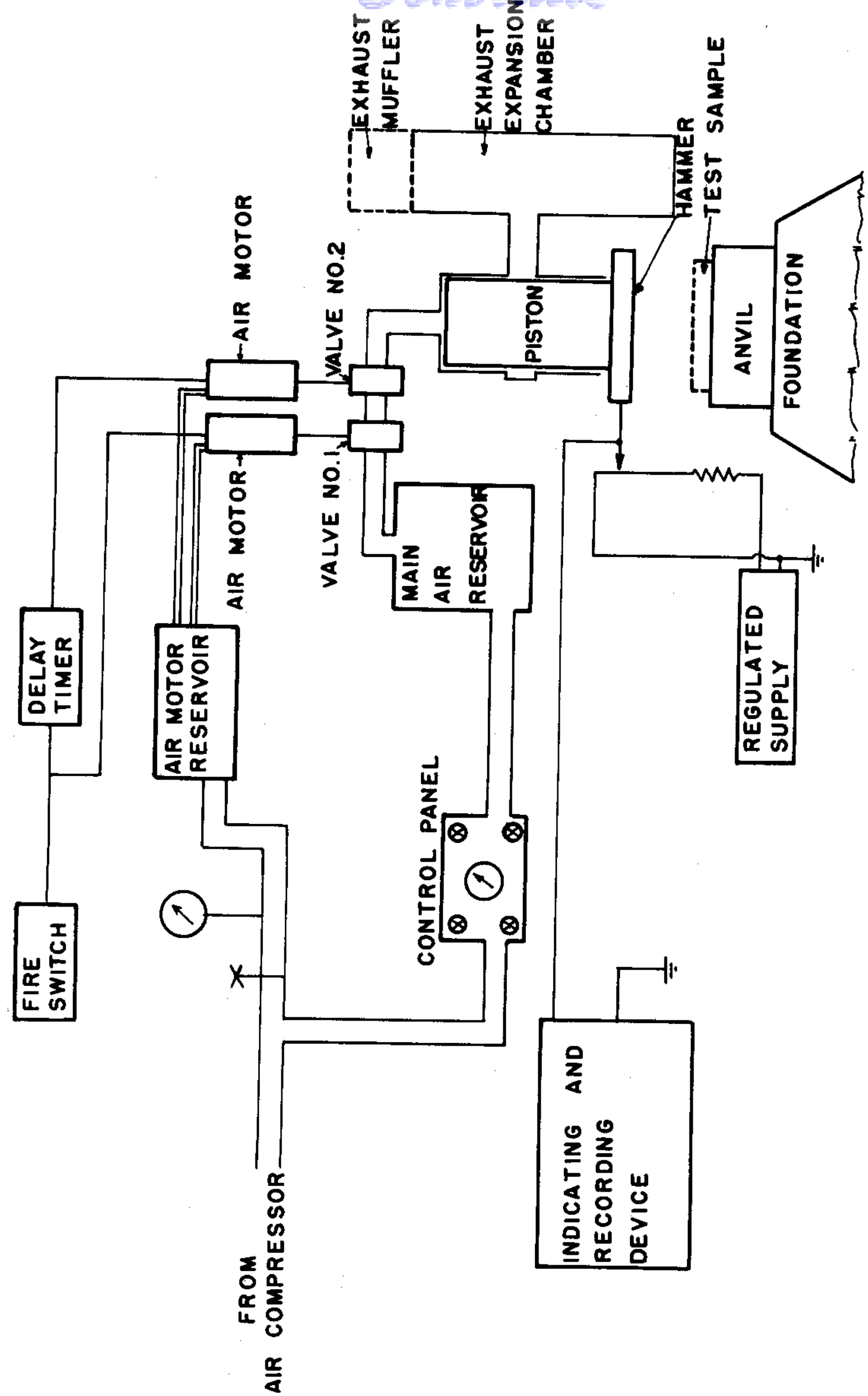


FIGURE I. GENERAL SCHEME OF TEST APPARATUS

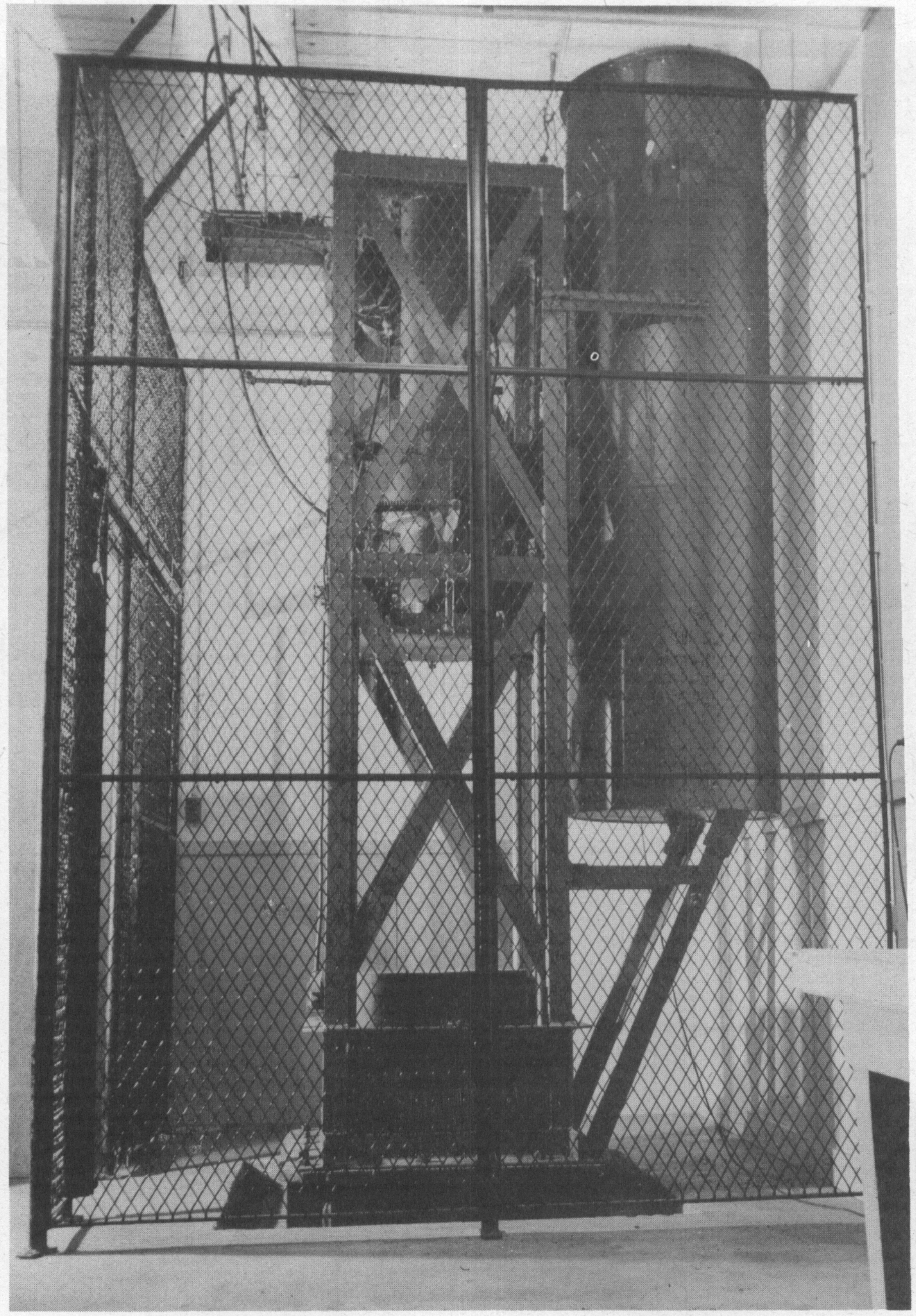


Figure 2

IMPACT TESTER ENCLOSED IN STEEL PROTECTIVE CAGE

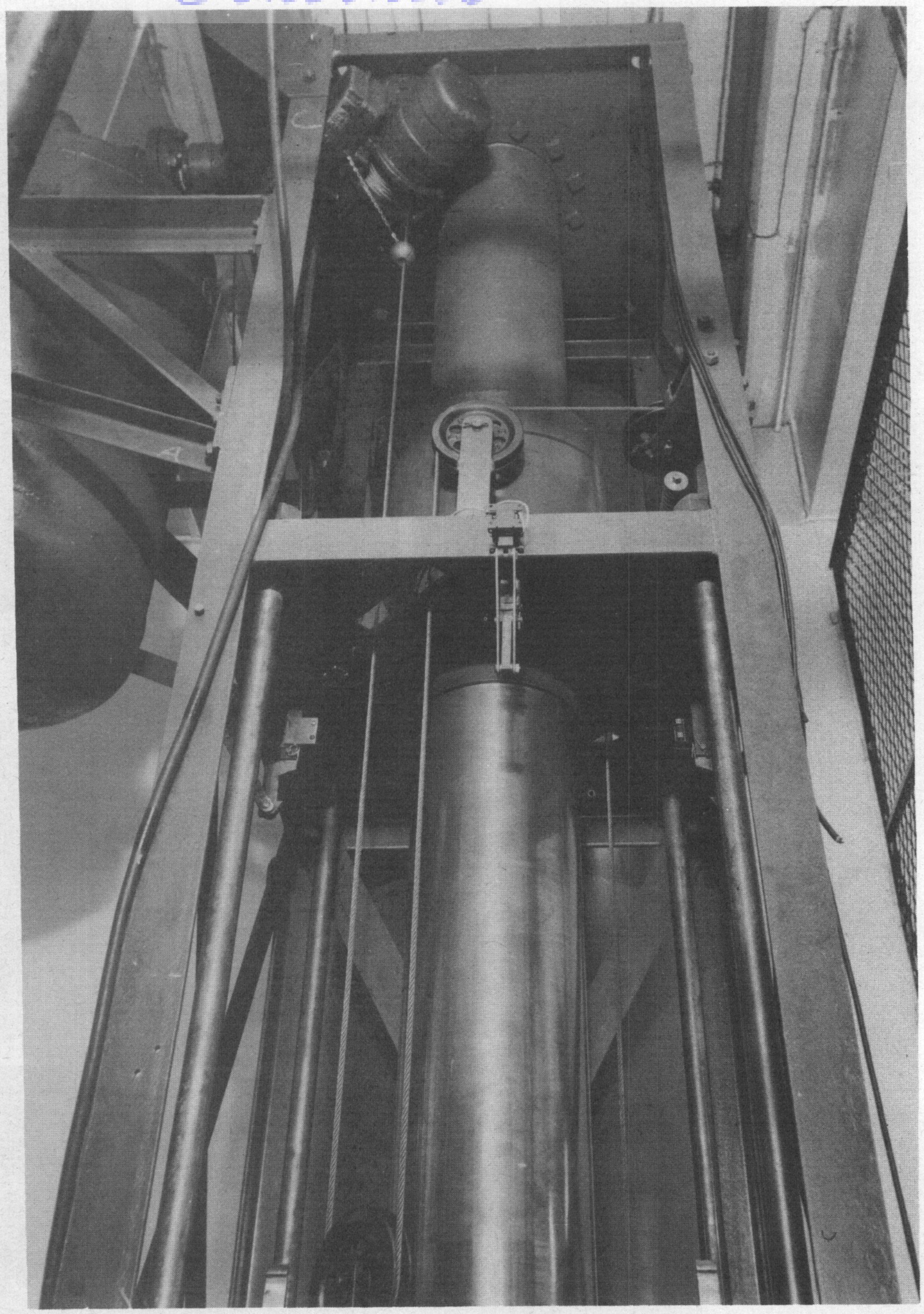


Figure 3

UPPER STRUCTURE OF IMPACT TESTER

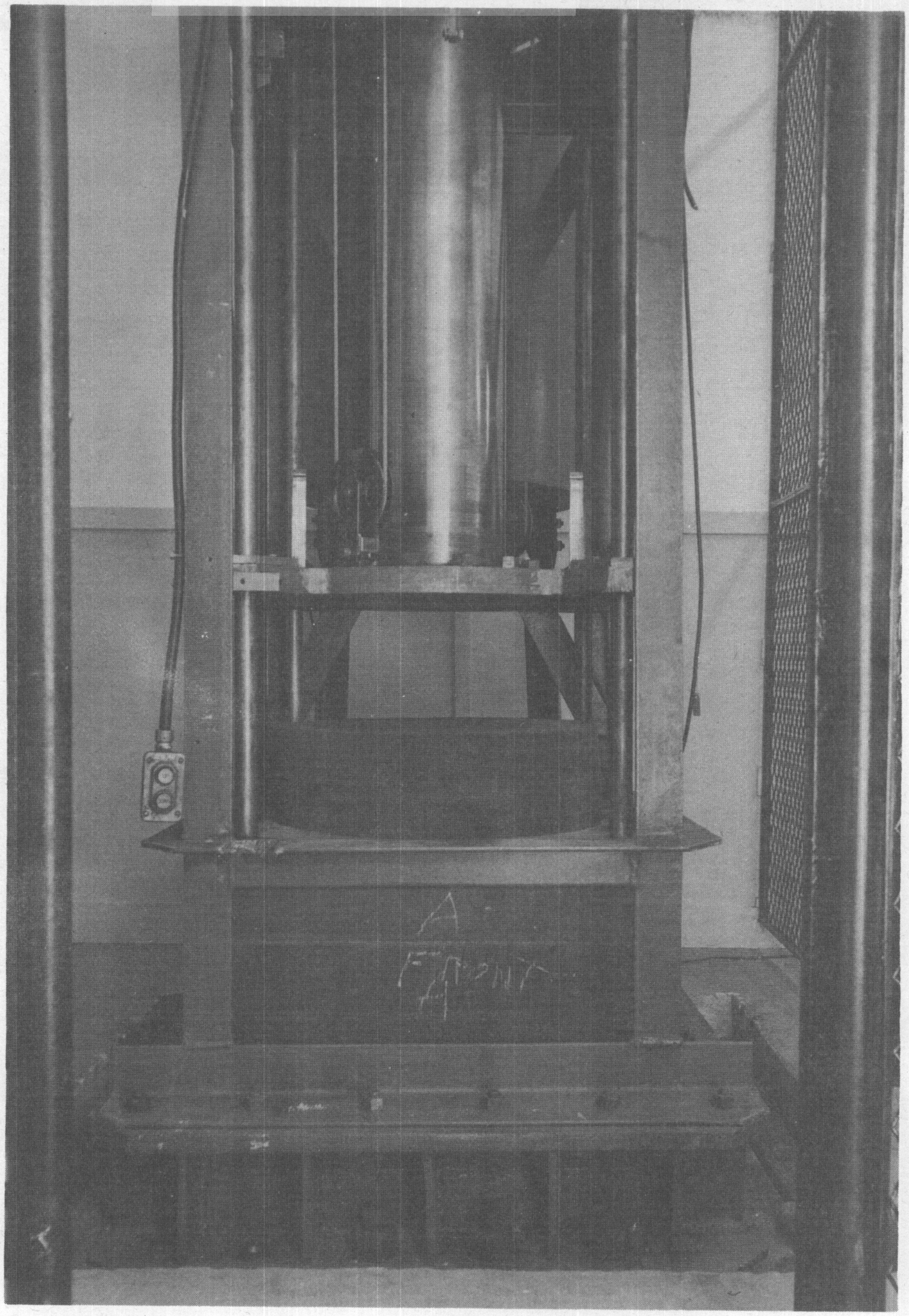


Figure 4

LOWER PORTION OF IMPACT TESTER

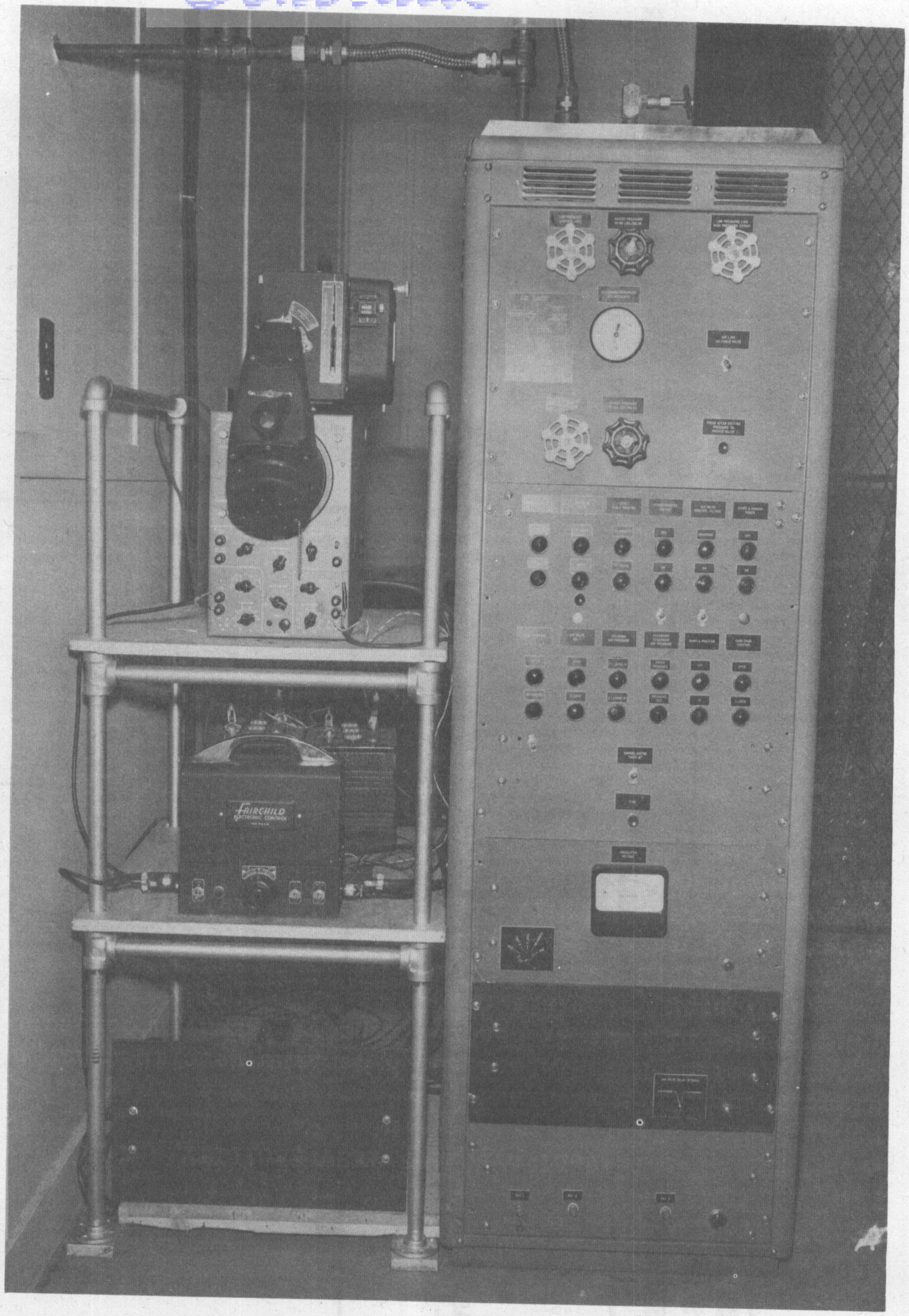


Figure 5

CONTROL PANEL AND HIGH SPEED CAMERA

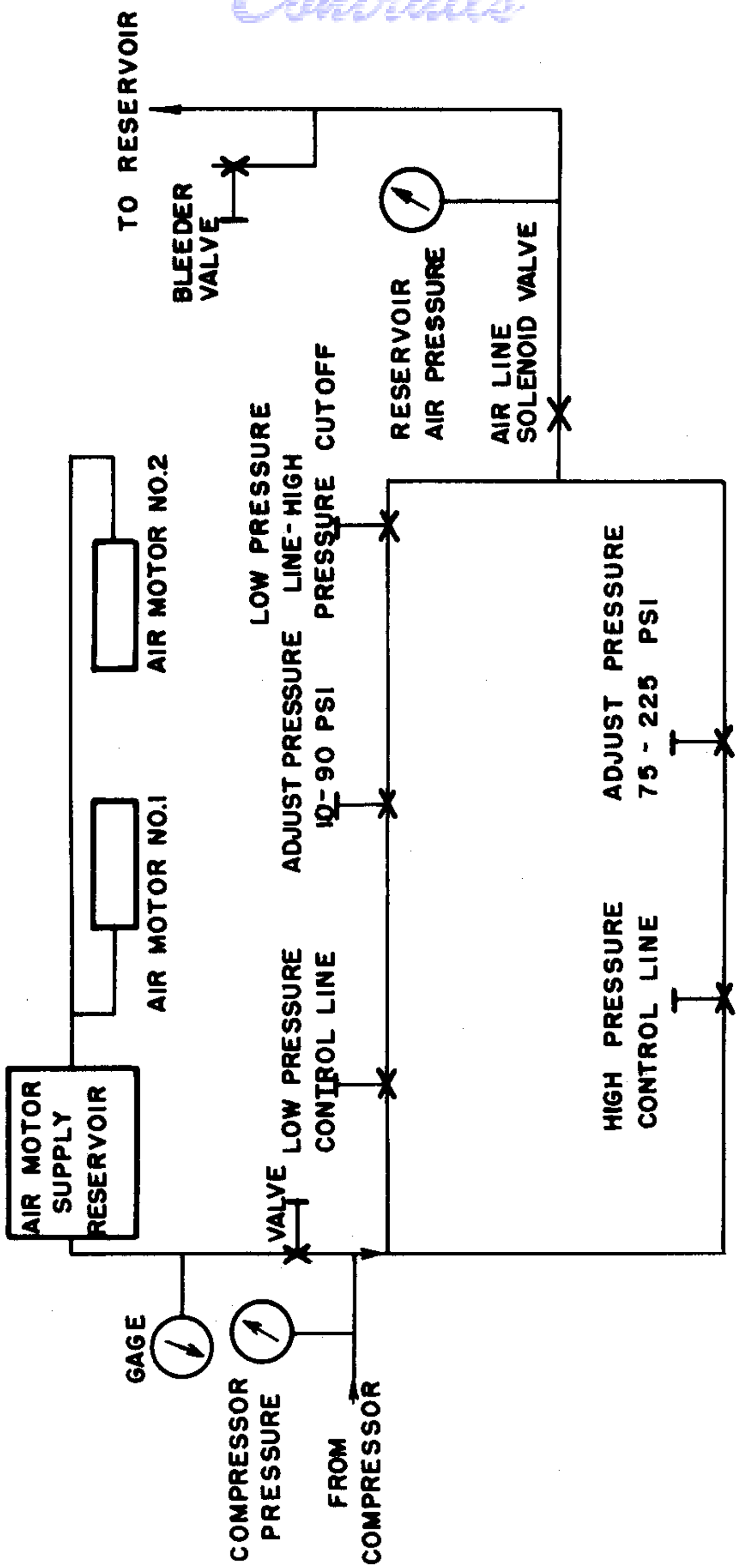


FIGURE 6. AIR CONTROL SYSTEM

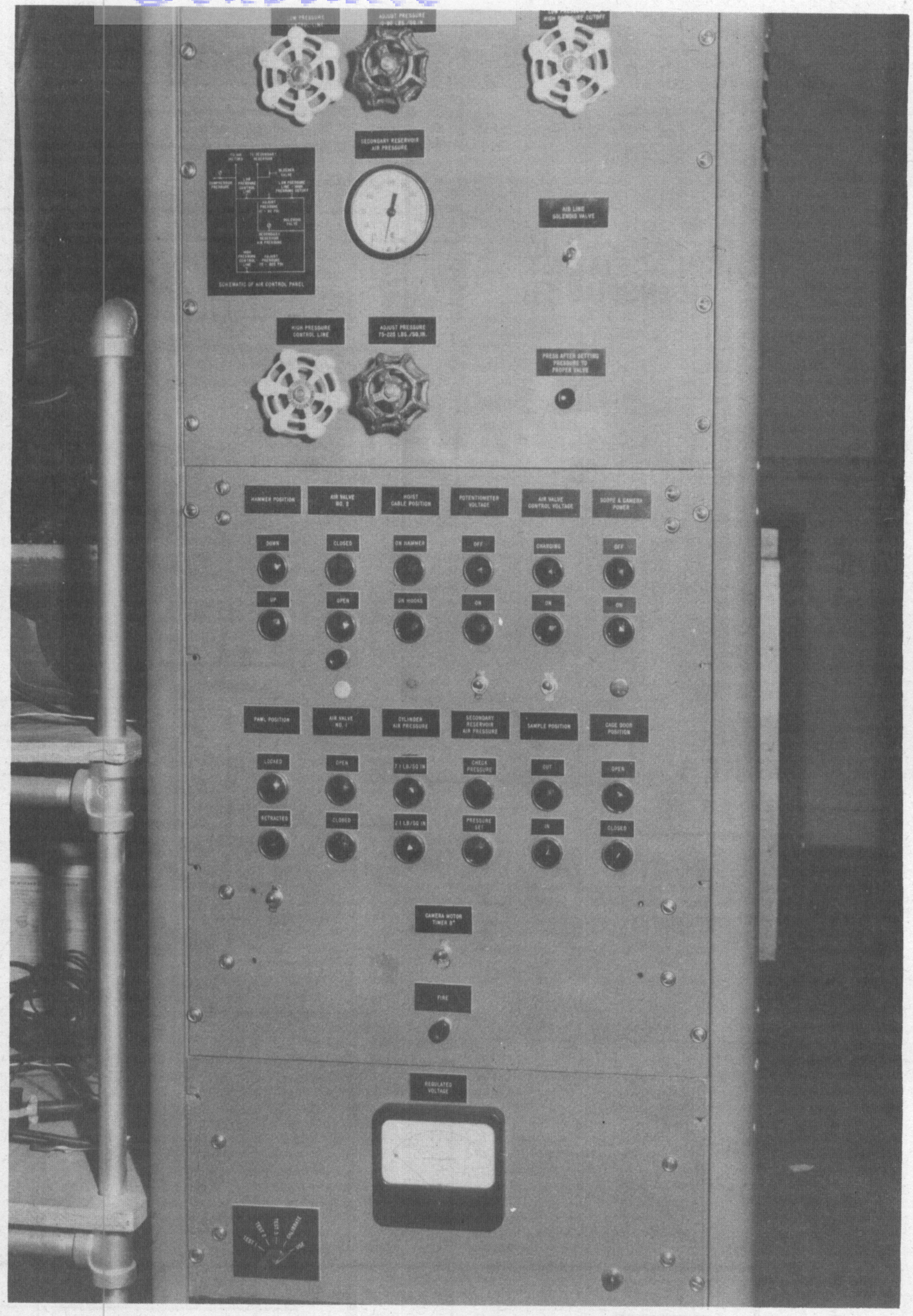


Figure 7

INDICATOR PANEL

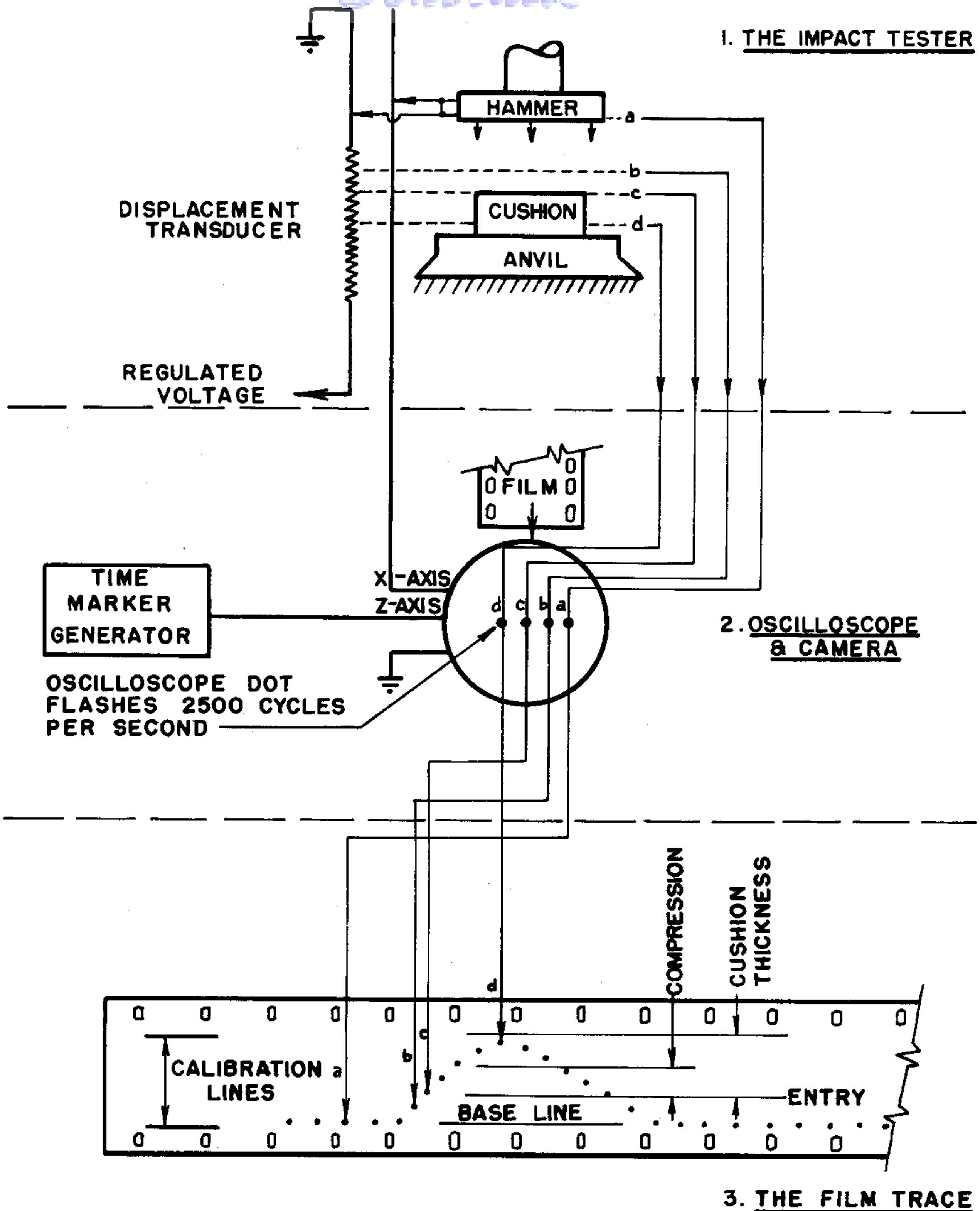
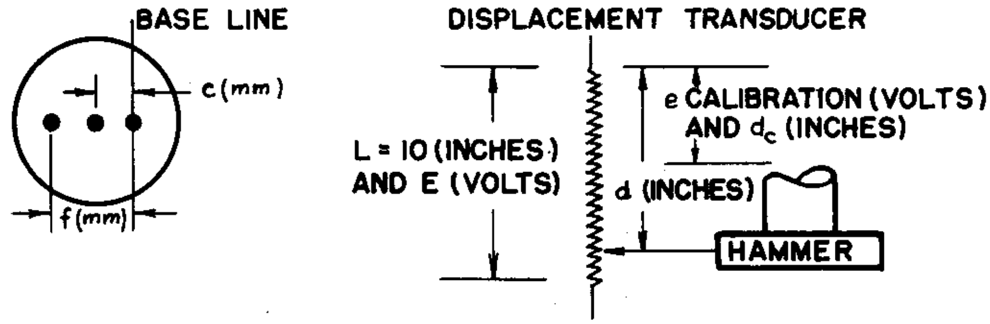


FIGURE 8. DIAGRAM OF GRAPHICAL METHOD OF ANALYSIS (THE DATA)

4. SCOPE CALIBRATION



Calibration distance c on scope corresponds to d_c inches on resistor and distance f on scope corresponds to d inches on resistor.

Then $\frac{d_c}{L} = \frac{e}{E}$

and $\frac{d}{d_c} = \frac{f}{c}$. Solving both equations for d,

$$d = \frac{e}{E} \cdot \frac{f}{c} \cdot L \text{ (inches)} \qquad \text{Equation (1)}$$

Also, where t is the time in intervals (1 interval = 0.0004 secs),

$$\text{Average velocity } \bar{v} = \frac{d_2 - d_1}{t_2 - t_1} = \frac{\Delta d}{\Delta t} \frac{\text{(inches)}}{\text{(interval)}}$$

However, $d = \frac{e}{E} \cdot \frac{L}{c} \cdot \Delta f$

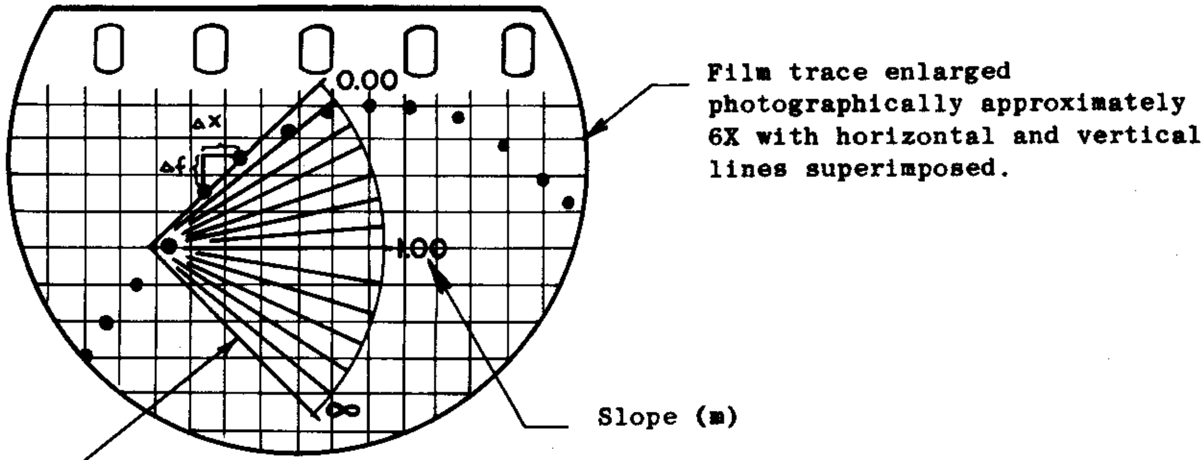
So, $\bar{v} = \frac{e}{E} \cdot \frac{L}{c} \cdot \frac{\Delta f}{\Delta t} \frac{\text{(inches)}}{\text{(interval)}}$

$$= \frac{e}{E} \cdot \frac{10}{c} \cdot \frac{\Delta f}{\Delta t} \cdot \frac{1}{.0004 \times 12} \frac{\text{(ft)}}{\text{(sec)}}$$

Or, $\bar{v} = \frac{e}{E.c} \cdot \frac{\Delta f}{\Delta t} (2,083) \frac{\text{(ft)}}{\text{(sec)}}$ Equation (2)

FIGURE 9. DIAGRAM OF GRAPHICAL METHOD OF ANALYSIS (SCOPE CALIBRATION)

5. MEASUREMENT OF VELOCITY



Protractor is applied successively to get slopes of dot pairs on film trace. Since film speed is constant, Δx is constant. Then, $\Delta f = m \cdot \Delta x$ and substitution is made in equation (2), Figure 9, to get velocity.

6. VELOCITY-TIME CURVE

$$G, \text{(Average Acceleration)} = \frac{\Delta v}{\Delta t} \left(\frac{ft}{sec} \cdot \frac{1}{interval}\right)$$

$$= \frac{\Delta v}{1} \left(\frac{ft}{sec} \cdot \frac{1}{.0004 \text{ secs}} \cdot \frac{1 \text{ g}}{32.2 \text{ ft/sec}^2}\right)$$

Therefore,

$$G = 77.8 \ \Delta v \ \text{(g's)} \qquad \text{Equation (3)}$$

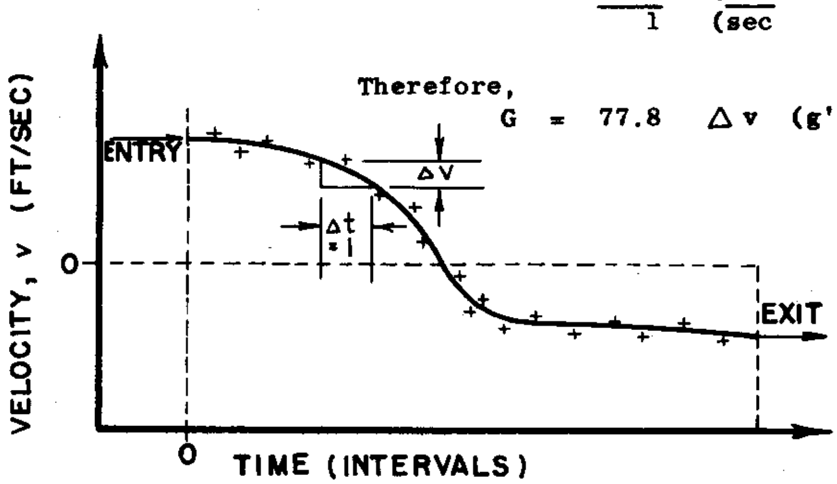
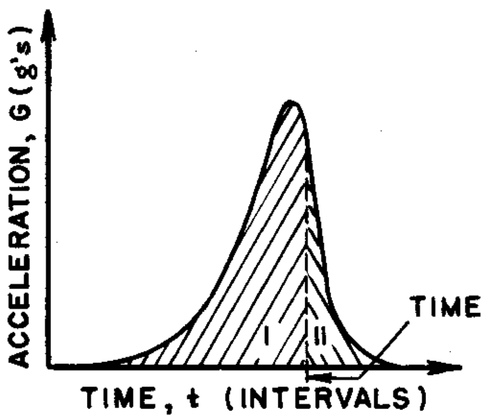


FIGURE 10. DIAGRAM OF GRAPHICAL METHOD OF ANALYSIS (VELOCITY)

7. ACCELERATION-TIME CURVE



Successive values of acceleration are obtained from the smoothened velocity-time curve.

8. IMPULSE-MOMENTUM CHECK

From the impulse-momentum theorem, where

F = the force exerted by the cushion on the hammer

m = mass of the hammer

and W = weight of the hammer (577 lbs),

$$\text{we have} \int F \,.\, dt = \Delta(m \,.\, v).$$

Then,

$$\int W \,.\, G \; dt = m\Delta v = W\int G \; dt = W \,.\, (\text{Area under G-t curve})$$

So, Area I (see above) $= m \,.\, v_{entry}$

and Area II $= m \,.\, v_{exit}$.

These relations are used as a check upon the process of graphical differentiation used in the analysis.

FIGURE II. DIAGRAM OF GRAPHICAL METHOD OF ANALYSIS (ACCELERATION AND IMPULSE-MOMENTUM)

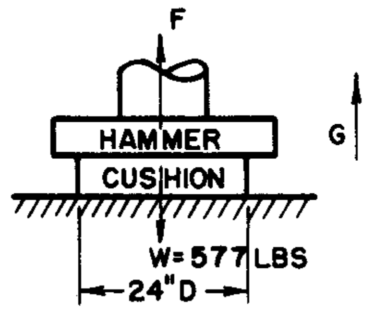
9. STRESS-STRAIN CURVE

From Newton's 2nd Law,

$$F - W = m \cdot a = W \cdot G$$

$$\text{So, } F = WG + W = W(G + 1)$$

But,

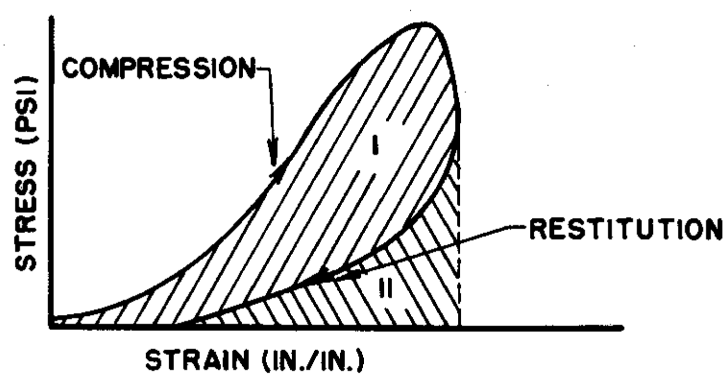


$$\text{Stress} = \frac{\text{Force}}{\text{Area}} = \frac{577\,(G + 1)}{\frac{\pi}{4} \cdot 24^2} = 1{,}28\,(G + 1) \text{ (psi)}$$

And,

$$\text{Strain} = \frac{\text{Compression}}{\text{Orig. Thickness}}$$

This ratio is obtained from the film trace (See Figure 8).



$$\text{Energy Transferred} = \int_{0}^{\text{Max. Strain}} \text{Stress} \cdot d\,(\text{Strain}) = \text{Area}_{I + II} \quad \frac{(\text{in. - lbs})}{(\text{in.}^3)}$$

$$\text{Energy Returned} = \int_{\text{Max. Strain}}^{0} \text{Stress} \cdot d\,(\text{Strain}) = \text{Area}_{II} \quad \frac{(\text{in.-lbs})}{(\text{in.}^3)}$$

FIGURE 12. DIAGRAM OF GRAPHICAL METHOD OF ANALYSIS (STRESS-STRAIN)

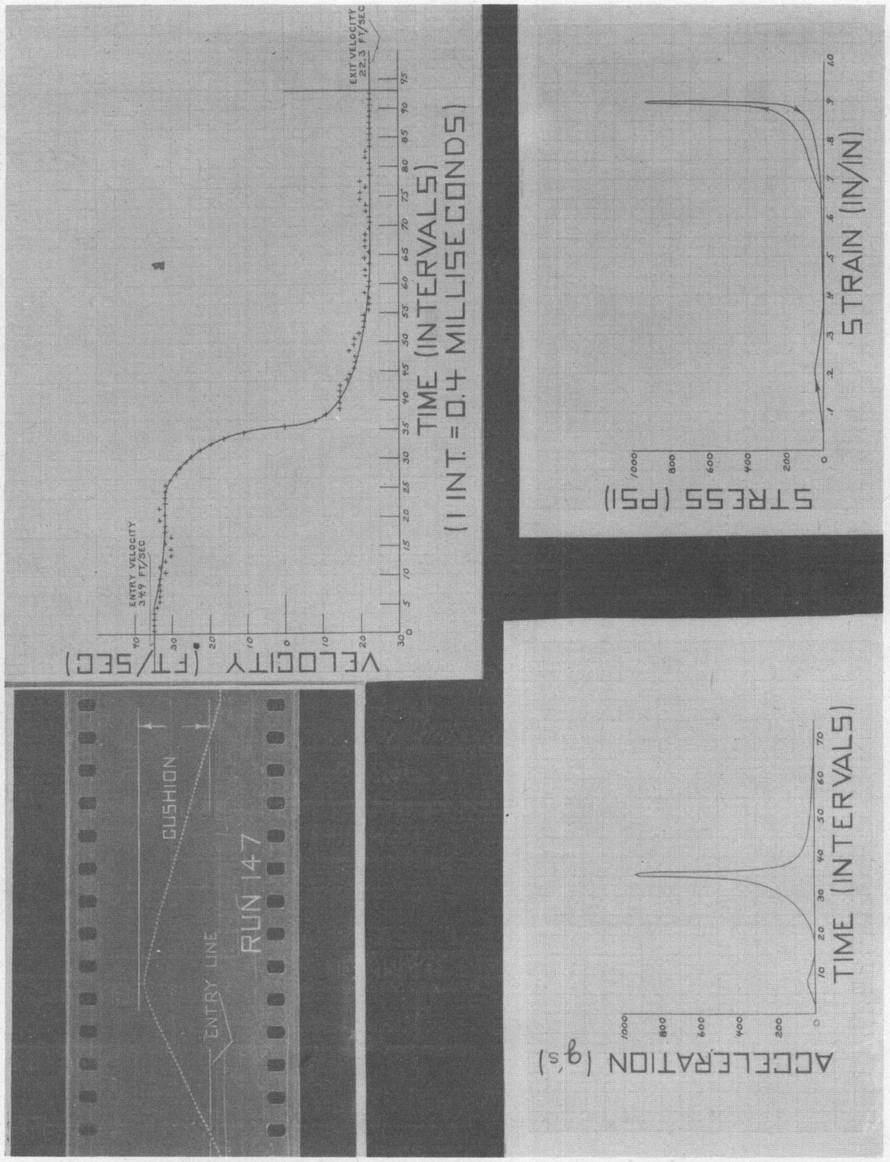


Figure 13

SAMPLE OF GRAPHICAL ANALYSIS

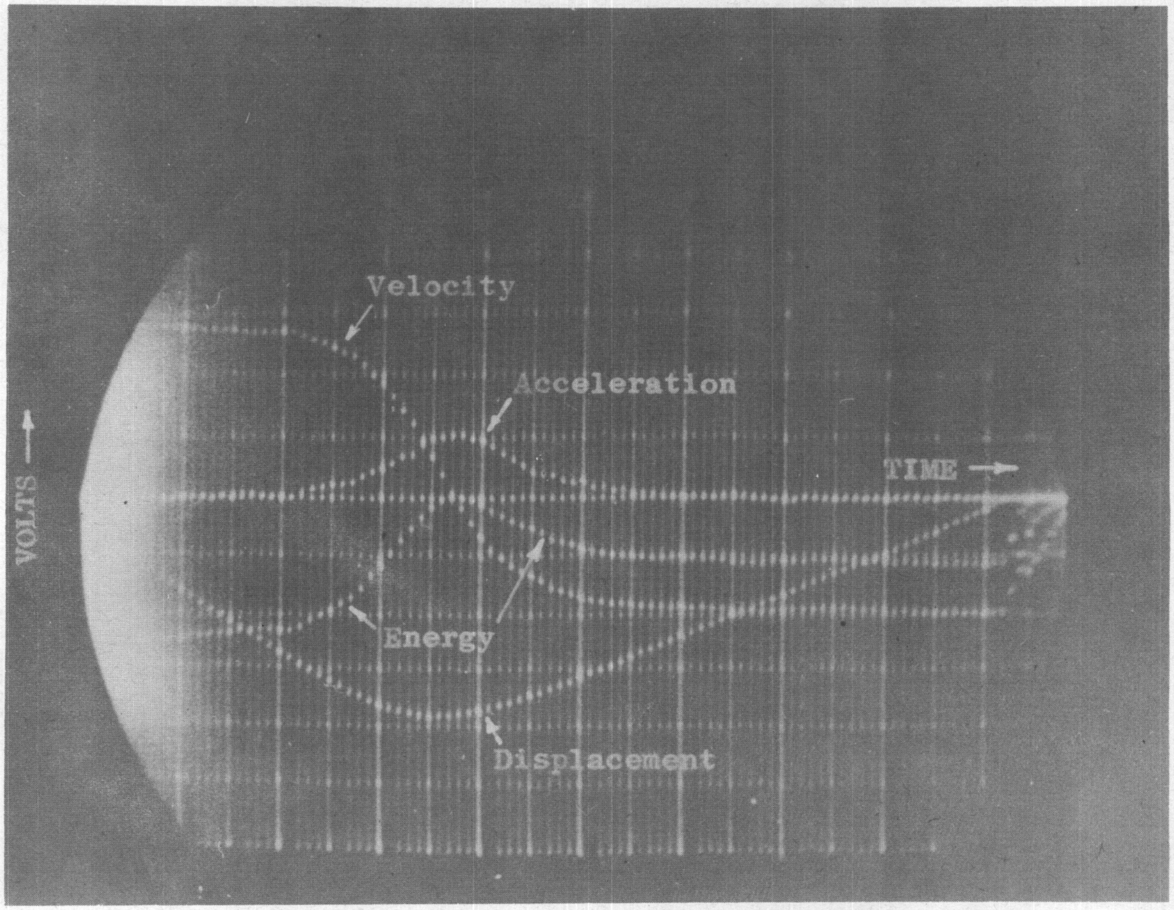


Figure 14

OSCILLOSCOPE RECORD OF TEST RUN WITH ANALOG COMPUTER

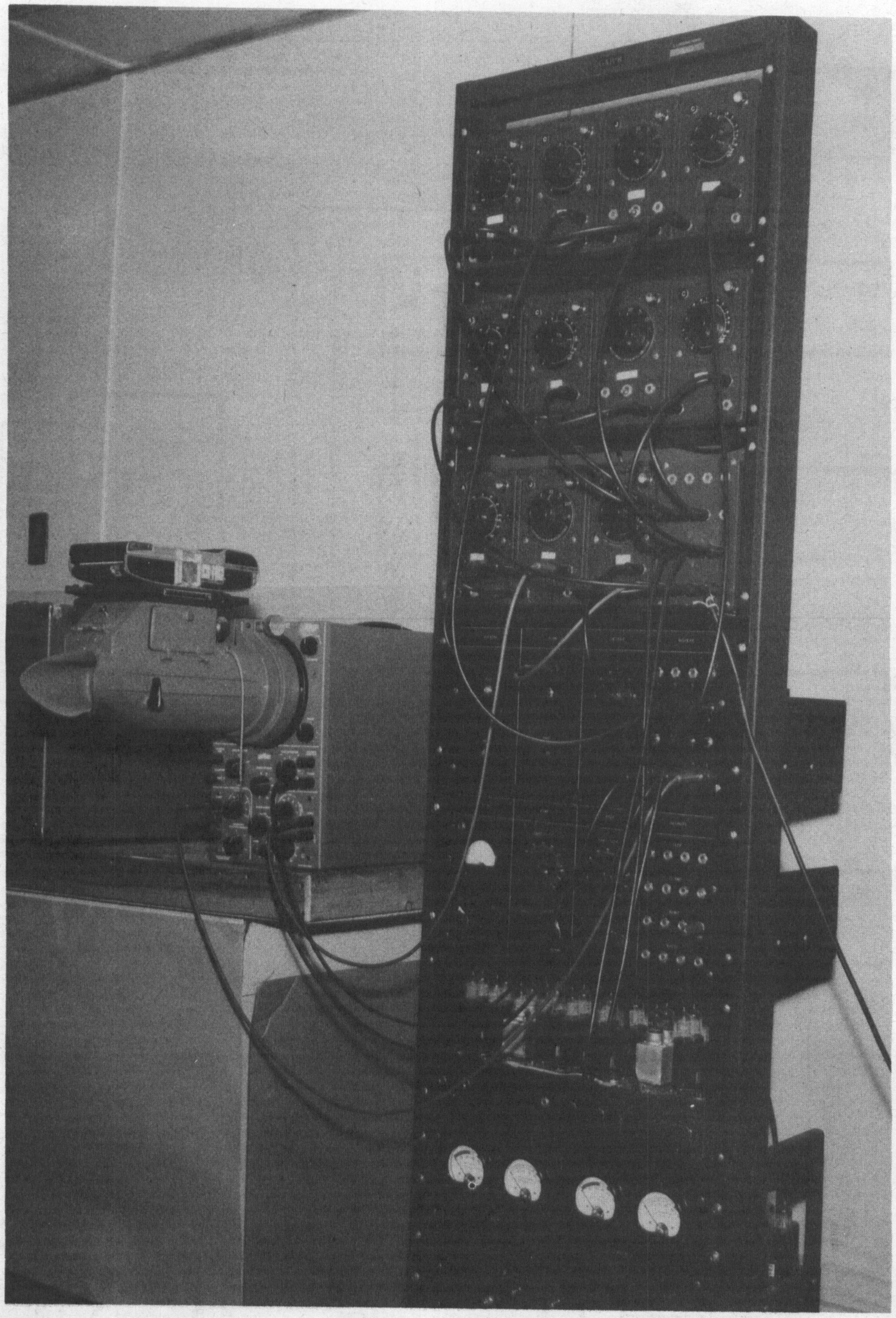


Figure 15

ANALOG COMPUTER

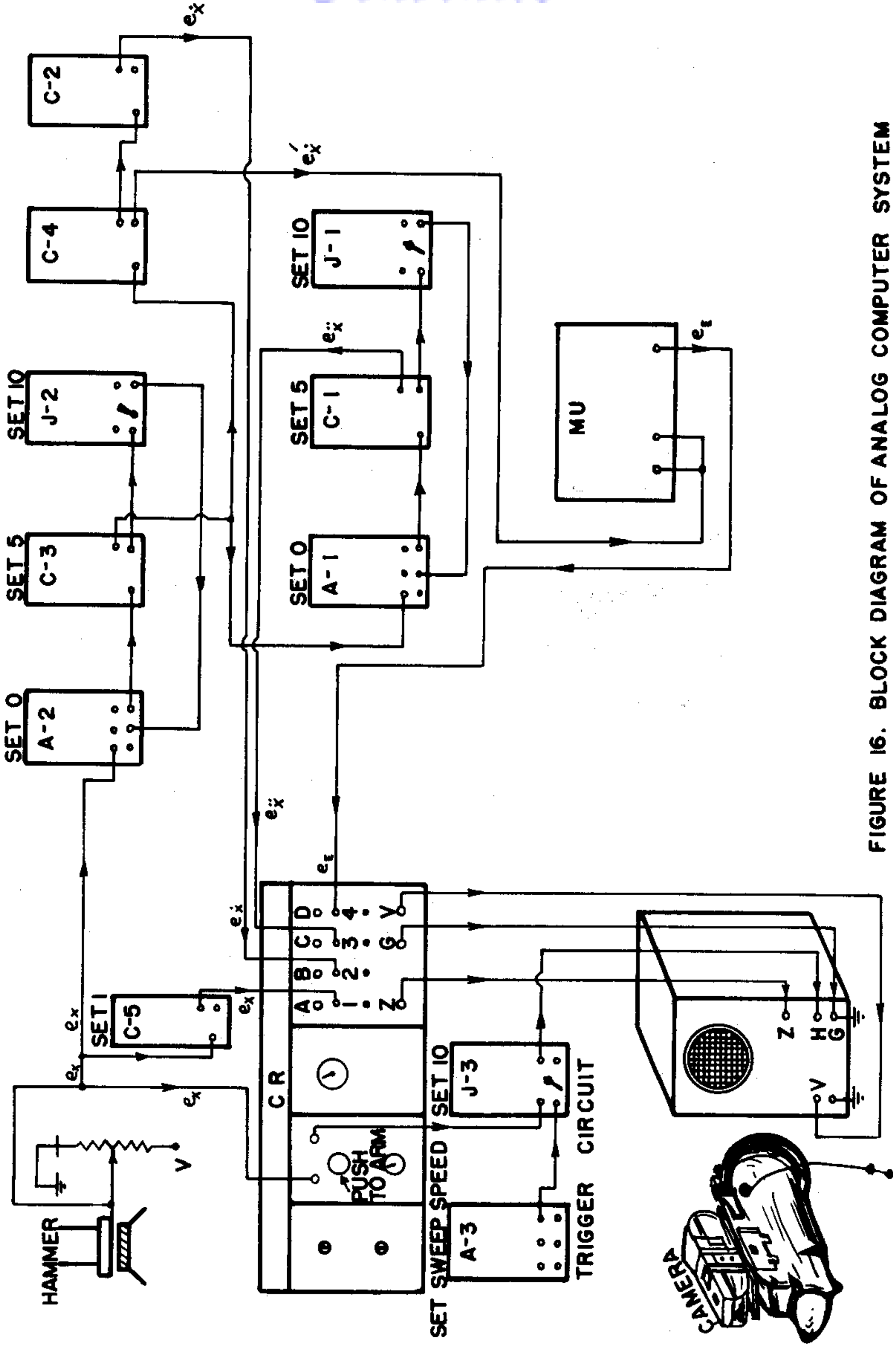


FIGURE 16. BLOCK DIAGRAM OF ANALOG COMPUTER SYSTEM

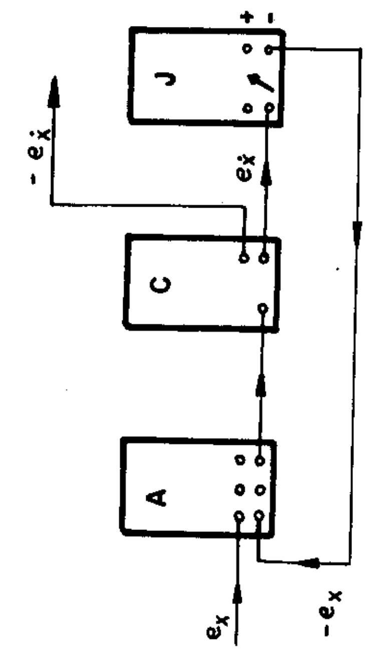


FIGURE 17. COMPUTING LOOP TO OBTAIN DERIVATIVE OF A FUNCTION