60 KIP CAPACITY SLOW OR RAPID LOADING APPARATUS

by

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AIR FORCE SPECIAL WEAPONS CENTER
Air Research and Development Command
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I INTRODUCTION

1. Purpose and Scope

In order to obtain dynamic tests of structures or structural components in the laboratory it was necessary to design and construct a machine which could provide the required load and stroke. Two of the machines (No. 1 and No. 2) described in this report were constructed for the Air Force under Part b of Exhibit "C" of supplemental agreement S 9 (55-102) Contract AF 33(616)-170. The general specifications for the machines are as follows:

a. Load capacity - 50,000 lb with approximately 75% of original full load acting at the end of stroke if used without controlled unloading.

b. Length of loading stroke - 12 to 18 inches, the longest possible stroke being the most desirable.

c. Time for load to reach peak value - less than 0.010 seconds with the least possible time being the most desirable.

d. Unloading time - maximum 0.030 seconds without control. The loading unit shall be designed to incorporate controlled unloading.

The machines were built with the intention of using them in carrying out the testing program set forth in the above contract. A similar machine (No. 3) was simultaneously constructed for the University of Illinois. This machine was later altered, giving it entirely different loading characteristics.

The operational characteristics determined from calibration tests and from a theoretical analysis are presented for both types of machine. Since the machines are in no way similar to the usual laboratory equipment, it is intended that this report be used as an operation manual for the machines; thus, working drawings of the component parts, and other miscellaneous information are included.
The design of the machines is based upon the University of Illinois' 20 kip Pulse Loading Machine designed by J. M. Massard\(^{(2)}\)*. The basic machines described in this report were designed and constructed under the supervision of F. L. Howland. The trigger mechanism, slide valves and minor parts of the machine were altered by the author and Professor Massard in order to insure reasonably consistent operational characteristics.

2. Acknowledgments

The machines described in this report were conceived and designed by staff members at the University of Illinois in cooperation with Wright Air Development Center and Air Force Special Weapons Center, Department of the Air Force, under Contract AF 33(616)-170. The project was conducted in the Structural Research Laboratory of the Department of Civil Engineering under the general direction of N. M. Newmark, Research Professor of Structural Engineering and Head of the Department. The project was under the direct supervision of F. L. Howland, Research Associate in Civil Engineering, and is now supervised by J. M. Massard, Research Assistant Professor of Civil Engineering.

Research Assistants and Associates who have been directly associated with the design, construction, and calibration of the machines include W. Egger, R. F. Wojcieszak, and J. H. Sams.

Instrumentation used throughout the calibration tests was, in general, the responsibility of V. J. McDonald, Research Assistant Professor of Civil Engineering.

In addition to those individuals named, the personnel of the Civil Engineering shop were instrumental in the development of these machines.

*Numbers in parenthesis refer to corresponding numbered entries in the Bibliography.
3. Notation

Although the notation is defined when it first appears in the text, the terms are assembled here for convenience in reference.

$A_a$ Area of an auxiliary piston

$A_t$ Area of main piston upon which the pressure in the top chamber acts

$A_b$ Area of main piston upon which the pressure in the bottom chamber acts

$A_o$ Area of orifice when open

$A_r$ Area of a slide valve rod

$A(t)$ Area of orifice at time, $t$

$F_o$ Force applied by machine without any movement of the piston

$F_\Delta$ Force applied by machine after the piston is moved the distance, $\Delta$

$F(t)$ Rapid load applied by machine at time, $t$

$g$ Gravitational acceleration

$k$ Ratio of specific heats of the gas

$K_o$ Initial force applied to the slide valve

$K(x,t)$ Force applied to the slide valve as a function of time and displacement

$m$ Mass

$P(x)$ Force applied to the slide valve from the gas in the auxiliary chambers

$P(x,t)$ Force applied to the slide valve from the gas in the main chamber

$p_{atm}$ Atmospheric pressure

$p_o$ Initial absolute pressure in a chamber

$p_{oe}$ Initial absolute pressure in the external chamber

$p_{fe}$ Final absolute pressure in both external and internal chambers without movement of the piston

$p(t)$ Absolute pressure at time, $t$

$p_i(t)$ Absolute pressure in the internal chamber at time, $t$

$p_e(t)$ Absolute pressure in the external chamber at time, $t$

$p_{o}$ Initial gage pressure in a chamber
\( p_{ce}^g \) Initial gage pressure in the external chamber

\( p_{ca}^g \) Initial gage pressure in the auxiliary chambers

\( p_{cm}^g \) Initial gage pressure in a main chamber

\( p_{ct}^g \) Initial gage pressure in the top chamber

\( p_{ob}^g \) Initial gage pressure in the bottom chamber

\( p_a^{(basic)} \) Basic gage pressure for the auxiliary system

\( p^g(t) \) Gage pressure at time, \( t \)

\( R \) Perfect gas constant

\( R_i, R_e, R_a \) Perfect gas constant for gas in the internal chamber, external chamber, and air respectively

\( T \) Absolute temperature

\( T_i, T_e, T_a \) Absolute temperature of gas in the internal chamber, external chamber, and air respectively

\( \tau \) Time measured from the time the slide valve begins to move

\( \tau_1 \) Time at which the slide valve begins to open the orifice

\( t \) Time measured from the time the slide valve begins to open the orifice

\( t_a \) Assumed time to evacuate the gas in the main chamber

\( t_f \) Time at which the orifice is fully open

\( t_c \) Time at which the pressure in the chamber reaches the critical pressure

\( V \) Volume of a chamber

\( V_o \) Initial volume of a main chamber before the piston is moved

\( V_e \) Volume of an external chamber

\( V_i \) Volume of an internal chamber (for Machine No. 3 \( V_i = V_t \) or \( V_b \))

\( V_b \) Volume of the bottom chamber

\( V_t \) Volume of the top chamber

\( \delta V \) Change in volume of a main chamber as a result of piston movement

\( \delta V_t \) Change in volume of the top chamber as a result of piston movement
\( \delta V_b \)  \( \text{Change in volume of the bottom chamber as a result of piston movement} \)

\( w(t) \)  \( \text{Weight of the gas at time,} \ t \)

\( v \)  \( \text{Acceleration} \)

\( x_o \)  \( \text{Distance which determines the initial volume of the auxiliary chambers} \)

\( x \)  \( \text{Distance the slide valve moves and consequently the distance the auxiliary pistons move.} \)

\( \eta \)  \( \text{Orifice coefficient} \)

\[
\beta = \left( \frac{2}{k+1} \right)^{1/k-1} \left( \frac{2kRT}{k+1} \right)^{1/2}
\]

\[
\lambda = -\eta \frac{\beta}{V} \frac{A_o}{2t_1} t^2
\]

\[
\lambda t_1 = -\eta \frac{\beta}{V} \frac{A_o}{2} t_1
\]

\[
\psi = -\eta \frac{\beta}{V} A_o (t - t_1)
\]
II SUMMARY AND CONCLUSIONS

4. Summary and Conclusions

Pneumatically operated testing machines, which use commercial nitrogen or helium as the pressure source, and associated apparatus have been developed. The machines are basically piston devices in which the load output is the result of differential pressure. Characteristics of machines No. 1 and No. 2 are as follows:

a. Stroke:

A maximum stroke of 18 inches is possible in either tension or compression.

b. Load:

A load having any selected value up to 62,800 lb can be applied either slowly or rapidly in either tension or compression. For a compressive loading, approximately 70 percent of the full load acts at the end of an 18-inch stroke or approximately 80 percent at the end of a 12-inch stroke.

c. Load Rise Time:

For a compressive loading using nitrogen as the pressure source the rise time will range between 0.005 and 0.018 seconds depending upon the pressure and piston position. For helium the compressive load rise time will range between 0.003 and 0.008 seconds.

d. Peak Duration:

The duration of the peak load may be varied from as little as 0.008 seconds to many hours.

e. Load Decay Time:

For a compressive unloading using nitrogen as the pressure source the load decay time will range between 0.005 and 0.043
seconds depending upon the pressure and piston position. For helium the compressive load decay time will range between 0.003 and 0.017 seconds. A system by which the time of unloading or loading may be extended is explained in this report.

The large external cylinders added to machine No. 3 reduce the noise of operation and also make it possible to use the machine in a relatively confined space by preventing an overpressure in the space. By initially pressurizing the internal chambers, machine No. 3 may be used in the same manner as machines No. 1 and No. 2; in which case it will have nearly the same characteristics. When the external cylinders are initially pressurized and the inner chambers pressurized by subsequent "implosion", the rise and decay times are considerably decreased from those for machines No. 1 and No. 2. For pulse loadings, the load drops off much more with piston displacement than for machines No. 1 and No. 2; although for step loadings, the load maintenance is nearly the same.

The 60-kip pulse loading machines described in this report have been used successfully in testing beams of both steel and reinforced concrete and in applying lateral loadings to steel columns. They have also been used in the study of beam to column connections and in the study of simple column base connections. All of the tests were conducted such that the specimen either collapsed or was deflected well into the strain hardening range of the specimen material.

The loading function for nearly all tests was essentially a step loading in which the load was applied within 0.015 to 0.025 seconds and maintained at approximately that level until the test was completed. For the test of a reinforced concrete beam a pulse loading was applied which had a rise time of
about 0.017 seconds, was maintained at a nearly constant level for about 0.023 seconds, and decayed in approximately 0.025 seconds.

A theoretical analysis is presented in which the machines are assumed to act against a rigid base. The few calibration tests which were conducted indicate that the theoretical solution is satisfactory for estimating the rise and decay time of the load when machines No. 1 and No. 2 are operated against a rigid base, although, there is some error involved, especially in the longer decay times. Since the present analysis is applicable only to the critical pressure which, for machine No. 3, occurs at about 50 percent of the applied load, the actual rise and decay times for this machine must be determined by an extensive theoretical analysis or by experiment.
III PULSE LOADING MACHINES NO. 1 AND NO. 2

5. General Description

The 60-kip pulse loading machines No. 1 and No. 2 shown in Appendix B, page 68, are basically piston devices in which the load output is the result of differential pressure. Commercially bottled nitrogen and/or helium is used as the energy source. The chamber volumes are relatively large so that a nearly constant load is maintained when the piston is moved through its full 18-inch stroke. The rapid application and release of load is achieved by using solenoid triggered slide valves to obtain timed pressure release from the two chambers of the device.

6. Loading Characteristics

These machines permit the application of a loading pulse that may begin from a "static" level ranging from 60-kips tension to 60-kips compression; undergo a rapid change of plus or minus 60 kips with the restriction that the prepulse load plus the dynamic change in load cannot exceed the limits of plus or minus 60 kips, and then return either slowly or rapidly to zero load. The duration of the peak load may be varied from a few milliseconds to many hours.

The machines consist of three basic components; the main piston assembly, the slide valve chamber assemblies, and the trigger assemblies which are described in more detail in the following paragraphs.

7. Main Piston Assembly

The main piston assembly consists of a main cylinder with a 10-inch internal diameter and a main piston and piston rod. An assembly is shown in Appendix B, page 71. The two chambers of the device are separated by the main
piston, and since the piston rod is continuous through only one chamber the effective area of the piston is less on that side. The piston areas are 78.54 in.\(^2\) and 70.24 in.\(^2\). Therefore, in order to obtain an initial zero load output of the potential rapid load, the pressures which produce the potential rapid load must differ on the two sides of the main piston. Hereafter the chamber with the larger piston area is referred to as the top main chamber and the chamber with the smaller piston area as the bottom main chamber. The ratio of the initial chamber pressures which result in zero load output is:

\[
\frac{P_{ob}}{P_{ot}} = \frac{A_t}{A_b} = 1.118
\]

(1)

where:

- \(A_b\) = Area of main piston upon which the pressure in the bottom chamber acts
- \(A_t\) = Area of main piston upon which the pressure in the top chamber acts
- \(P_{ot}\) = Initial gage pressure in the top chamber
- \(P_{ob}\) = Initial gage pressure in the bottom chamber

The maximum pressures to be used in the machines are 800 psi in the top main chamber and/or 895 psi in the bottom main chamber which can result in a maximum load of 62.8 kips in either tension or compression.

A plate at either end of the main cylinder with an internal diameter of 9 inches is used to stop the main piston from slipping out of the main cylinder and into the larger diameter slide valve cylinders attached to the main cylinder. The plate on the end of the main cylinder which has the piston rod also serves as an alignment plate for the second loading rod support located in the slide valve chamber assembly.

At either end of the machine the slide valve chamber assemblies and stop or alignment plates are bolted to flanges on the main cylinder. To
maintain the load within about 75% of its peak value during compression loading the storage chamber, which has a 13-inch internal diameter, is inserted between the stop plate and slide valve chamber assembly.

8. **Slide Valve Chamber Assemblies**

The slide valve chambers shown in Appendix B, page 77, are nominally 12 inches internal diameter. The slide valve orifices are located 2 3/16 inches from the ends of the chamber nearest the main cylinder and have a total area of approximately 82 in.\(^2\), however, the slide valve does not completely clear this area when in the open position. The effective area is approximately 55 in.\(^2\).

Two auxiliary pistons are connected to each slide valve by the slide valve rods. These pistons and rods with their respective applied pressures supply the force that moves the slide valve. Since the force applied to the slide valve is somewhat dependent on the variable main chamber pressures the pressures used in the auxiliary cylinders are adjusted so that the force on the slide valve is nearly a constant for any main chamber pressure.

Each auxiliary cylinder has a 4-in. internal diameter and an effective piston area of 11.35 in.\(^2\). The area of each slide valve rod is 1.23 in.\(^2\). Then, the initial force applied to the slide valve is:

\[
K_o = 2 \left[ p_g A_r + p_g A_a \right] \quad \text{Constant} \tag{2}
\]

where:
- \(A_a\) = Area of an auxiliary piston
- \(A_r\) = Area of a slide valve rod
- \(K_o\) = Initial force applied to the slide valve
- \(p_g\) = Initial gage pressure in the auxiliary chambers
- \(p_g\) = Initial gage pressure in a main chamber

For no main pressure:

\[
K_o = 2 A_a p_g \quad \text{Basic gage pressure for the auxiliary system} \tag{3}
\]

where: \(p_g\) = Basic gage pressure for the auxiliary system
The pressure in the auxiliary cylinders for any main chamber pressure must be:

$$p_{oa}^g = p_{a \text{(basic)}}^g - p_{om}^g \frac{A_r}{A_a}$$

(4)

The ratio of $A_r/A_a = 0.10$ is used since it is very nearly the correct value (0.108) and is an easy value to remember. A few tests were conducted to determine the best basic pressure to use in the auxiliary cylinders; however, these tests were conducted before the slide valves were altered. They indicated that there was very little variation in operating time for auxiliary pressures from 300 to 500 psi and that there was an increased operating time for less pressure. From this result a basic pressure of 380 psi was selected. With this basic auxiliary pressure, the initial auxiliary pressure may be determined from the following relationship:

$$p_{oa}^g = 380 - 0.10 \ p_{ot}^g$$

(5)

where: $p_{ot}^g$ has been substituted for $p_{om}^g$ since the gage for the bottom chamber does not measure pressure directly but measures a balance of the potential load.

The minimum initial force on the slide valve is approximately 8,620 lb and is obtained when the main chambers are not pressurized. Since the pressures in the main chambers differ because of the different effective areas of the main piston, the maximum force moving the slide valves differ depending on whether the slide valve being considered is in the top or bottom chamber. The maximum force on the slide valve in the bottom chamber is approximately 9,010 lb and the maximum force on the slide valve in the top chamber is approximately 8,780 lb.

When the slide valves are installed in the slide valve assemblies, they are set 1 l3/32 in. from the end of the slide valve cylinder nearest the main cylinder in order that the slide valve is centered over the orifices.
After the slide valve is centered correctly, the tappets on the end of the slide valve rods are adjusted so that they are in contact with the roller bearings on the link assembly of the trigger.

During the firing operation the slide valve moves 2 7/16 inches before it strikes the head, however, the gas in the auxiliary cylinders is allowed to evacuate through small orifices in the auxiliary cylinders after the slide valve has moved only 2 inches. The fact that no provision was made for absorbing the kinetic energy of the slide valves may eventually require the replacement of the slide valve rods.

9. **Trigger Assemblies**

The trigger assembly shown in Fig. 1 in the ready position and in Fig. 2 in the fired position consists of two slide valve restraining links, connecting bar, supporting frame, trigger pistons, sear assembly and solenoid. Energizing the solenoid starts the chain of events which terminate in the opening of the orifices for the evacuation of the gas from a main chamber. The sequence of operations of the triggers is as follows: 1) the solenoid is activated which 2) releases the catch or sear restraining the connecting bar and link assembly then, 3) the trigger pistons force the restraining links out of the way of the slide valve rods so that the slide valve can move resulting, 4) in the evacuation of the gas in a main chamber.

The actual release of the sear is slightly more complex, in that during the first half inch of travel of the solenoid core, the core releases a safety device that prevents the sear from becoming disengaged until the solenoid is energized. At the end of the one-half inch of travel a small mass on the actuating link, which is connected to the solenoid core, comes into contact with the sear and through the remaining 1/4 inch of movement of the solenoid core the sear is disengaged. The actuating link is held in position by a friction spring and the sear is held into the locked position by the reset spring.
When resetting the trigger, after the slide valves have been repositioned, the safety must be moved out of the way in order that the sear can be moved to allow the slide valve restraining link assembly to be repositioned.

Each trigger piston has a one-inch diameter and the pressure applied to the trigger pistons is the same as that applied to the auxiliary pistons. The normal force on the sear, neglecting any eccentricity of the restraining link assembly with the slide valve rods, is approximately 370 to 465 lbs. The frictional force against which the solenoid would have to act in a conventional design would be 46.5 lb, assuming a coefficient of friction of 0.10. The solenoids required to release this force would either be excessively large or require a very long stroke. In order to circumvent this problem two things have been done. 1) The face of the sear was made such that the sum of the normal and frictional force moments about the center of rotation of the sear is zero. Thus, theoretically, no force is required to disengage the sear. (When this system was first tried, the jar resulting from firing one end of the machine was sufficient to automatically fire the other end; which is one reason the safety device was added to the trigger mechanism.) 2) A short duration high amplitude power supply is used for the solenoids which is slightly more consistent than the power supply specified for the solenoids. The electronic diagram of the power supply is shown in Fig. 3. A pulse from the solenoid power supply is also put on the timing trace in order to have a record of the time delay between energizing the solenoids and the time the load is applied.

Two microswitches have also been included in the triggering mechanism. These switches are connected into the circuit of the timing trace and when activated by the movement of the connecting bar the timing trace is displaced slightly. Thus a record is obtained of the time delay between energizing the solenoid and the movement of the connecting bar.
In order to prevent a reverse firing by the machine for a compressive loading pulse, another switch has been installed in the circuit of the top solenoid. This switch is connected to the trigger and activated by the movement of the bottom connecting bar. Using this safety switch a pulse loading with a 0.020 second delay between energization of the solenoids can be obtained.

The microswitches and the safety switch are indicated in the wiring diagram in Fig. 3.

10. Supplementary Systems

In addition to the loading device described in the previous section several supplementary systems are required for the operation of the machines. These include the pressurizing system, the sequence control unit, and the test frame for supporting the loading machine and the test specimens.

The pressurizing system consists of the gas manifolds, the control panel, and the tubing required to connect the loading machines to the gas supply. At present, two manifolds are being used, making it possible, if desired, to supply both nitrogen and helium simultaneously. Portions of the manifolds, each of which has a capacity of 4 cylinders of compressed gas, are shown in Figs. 4 and 5. The gas supply at each manifold is regulated by a pressure regulating valve so that only the manifold is subjected to the full cylinder pressure.

From the manifold the gas passes through the control panel shown in Figs. 5 and 6. The needle valves in the control panel allow the component parts of the machines to be pressurized from individual gas supplies or interconnected in any desired manner. Provisions also have been made for the interconnection of several control panels so that more than one machine can be supplied from the same manifold.

Control of the gas flow to the loading machines is obtained by means of the line valves shown in Fig. 6. Both the supply and the gage lines are
provided with bleeder lines. The gage for the bottom main chamber has been re-calibrated so that it indicates the same pressure as the top main chamber gage, when there is a zero load output.

The control of the sequence of operations during a test is obtained with a ten-channel timing system manufactured by Electro-Pulse shown in Fig. 4. Each single operation can be preset to occur at a fixed time, from 0.000 to 9.999 seconds from initiation of the timer. In Fig. 6 four channels of the timer are shown diagramatically with the operations which they initiate in a typical test.

The supporting frame for the loading machines consists of two A-frames bolted to a rectangular horizontal base frame. The frame, shown in Fig. 7, was designed to provide for the testing of both beams and frames. The proportions of the components of the frame are such that the frame should be capable of providing a reasonably rigid support for a test in which two loading machines, suspended horizontally, each apply a lateral load of 50 kips to a frame specimen. For beam type specimens the loading machine is mounted vertically in the supporting frame.

The clear distance between the A-frames is about 40 inches to allow sufficient space for one to work on the instrumentation for a specimen and to be able to use machine No. 3, described in the next section, in the same supporting frame. Since the connections provided on the main cylinders were only approximately 24 inches apart, spacers are used on either side of the machines to attach the machines to the supporting frames. These spacers were constructed so that it is possible to rotate the machine through 360 degrees making it a much easier task to place the machines into or remove them from the frame. The spacers have a bolted friction joint which ordinarily prevents rotation of the machine, however, if the transverse load on the loading rod becomes excessively
large, the joint will slip allowing the machine to rotate and reducing the possibility of damage to the machine. During the testing of a concrete beam, the loading point of the beam did translate and no damage was observed in the loading rod. The spacer joints are shown in Fig. 7 attached to the machine and frame.
IV PULSE LOADING MACHINE NO. 3

11. General Description

The component parts of Pulse Loading Machine No. 3 are identical to Machines No. 1 and No. 2 except that large 36-in. O.D. cylinders have been clamped around each of the slide valve chambers and the storage chamber is not used. Special "O" rings at the head and at the flange of the main cylinder and around each bolt seal the chamber from the atmosphere. The machine can be seen in the background in Fig. 7.

12. Loading Characteristics

When Machine No. 3 is pressurized in the same manner as are Machines No. 1 and No. 2 and operated in the same way (by evacuating the gas from the main chambers), it has nearly the same loading characteristics as the other two machines. The only advantage that Machine No. 3 has when operated in this manner is to reduce the noise of operation. Since the critical pressure is reached sooner than when the gas is evacuated into the atmosphere, the rise and decay time of the load are slightly greater than for Machines No. 1 and No. 2. The maximum load is slightly smaller because the gas is released into a confined space and the load decays more as a result of piston movement than do Machines No. 1 and No. 2. Because of the greater initial pressure and initial volume which is usually associated with the bottom chamber, a tension force remains after both the upper and lower chambers have been evacuated. This tension force must be restrained in some manner or it may cause the machine to be damaged.

A more desirable performance is exacted from the machine when the external cylinders are initially pressurized and the gas is "imploded" onto the main piston. When the machine is operated in this manner, a loading
pulse may be applied which may begin from a "static" level ranging from approximately 60 kips tension to approximately 60 kips compression by pressurizing one of the internal chambers. Upon imploding the gas the load can undergo either one or both of two rapid changes: (a) a rapid change in load of the same sign as the prepulse load with the restriction that the prepulse load plus the dynamic change in load cannot exceed a 60 kip range in tension or compression, or (b) a rapid change in load ranging from 0 to 60 kips which is of opposite sign to the prepulse load. When reference is made to the loading characteristics of Machine No. 3 in this report they are the characteristics which result from operating the machine as an "implosion" machine.

Because only a small amount of gas moves from one chamber to the other the rise and decay times are about 1/4 to 1/2 of those for Machines No. 1 and No. 2. Testing schedules to date have made it impossible to calibrate this machine; however, some theoretical work and one preliminary proof test have been conducted. In the proof test the loading rod acted against a hydraulic ram which presumably caused the oscillations of the load shown in Fig. 8.

Section 12 was revised on 28 July 1958 from that which was presented in report AFSWC-TR-57-22.
13. Purpose and Scope

To use the loading machines in a test program in which the test specimens are to be subjected to a preset load pulse, two aspects of the operation of the loading machines must be evaluated: 1) the rise and decay times of the load and 2) the time required for the operation of the trigger mechanism. The time of operation of the trigger is measured from the time the solenoids are energized to the time the microswitches are activated and to the time the loading or unloading process begins. In addition to determining the operating time, an estimate of the variation of the operating time and rise time must be obtained in order to be able to determine the minimum possible time interval between loading and unloading.

The machines were initially calibrated in July 1955; but, the triggering mechanism and the slide valve design proved to be inadequate in providing a consistent time of operation. The slide valves and trigger systems were altered until a satisfactory operation was obtained. Because of the time and expense involved in altering the machines and the fact that an experimental testing program was in progress, a smaller number of calibration tests were made rather than the 40 tests previously conducted.

Four calibration tests were conducted involving the application and removal of the load. Only nitrogen was used as the pressurizing medium in the calibration tests since helium gas was virtually taken off of the market between the time the two series of calibration tests were conducted. The previous calibration tests and the analytical work which has been done indicates that the rise and decay times for helium are approximately one-half of those for nitrogen. Since the time in which the load is applied or released is more a function of the
chamber volume than of the initial pressure except for very small pressures, initial pressures of 400 psi in the top chamber and 476 in the bottom chamber were used for all of the tests and the chamber volume was varied. The volumes of the chambers were governed by the following conditions:

Test 1. 18-inch compression stroke available, with storage chamber
Test 2. 18-inch compression stroke available, without storage chamber
Test 3. 0-inch compression stroke available, without storage chamber
Test 4. 0-inch compression stroke available, with storage chamber.

Since only the piston position governs the volume in the lower chamber two records are obtained for the extreme volumes of this chamber and serve as an indication of the repeatability of the system.

14. Instrumentation

Two load dynamometers were connected in series with the loading rod and rigidly attached to the base beam of the frame. Strains were measured in both load dynamometers by means of SR4 strain gages connected into 4 arm bridge circuits. The upper dynamometer, which had previously been calibrated in a standard commercial testing machine, was used to calibrate the lower dynamometer which was connected to continuous recording equipment. The upper dynamometer was also used to obtain a check on the magnitude of the load before the test, after the gas in the lower chamber was evacuated, and after the test was completed. Various shunt resistors were placed across one arm of the bridge of the lower dynamometer before and after each test. The load which these shunt resistors represented was determined by comparing the trace deflection that they produced with the trace deflection obtained from the dynamometer when known loads were applied.

15. Results of Calibration Tests

Each of the load-time relationships obtained in the calibration tests is reproduced in Figs. 9 through 14. Before each rise or decay of the load, for
machines No. 1 and No. 2, a deflection of the trace occurs opposite to the trace deflection which occurs when the actual load is applied or released. For machine No. 3 a similar trace deflection occurs before each rise or decay; however, it is in the same direction as the deflection produced by applying or releasing the load as shown in Fig. 8, indicating that this initial trace deflection is probably a result of the restraining link assembly striking the top beam of the trigger frame.

Because of the small oscillations of the load trace around zero and the maximum load, the rise and decay times of the calibration tests are defined as the time interval between the time at which the load begins to change, neglecting the oscillations, to the time at which the load corresponding to the critical pressure is reached. These times are summarized in Table 2 and a comparison is made with the rise and decay times obtained from the theoretical analyses. The minimum rise time is about 0.007 seconds with no available stroke and the maximum is approximately 0.014 second with an available stroke of 18"; whereas, the decay time may vary from about 0.008 seconds to approximately 0.030 seconds depending on the piston position and whether or not the storage chamber is used.

The operating times of the trigger mechanisms will vary with each trigger but, in general, the microswitches will be activated in approximately 0.017 seconds and the beginning of the load change will occur about 0.034 seconds after energization of the solenoids. Since the shortest possible time between energizing the solenoids is 0.020 seconds, the resultant pulse will have a plateau of about 0.008 to 0.022 seconds depending on the rise time and the variation in operating time as shown in Fig. 15. If a plateau on the loading pulse is not desired a mechanical safety should be used to prevent a reverse movement of the piston and the time of operation should be estimated by the delay times obtained from a proof test in which the main chambers are not pressurized. The pulses
shown in Figs. 16 and 17 each had a 0.020 second delay between energization of the solenoids. The oscillograms indicate the type of pulse load which can be applied by the machines and the difference between the load rise of the pulse for helium and nitrogen gas. The load decay for both tests was a result of evacuating nitrogen gas from the top chamber. In the figures, the period of the timing trace is 0.002 seconds per cycle and the peak load is approximately 35 kips compression.
VI CHARACTERISTICS OF MACHINES WITH PISTON DISPLACEMENT

16. Load Rise-Time Characteristics

When the machines are used with a flexible specimen the resulting acceleration of the piston and piston rod delays the time in which the load is applied. In Fig. 18 the load and deflection record obtained from the test of a 4WF beam are reproduced\(^{(3)}\). The force output of the gas at the piston was partially resisted by the specimen and its inertia force, and partially resisted by the friction and inertia force of the piston and piston rod of the machine. The instantaneous inertia force of the piston and piston rod plus any frictional forces must be added to the instantaneous recorded force in order to obtain a load-time relationship at the piston that can be compared to the load-time relationship of the calibration tests or of a theoretical analysis. From the acceleration record of this test the inertia force of the piston rod was determined and is presented in the figure. Although frictional forces have been neglected and the precise initial piston position was not recorded, the rise time is about as would be expected for this test.

17. Maintenance of Load

Since no additional gas is introduced into the machine, the load decreases when the piston moves as a result of specimen deflection. For machines No. 1 and No. 2 the ratio of the load for any movement of the piston to the initial applied load may be obtained from the following relationship:

\[
\frac{F_\Delta}{F_o} = \frac{V_o}{V_o + \delta V} - \frac{p_{\text{atm}}}{p_{\text{cm}}} \left( \frac{\delta V}{V_o + \delta V} \right) \quad (6)
\]

where:

- \(F_o\) = Force applied by machine without any movement of the piston
- \(F_\Delta\) = Force applied by machine after the piston is moved the distance, \(\Delta\)
\[ P_{\text{atm}} = \text{Atmospheric pressure} \]

\[ V_0 = \text{Initial volume of a main chamber before the piston is moved} \]

\[ \delta V = \text{Change in volume of a main chamber as a result of piston movement} \]

For an 18-inch stroke and initial compressive loads of 7.85 kips (100 psi in the top chamber) or 62.8 kips (800 psi in the top chamber), the final loads are 69 or 72 percent of the initial loads respectively. For a 12-inch stroke in which the initial piston position is such that the full 18-inch stroke is available, the final loads are 82 or 84 percent of the initial loads respectively.

For machine No. 3 the ratio of the compressive load at a specific piston displacement to the initial compressive load can be obtained from the following relationship:

\[
\frac{F_A}{F_0} = \frac{V_t + V_e}{V_t + V_e + \delta V_t} - \frac{P_{\text{atm}}}{P_{\text{poe}}} \left( \frac{V_t + V_e}{V_t + V_e + \delta V_t} \right) \left( \frac{\delta V_t}{V_e} \right)
\]

\[
= \left[ \frac{P_{\text{atm}}}{P_{\text{poe}}} \left( \frac{A_b}{A_t} \right) \left( \frac{\delta V_b}{V_b + \delta V_b} \right) \left( \frac{V_t + V_e}{V_e} \right) \right] \left( \frac{\delta V_t}{V_t} \right)
\]

where:

\[ \frac{\delta V_t}{V_t} = \text{Initial gage pressure in the external chamber} \]

\[ V_e = \text{Volume of an external chamber} \]

\[ V_b = \text{Volume of the bottom chamber} \]

\[ V_t = \text{Volume of the top chamber} \]

\[ \delta V_t = \text{Change in volume of the top chamber as a result of piston movement} \]

\[ \delta V_b = \text{Change in volume of the bottom chamber as a result of piston movement} \]

The term in brackets is a reduction in load caused by the increase in pressure in the bottom chamber as a result of the decreased volume. Leaving the slide
valve orifice open in the bottom chamber will greatly increase the load maintenance characteristics of the machine. For this condition \( \frac{8V_b}{V_b + 8V_b} \) becomes

\[
\frac{8V_b}{V_b + V_e + 8V_b}.
\]

For initial pressures of 100 or 800 psi in the top external chamber the initial loads for machine No. 3 are 7.23 kips and 57.7 kips respectively for an initial piston position in which a full 18-inch stroke is available. After the piston moves 18 inches the final loads are 50 and 80 percent respectively of the initial load with the bottom slide valve closed and 76 and 82 percent respectively of the initial load with the bottom slide valve open.
18. Theoretical Load Rise and Decay for Machines No. 1 and No. 2

The loading or unloading process of the 60-kip pulse loading machines involves the release through an orifice of a gas confined in a chamber. The solution of the time-pressure relationship is simplified by assuming that the gas is ideal, the expansion of the gas is an adiabatic process and the temperature of the gas in the chamber remains constant throughout the process. The solution must also include the fact that the orifice area varies with time.

With the assumption that the gas being released is of one type and ideal (Actually the gas is a mixture of air, which was initially in the chamber at atmospheric pressure, and either nitrogen or helium which is introduced into the chamber. For nitrogen this assumption is satisfactory; however, some error, which has not been evaluated, is introduced when helium gas is used as the pressurizing medium.), the pressure, volume, weight of gas in the chamber, and the temperature of the gas are related by the perfect gas equation:

\[
\frac{p^a(t)}{V} = \frac{RT}{V} w(t) \\
0 \leq t \leq t_c
\]  

(8)

where:

\( p^a(t) \) = Absolute pressure at time, t

\( R \) = Perfect gas constant

\( T \) = Absolute temperature

\( t_c \) = Time at which the pressure in the chamber reaches the critical pressure

\( V \) = Volume of a chamber

\( w(t) \) = Weight of the gas at time, t

Since only the weight and pressure are functions of time the relationship can be written:

\[
\frac{dp}{dt} = \frac{RT}{V} \frac{dw}{dt} \\
0 \leq t \leq t_c
\]  

(9)
For an isentropic flow of gas through an orifice and for \( p(t) \) greater than the critical pressure, the rate of flow can be expressed as:

\[
\frac{dw}{dt} = -\eta A(t) p^a(t) \left( \frac{2}{k+1} \right)^{1/k-1} \left[ \frac{2gk}{RT(k+1)} \right]^{1/2} \quad 0 \leq t \leq t_c \tag{10}
\]

where:
- \( A(t) \) = Area of orifice at time, \( t \)
- \( g \) = Acceleration of gravity
- \( k \) = Ratio of the specific heats of the gas
- \( \eta \) = Orifice coefficient

Substituting (10) in (9), the equation expressing the pressure as a function of time is:

\[
\frac{dp}{dt} = -\eta \frac{A(t) p^a(t)}{V} \left( \frac{2}{k+1} \right)^{1/k-1} \left( \frac{2gkRT}{k+1} \right)^{1/2} \quad 0 \leq t \leq t_c \tag{11}
\]

Let \( \beta = \left( \frac{2}{k+1} \right)^{1/k-1} \left( \frac{2gkRT}{k+1} \right)^{1/2} \), a constant which is dependent only on the type of gas. Simplifying equation (11):

\[
\frac{dp}{dt} = -\eta \frac{\beta}{V} A(t) p^a(t) \quad 0 \leq t \leq t_c \tag{12}
\]

The solution of this differential equation is:

\[
p^a(t) = C e^{-\eta \frac{\beta}{V} \int A(t) \, dt} \quad 0 \leq t \leq t_c \tag{13}
\]

Where \( C \) is determined by the initial conditions in the chamber:

\[
C = p_o^a \quad \text{the initial absolute pressure in the chamber at } t = 0.
\]

Now (13) can be written:

\[
p^a(t) = p_o^a e^{-\eta \frac{\beta}{V} \int A(t) \, dt} \quad 0 \leq t \leq t_c \tag{14}
\]

The orifice opening as a function of time can be obtained from the movement of the slide valve as a function of time. Neglecting any frictional forces, the force \( K(x,t) \) which moves the slide valve system is the result of
1) the force $P(x)$ caused by the pressure on the auxiliary pistons, and 2) the force $P(x, t)$ caused by the pressure in the main chamber on the slide valve rods.

As the volume of the auxiliary chambers is increased by the movement of the auxiliary pistons the force $P(x)$ is reduced, since no additional gas is introduced into the auxiliary cylinders. This force can be expressed approximately:

$$P(x) = 2 A \frac{p_g^B}{x + x_0} 0 \leq x \leq 2.125 \quad (15)$$

where:

$P(x)$ = Force applied to the slide valve from the gas in the auxiliary chambers

$x_0$ = Distance which determines the initial volume of the auxiliary chambers

$x$ = Distance the slide valve moves and consequently the distance the auxiliary pistons move.

The force on the slide valve system from the slide valve rods is a function of the pressure in the main chamber. Since this force is only a small part of the total force on the slide valve system, a rough approximation is satisfactory.

The pressure on the slide valve rods remains constant until the slide valve begins to open. After the slide valve begins to open the pressure in the chamber may be assumed to vary linearly with time and to dissipate in the time, $t_a$.

Then:

$$P(x, t) = 2 A \frac{p_g^B}{x + x_0} \quad \text{at} \quad x = 0.813; \quad \tau = \tau_1 \quad (16)$$

$$P(x, t) = 2 A \frac{p_g^B \left( \frac{\tau_1 + t_a - \tau}{t_a} \right)}{x} \quad 0.813 \leq x \quad \text{and} \quad \tau_1 \leq \tau \leq \tau_1 + t_a \quad (17)$$

$$P(x, t) = 0 \quad \tau_1 + t_a \leq \tau \quad (18)$$
where: \( P(x,t) \) = Force applied to the slide valve from the gas in the main chamber

\( \tau \) = Time measured from the time the slide valve begins to move

\( \tau_1 \) = Time at which the slide valve begins to open the orifice

\( t_a \) = Assumed time to evacuate the gas in the main chamber

The expression \( m\ddot{x} = K(x,t) \) was solved by numerical integration for the time-displacement relationships of the slide valve for the following conditions: no pressure in the main chamber, 800 psi in the top chamber and 895 psi in the bottom chamber. Since the time-displacement relationships of the slide valve are very nearly linear over the length of the orifice the relationships were linearized and a constant velocity of 342 in./sec. was assumed for the slide valve during the opening of the orifice. These relationships are presented in Fig. 19.

With the constant velocity of the slide valve the area of the orifice can now be expressed as:

\[
\int A(t) \, dt = \frac{A_o}{\tau_1} t^2 \quad 0 \leq t \leq \tau_1 \quad (19)
\]

\[
\int A(t) \, dt = A_o (t - \tau_1) \quad \tau_1 \leq t \leq \tau_c \quad (20)
\]

where: \( A_o \) = Area of orifice when open

\( t \) = Time measured from the time the slide valve begins to open the orifice

\( \tau_1 \) = Time at which the orifice is fully open

The expression for the pressure in the chamber can now be written as:

\[
P_o^a(t) = \frac{p_o^a}{e^{\lambda}} \quad 0 \leq t \leq \tau_1 \quad (21)
\]

\[
P_o^a(t) = \frac{p_o^a}{e^{\lambda \tau_1}} e^\psi \quad \tau_1 \leq t \leq \tau_c \quad (22)
\]
\[ \lambda = -\eta \frac{\beta}{V} \frac{A_0}{2} t_1^2 \]  \hspace{2cm} (23)

\[ \lambda t_1 = -\eta \frac{\beta}{V} \frac{A_0}{2} t_1 \]  \hspace{2cm} (24)

and:

\[ \psi = -\eta \frac{\beta}{V} A_0 (t - t_1) \]  \hspace{2cm} (25)

Since these pressures are absolute pressures and the load measurement is a result of only the applied or gage pressures \( p^a = p^g + p^{atm} \), the relationships must be solved for the gage pressures.

\[ p^g(t) = p^g e^{t \lambda} - p^{atm} (1 - e^{t \lambda}) \] \hspace{2cm} 0 \leq t \leq t_1 \hspace{2cm} (26)

\[ p^g(t) = p^g e^{t_1 \lambda} e^{\psi} - p^{atm} (1 - e^{t_1 \lambda} e^{\psi}) \] \hspace{2cm} t_1 \leq t \leq t_c \hspace{2cm} (27)

where:

\[ p^g(t) = \text{Gage pressure at time, } t \]

For a compressive loading by the machines both chambers are pressurized in such a way that initially there is no unbalance of the potential rapid load. Then the gas in the bottom chamber is released and the gas confined in the top chamber applies the load.

For compressive loading the expressions for the load become:

\[ \frac{F(t)}{F_0} = (1 + \frac{p^{atm}}{p^{ob}}) (1 - e^{\lambda}) \] \hspace{2cm} 0 \leq t \leq t_1 \hspace{2cm} (28)

\[ \frac{F(t)}{F_0} = (1 + \frac{p^{atm}}{p^{ob}}) (1 - e^{t_1 \lambda} e^{\psi}) \] \hspace{2cm} t_1 \leq t \leq t_c \hspace{2cm} (29)

where:

\[ F(t) = \text{Rapid load applied by machine at time, } t \]

For compressive unloading, the expressions for the load become:

\[ \frac{F(t)}{F_0} = e^{t \lambda} - \frac{p^{atm}}{p^{ot}} (1 - e^{\lambda}) \] \hspace{2cm} 0 \leq t \leq t_1 \hspace{2cm} (30)

\[ \frac{F(t)}{F_0} = e^{t_1 \lambda} e^{\psi} - \frac{p^{atm}}{p^{ot}} (1 - e^{t_1 \lambda} e^{\psi}) \] \hspace{2cm} t_1 \leq t \leq t_c \hspace{2cm} (31)
19. Theoretical Rise and Decay for Machine No. 3

The load-time characteristics of machine No. 3 can be obtained by relating the weight of the gas in the chambers at any time to the initial weight of the gas.

\[
\frac{p_1(t) V_i}{R_1 T_1} + \frac{p_e(t) V_e}{R_e T_e} = \frac{p_oe V_e}{R_e T_e} + \frac{p_{atm} V_i}{R_a T_a}
\]  

where:
- \( p_1(t) \) = Absolute pressure in the internal chamber at time, \( t \)
- \( p_e(t) \) = Absolute pressure in the external chamber at time, \( t \)
- \( p_oe \) = Initial absolute pressure in the external chamber
- \( R_i, R_e, R_a \) = Perfect gas constant for gas in the internal chamber, external chamber and air respectively
- \( T_i, T_e, T_a \) = Absolute temperature of gas in the internal chamber, external chamber and air respectively
- \( V_i \) = Volume of an internal chamber

Assuming that the gas is of one type throughout the chambers, the gas constant can be eliminated and, assuming a constant temperature, the pressure in the internal chamber which determines the loading function reduces to:

\[
p_1(t) = \frac{V_e}{V_i} \left[ \frac{p_oe - p_e(t)}{p_e(t)} \right] + p_{atm}
\]  

This relationship satisfies the initial condition of \( p_1(t) = p_{atm} \) and the final condition in which \( p_1(t) = p_{fei} \) the final gas pressure in both the internal and external chambers without movement of the piston.

The load-time relationships for machine No. 3 are obtained in a similar way to the relationships for machines No. 1 and No. 2 and are as follows:

For compressive loading:
For compressive unloading:

\[
\frac{F(t)}{F_0} = (1 + \frac{P_{atm}}{P_{oe}}) \left(1 + \frac{V_e}{V_t}\right) (1 - e^{\lambda t}) \quad 0 \leq t \leq t_1
\]  

(34)

\[
\frac{F(t)}{F_0} = (1 + \frac{P_{atm}}{P_{oe}}) \left(1 + \frac{V_e}{V_t}\right) (1 - e^{\lambda t}e^{\psi}) \quad t_1 \leq t \leq t_c
\]  

(35)

Since the gas in the external chambers is evacuated into a confined space, the critical pressure is much larger for machine No. 3 than for machines No. 1 and No. 2. Although the analysis does not hold for more than about one half of the maximum load, because of the critical pressure being reached, it does indicate that the load rises and decays much more rapidly for this machine than for machines No. 1 and No. 2.

20. **Approximate Analysis for Machines**

If, in the relationships derived for the load-time characteristics for the machines, the terms involving the atmospheric pressure are eliminated, the relationships become independent of the applied pressure. The procedure is equivalent to assuming that the fundamental gas laws are based on gage pressure and not absolute pressure. The load-time relationships for this condition follow:

For Machines No. 1 and No. 2
For compressive loading:

\[
\frac{F(t)}{F_0} = 1 - e^{\lambda t} \quad 0 \leq t \leq t_1 \quad (38)
\]

\[
\frac{F(t)}{F_0} = 1 - e^{\lambda t_1 e^{\psi}} \quad t_1 \leq t \quad (39)
\]

For compressive unloading:

\[
\frac{F(t)}{F_0} = e^{\lambda t} \quad 0 \leq t \leq t_1 \quad (40)
\]

\[
\frac{F(t)}{F_0} = e^{\lambda t_1 e^{\psi}} \quad t_1 \leq t \quad (41)
\]

For machine No. 3

For compressive loading:

\[
\frac{F(t)}{F_0} = (1 + \frac{V_e}{V_t}) (1 - e^{\lambda t}) \quad 0 \leq t \leq t_1 \quad (42)
\]

\[
\frac{F(t)}{F_0} = (1 + \frac{V_e}{V_t}) (1 - e^{\lambda t_1 e^{\psi}}) \quad t_1 \leq t \quad (43)
\]

For compressive unloading:

\[
\frac{F(t)}{F_0} = 1 - (1 + \frac{V_e}{V_b}) (1 - e^{\lambda t}) \quad 0 \leq t \leq t_1 \quad (44)
\]

\[
\frac{F(t)}{F_0} = 1 - (1 + \frac{V_e}{V_b}) (1 - e^{\lambda t_1 e^{\psi}}) \quad t_1 \leq t \quad (45)
\]

21. **Summary of Theoretical Analyses**

The values of the various constants appearing in the load-time relationships for the machines are summarized in Table 1. With these expressions and constants, the load-time characteristics of the machines were obtained for piston positions of 18 and 0 inches for both helium and nitrogen gas as the pressurizing medium and nominal initial gas pressures of 100, 400, and 800 psi.
The approximate load-time relationships for machines No. 1 and No. 2 were also obtained. The relationships are presented in Figs. 20 through 26.

For machines No. 1 and No. 2 the rise and decay times, which can be obtained with the machines operating against a rigid system are presented in Table 2, where the time is measured from the time the load begins to change to the time at which the load reaches the critical pressure.

Since the analysis does not hold beyond the critical pressure, which occurs at approximately 50 percent of the final load when the external cylinders are initially pressurized, the load-time relationships are included only as an indication of the rise and decay times. The rise or decay time for machine No. 3 is probably less than ten milliseconds since the initial rise and decay of the load is much more rapid under all conditions of volume and pressure than those for machines No. 1 and No. 2.
BIBLIOGRAPHY


Table 1

Constants Used In The Theoretical Analysis

\[ A_c = 55 \text{ in}^2 \]
\[ A_t = 78.54 \text{ in}^2 \]
\[ A_b = 70.24 \text{ in}^2 \]
\[ V_e = 6270 \text{ in}^3 \]
\[ 8V_t = A_t \text{ (stroke)} \]
\[ 8V_b = A_b \text{ (stroke)} \]
\[ \eta = 1.0 \]
\[ g = 32.2 \text{ ft/sec}^2 \]
\[ t_1 = 0.00475 \text{ sec} \]
\[ T = 460 + T_{oF} \text{ degrees Rankine (°R)} \]

For Nitrogen Gas:          For Helium Gas:
\[ k = 1.4 \]                  \[ k = 1.659 \]
\[ R = 55.2 \text{ ft/°R} \]   \[ R = 386 \text{ ft/°R} \]
\[ \beta = 660 \text{ ft/sec} \] \[ \beta = 1870 \text{ ft/sec} \]

Basic Volumes With An 18 Inch Compression Stroke Available
\[ V_b = 1880 \text{ in}^3 \]
\[ V_t = 570 \text{ in}^3 \text{ without storage chamber} \]
\[ V_{t'} = 3780 \text{ in}^3 \text{ with storage chamber} \]

Basic Volumes For Other Piston Positions
\[ V_b = 1880 - A_b \text{ (18 - available stroke)} \]
\[ V_t = 570 + A_t \text{ (18 - available stroke)} \text{ without storage chamber} \]
\[ V_{t'} = 3780 + A_t \text{ (18 - available stroke)} \text{ with storage chamber} \]
Table 2

<table>
<thead>
<tr>
<th>Gas</th>
<th>Compression Load, kips</th>
<th>Available Compression Stroke, Inches</th>
<th>Pressure in Evacuating Chamber, psi</th>
<th>Per Cent of Load at Critical Pressure</th>
<th>Theoretical Rise Time, 0.001 sec</th>
<th>Experimental Rise Time, 0.001 sec</th>
<th>Pressure in Evacuating Chamber, psi</th>
<th>Per Cent of Load at Critical Pressure</th>
<th>Theoretical Decay Time, 0.001 sec</th>
<th>Experimental Decay Time, 0.001 sec</th>
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Fig. 1 Trigger Assembly in Ready Position

Fig. 2 Trigger Assembly in Fired Position
Fig. 3 Circuit Diagram for Solenoids, Microswitches, Safety Switch, Timer, and Oscillographs

NOTE
1 Motor Driven Switches Modulate Amplitude Of Timing Signal.
2 DC Used On Slow Tests; AC Of Known Frequency Used On Rapid Tests.
3 TI Serves To Insert Step When Solenoids Are Activated. Steps Are In Opposite Directions.
4 Top And Bottom Microswitches Operated By Movement Of Respective Connecting Bars Of Triggers.
5 Charging Circuits Supply Approximately 25 Amperes For 0.020 Seconds To Fire Solenoids.
6 Safety Switch Operated By Movement Of Connecting Bar Of Bottom Trigger.
Fig. 4 Gas Manifolds and Timer

Fig. 5 Gas Manifolds, Control Panel and Oscillographic Recording Instruments
Fig. 6 Diagram of the Control Panel
Fig. 7 Testing Frame and Machines No. 1 and No. 3
Fig. 8 Proof Test Load Record—Machine No. 3
Fig. 9 Load Rise--Time Relationship Machine No. 1
Fig. 10 Load Rise--Time Relationship Machine No. 1
Fig. 11 Load Decay--Time Relationship Machine No. 1
LOAD-TIME RELATIONSHIP
Nitrogen Gas
0° Piston Position
Without Storage Chamber
Theoretical
Experimental

Fig. 12 Load Decay--Time Relationship Machine No. 1
Fig. 13  Load Decay--Time Relationship Machine No. 1
Fig. 14  Load Decay--Time Relationship Machine No. 1
Fig. 15 Operating Time of Machine No. 1 Showing Variations in Rise and Decay Time and Possible Variation in a Pulse Loading Safe from Reverse Firing.
Fig. 16 Typical Oscillogram of Pulse Loading

Fig. 17 Typical Oscillogram of Pulse Loading
Fig. 18 Example of Load, Deflection, and Inertia Force from a Typical Test of a Flexible Specimen
2.8
0 psi Top or Bottom
800 psi Top
895 psi Bottom
Slide Valve Strikes Head

Average Velocity: 342 in./Sec
Time For Orifice To Open: 0.00475 Sec
Auxiliary Orifice Opens

Slide Valve Begins To Clear Orifice

Fig. 19 Time-Displacement Relationships of Slide Valves
Fig. 20 Theoretical Load-Time Relationships for Machines No. 1 and No. 2
Fig. 21 Theoretical Load-Time Relationships for Machines No. 1 and No. 2
Fig. 22 Theoretical Load-Time Relationships for Machines No. 1 and No. 2
Fig. 23 Theoretical Load-Time Relationships for Machines No. 1 and No. 2
Fig. 24 Theoretical Load-Time Relationships for Machines No. 1 and No. 2
Fig. 25 Theoretical Load-Time Relationships for Machines No. 1 and No. 2
Fig. 26 Theoretical Load-Time Relationships for Machine No. 3
APPENDIX A  SPECIFICATIONS FOR MATERIALS AND COMPONENTS

1. Steel:
   a. Cylinders: API plain end seamless pipe, Grade B, Minimum Test Pressure 1000 psi.
   b. Heads, Flanges, Rolled Shapes, etc.: ASTM A-7
   c. Rods and special nuts: AISI 4130, normalized

2. Aluminum Plate:
   All aluminum components fabricated from 6061 wrought alloy in T6 temper.

3. Bolts:
   American Standard Semifinished Hex. Head; Conforming to ASTM specification A325-49T.

4. Seals:
   Linear "O" ring seals, style LC A 90-0

5. Solenoids:
   General Electric Heavy Duty Solenoids Type CR 9503-209 C, Pull Type, 110 volt D. C. 1-inch stroke.

6. Bearings:
   Torrington Bearings Type RC Cat. No. FDT-14

7. Timer:
   Electro Pulse Model 272-A, ten channel timer with thyratron output circuits with 5 amp. capacity

8. Manifold System:
   Linde Oxweld High Pressure Gas Manifold, Type M 26 equipped with Linde Oxweld Pressure Regulators, Type R 89

9. Control Panel:
   a. Pressure gauges: Ashcroft Pressure Gauges Type 1057, Range 0 to 1000 psi, Alumalife Case
   b. Control Valves: Hoke Direct Mounting Panel Valves No. 379
   c. Tubing and Fittings: Copper Tubing and Fittings suitable for working pressures of up to 1000 psi.
APPENDIX B  WORKING DRAWINGS OF MACHINE PARTS

Since the machined parts or the loading units were essentially hand-made, there are some small discrepancies in the dimensions of supposedly identical pieces. For the most part, machines No. 1 and No. 2 were matched as closely as possible insofar as the major parts of the machine are concerned. If replacements of any part are required, it is suggested that the critical dimensions be measured rather than blindly using the dimensions shown on the working drawings. Some parts of the machines were altered after other parts serving the same function were made, thus the working drawings do not always correspond to the parts in a particular machine. The working drawings in all cases correspond to the last alteration made to any of the three machines and it is suggested that any replacement parts for any of the machines be made to correspond to the working drawings.
Parts List for 60-Kip Pulse Loading Machines

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**Trigger Mechanism Assembly**

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Detail "C"

I-"O" RING (1820-19)

Detail "D"

I-"O" RING (1820-75)

UNIVERSITY of ILLINOIS
60 KIP
PULSE LOADING MACHINE

Part Number
Page
2-MAIN CYLINDER

O-Ring detail A-3

1/16 Drill 16 Holes

UNIVERSITY of ILLINOIS
60 KIP PULSE LOADING MACHINE

Part Number
Page

2
3 - MAIN PISTON
1 Req'd
Aluminum

O-Ring detail A-2

O-Ring detail B-5

10.000 ±0.003 in.

2.995 ±0.005 in.

3/4

1/4
i
1/16 Drill
16 Holes

8 - ALIGNMENT PLATE
1 Req'd
A-7 Steel

9 - STOP PLATE
1 Req'd
A-7 Steel

UNIVERSITY of ILLINOIS
60 KIP
PULSE LOADING MACHINE

Part Number 8,9
O-Ring detail A-3

10-STORAGE CHAMBER
1 Req'd

UNIVERSITY of ILLINOIS
60 KIP
PULSE LOADING MACHINE
No. 7 Drill  3/4 Deep
1/4-20 Tap  1/2 Deep
20 Holes

I/2 Drill
2 Holes

1 1/2 Drill
5 1/4 D.

1 1/16 Drill
16 Holes

22.5°

II-HEAD

2 Req'd (See Note)
A-7 Steel

NOTE: 1 Req'd With Centerhole
1 Req'd Without Centerhole

UNIVERSITY of ILLINOIS
60 KIP
PULSE LOADING MACHINE
13-AUXILIARY CYLINDER
4 Req'd

9/32 Drill
13/32 C'bore
3/16 Deep
10 Holes

1/4 Drill 24 Holes
O-Ring detail A-1
O-Ring detail B-2

O-Ring detail B-2

13-AUXILIARY CYLINDER
4 Req'd

1/4 R.

UNIVERSITY of ILLINOIS
60 KIP
PULSE LOADING MACHINE
Part Number 13
14-AUXILIARY PISTON
4 Req'd
Aluminum

O-Ring detail B-4

O-Ring detail C

UNIVERSITY of ILLINOIS
60 KIP
PULSE LOADING MACHINE

Part Number 14 Page 18
15-VALVE ROD
4 Req'd
AISI-4130 Steel (hardened)

16-VALVE ROD TAPPET
4 Req'd
AISI-4130 Steel (hardened)

17-LOCK NUT
8 Req'd
Size: 1-14 Hex Nut

18-NUT
8 Req'd
Size: 1-14 Hex Nut

19-WASHER
16 Req'd
Size: 1 I.D., 1 1/2 O.D., 1/8 Thick

20-WASHER
4 Req'd
Size: 1 I.D., 2 O.D., 1/4 Thick

21-LOCK WASHER
4 Req'd
Size: 1 I.D.

UNIVERSITY of ILLINOIS
60 KIP
PULSE LOADING MACHINE
Part Number
15, 16, 17, 18, 19, 20, 21
The diagram illustrates a 23-SLIDE VALVE with the following specifications:

- 2 Req'd
- Aluminum

Key dimensions include:

- O-Ring detail B-6 and O-Ring detail D
- 1 1/16 Drill C'bore 1 1/2
- 1/8 Deep
- Back side
- 2 Holes

Match drill, tap, 
& c'bore for 3/8-16
Allen hd cap sc
14 Holes

The diagram shows a cross-sectional view of the valve with detailed measurements and notes for each feature.
TRIGGER FRAME AND LINK ASSEMBLY
Drill & c'bore for 3/8 soc hd cap sc 3 Holes
3/4 R.

Ream 3/4
3/4 R.

Drill & tap 6-32

Drill & tap 5/16-18
4 Holes

Drill & tap 5/16-18
4 Holes

Drill & c'bore
for 3/8 soc hd cap sc 3 Holes
3/4 R.

3/4 R.

29-LINK PIN
4 Req'd
A-7 Steel

30-LINK PIN
4 Req'd
A-7 Steel

31-WASHER
8 Req'd, 1/8 Thick

32-BEARING
4 Req'd (See materials & components specifications)

UNIVERSITY of ILLINOIS
60 KIP
PULSE LOADING MACHINE

Part Number
27,28,29,30,31,32
Drill & tap 3/8-16
4 Holes

NOTE: Locate, drill, & tap during assy.

Drill 5/16-18
tap 4 Holes

NOTE: Locate, drill, & tap during assy.

33—CONNECTING BAR
2 Req'd
Aluminum

Steel wear plate
4-40 Flat hd sc
4 per plate

UNIVERSITY of ILLINOIS
60 KIP
PULSE LOADING MACHINE
TRIGGER ADJUSTMENT ARM AND TRIGGER PISTON ASSEMBLY
35-PACKING NUT
4 Req'd, A-7 Steel

1 1/4-12 NF2

3 3/4

9/16 Ream

5/16 Drill
4 Holes

36-TRIGGER PISTON
4 Req'd
Aluminum

1/4-28

1/4-12 NF2

Top

5/8 Drill
4 Holes

1/8 Pipe thd

1 1/8 Dia

1/8 Chamfer

Tap 1/4-12 NF2

34-TRIGGER CYLINDER
4 Req'd
A-7 Steel

1/4-12 NF2

1/2 D.

1/2 D.

3/4

1/4

9/16 D.

7/16

1 9/16

2

2 3/4

3/8

3/4

1 3/4

3/4

1 1/2

1 1/2

3/4

1 7/8

1 7/8

1 7/8

3/16 Drill
4 Holes
Drill & c'bore for 5/16 soc hd cap sc
3 Holes

37-TRIGGER ADJUSTMENT ARM
Req'd:
2 - Right Hand
2 - Left Hand
A-7 Steel

Drill & c'bore for 5/16 soc hd cap sc

38-ADJUSTMENT ARM PIN
4 Req'd
A-7 Steel

39-TRIGGER BAR
10 Holes
2 Req'd
A-7 Steel

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PULSE LOADING MACHINE
Part Number
37, 38, 39
Page 9-2
Drill & tap 6-32

Drill & ream 3/8

Drill 7/32 c'bore 5/16

Drill 3/16

Drill 3/32 c'bore 5/16

Drill 3/32 for 3/32 pin

Drill & tap 5/16-18

3 Holes

NOTE: Corners rounded with 1/8 radius.
Drill & tap 10-32

Drill & tap 1/8

Drill & tap 6-32

Drill & tap 1/4

Drill & tap 1/4-20

Turn to 3/32
Round end as req.

43-SAFETY-PIN
4 Req'd
Size: 6-32 x 5/8

42-SAFETY-FORK
2 Req'd
A-7 Steel

41-SAFETY-BRACKET
2 Req'd
Aluminum

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Part Number
41,42,43
Drill for 5/16
Soc hd cap sc
C'core 1/8 deep
2 Holes

44-SEAR CATCH
2 Req'd
SAE-4140 Steel
Hardened To Approximately
Rockwell C-50

Center section to be finished during assembly

Drill 5/16
Drill & c' bore 1/4 for 5/16 soc hd cap sc 3 Holes

Drill & ream 1/4

46-SEAR, BRACKET
2 Req'd
A-7 Steel

Drill 1/4

45-SEAR
2 Req'd
SAE-4140 Steel
Hardened To Approximately
Rockwell C-50.
54-SAFTY-ROD
2 Req'd
AISI-4130 Steel

Drill & tap 10-32

5/16

3/32

5/32

1 3/8

9/16

1/2

2

7/16

10-32 Thd

55-SAFTY-TAPPET
2 Req'd
AISI-4130 Steel

5/32

3/16

3/32

21/32

56-SAFTY-SEAR FOLLOWER
2 Req'd
AISI-4130 Steel

5/16 D

1/2

5/32 D

3/16 D

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Part Number
54,55,56,57

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58-MACHINE MOUNT
2 Req'd
A-7 Steel

59-FRAME MOUNT
2 Req'd
A-7 Steel

60-FRICTION CLUTCH
2 Req'd
A-7 Steel

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PULSE LOADING MACHINE
Part Number 58,59,60
NOTE: "O" Ring, vulcanized from 66 5/8" length of 1/4" rubber.
NOTE: Finish I.D. After Welding

Drill & tap 3/4-10
8 Holes 7/8 Deep

O.D. of Main Cyl Flange:
33 7/8 ± 1/16

2 3/4

3/32 of Pressure Hdl ± 1/32

Bevel for welding

62-EXTERNAL CYLINDER HEAD

4 Req'd
A-7 Steel

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PULSE LOADING MACHINE

Part Number
62

Page 10
Drill 8 Holes

Section A-A

63-INNER SEALING RING
2 Req'd
A-7 Steel

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60 KIP
PULSE LOADING MACHINE

Part Number
63

Page
5
64-OUTER SEAL AND RETAINER RING
2 Req'd
A-7 Steel

13/16 Drill 8 Holes
Each plate

Drill & tap
1/2-13 16 Holes

65-INNER RETAINER RING
2 Req'd
A-7 Steel

Section A-A

Section B-B

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60 KIP PULSE LOADING MACHINE
Part Number 64,65