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NATIONAL DEFENSE RESEARCH COMMITTEE

of

OFFICE OF SCIENTIFIC RESEARCH AND DEVELOPMENT

WAR METALLURGY DIVISION

TECHNICAL REPORT NUMBER  
NS-109  
THE DESIGN OF A HYDRO-PNEUMATIC MACHINE FOR RAPID LOAD TENSILE TESTING

Progress Report

on

BEHAVIOR OF METALS UNDER DYNAMIC CONDITIONS (NS-109):  
THE DESIGN OF A HYDRO-PNEUMATIC MACHINE FOR RAPID LOAD TENSILE TESTING

by

D. A. ELMER, D. S. CLARK, AND D. H. HYERS  
CALIFORNIA INSTITUTE OF TECHNOLOGY

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February 19, 1945

To: Dr. James B. Conant, Chairman  
National Defense Research Committee of the  
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From: War Metallurgy Division (Div. 18), NDRC

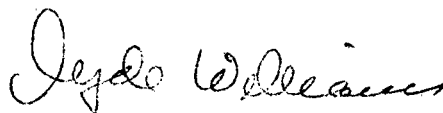
Subject: Progress Report on "Behavior of Metals Under Dynamic  
Conditions (NS-109): The Design of a Hydro-Pneumatic  
Machine for Rapid Load Tensile Testing".

The attached progress report submitted by D. S. Clark,  
Technical Representative on NDRC Research Project NRC-82, has been  
approved by representatives of the War Metallurgy Committee in  
charge of the work.

This report describes the equipment designed on this project  
for studying the influence of rapid loading and time at load on the  
start of plastic deformation of metals in tension.

Acceptance as a satisfactory progress report under Contract  
OEMsr-348 with the California Institute of Technology is recommended.

Respectfully submitted,



Clyde Williams, Chief  
War Metallurgy Division, NDRC

Enclosure

PREFACE

This report is pertinent to the problems designated by the Office of the Coordinator of Research and Development, Navy Department, as NS-109, and to the project designated by the War Metallurgy Committee as NDRC Research Project NRC-82.

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Division 18

NRC Project 82

OSRD Contract OEMsr-348

California Institute of Technology

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Report No. XLI

The Design of A Hydro-Pneumatic Machine

for

Rapid Load Tensile Testing

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23 December 1944

Submitted by:

D. A. Elmer, Ens., USNR

D. S. Clark

D. H. Hyers

Approved by:

D. S. Clark

Technical Representative

for Contract OEMsr-348

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The Design of a Hydro-Pneumatic Machine

For

Rapid Load Tensile Testing

ABSTRACT

This report presents the design of equipment for studying the influence of rapid loading and time at load on the initiation of plastic deformation of metals in tension. The machine consists of a cylinder and piston actuated by a hydro-pneumatic system which permits a load as high as 20000 lb to be applied to a specimen 0.500 in. in diameter in an interval of 0.005 sec or more. This load or a smaller load can be maintained on the specimen for any desired length of time greater than 0.001 sec. The principle features of the design and operation of the equipment are discussed. The application of the data obtained from tests with this equipment is considered. It is estimated that this equipment would cost around \$2500 and would provide very important data on the behavior of metals under dynamic loading which could be applied directly to structures.

Introduction

In studying the influence of strain rate it has been shown in a previous report <sup>1)</sup> that the proportional limit and ultimate strength of some metals increase with increasing strain rate. This effect may be represented in the manner shown in Fig. 1. It is believed that if a load is applied to a material rapidly enough, a stress higher than the static elastic limit can be

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- 1) The influence of pure strain rate on the tensile properties of three types of ship plate, by P. E. Duwez, H. E. Martens, D. A. Elmer, and D. S. Clark. OSRD Contract OELsr-348. Report No. XXXIX.

sustained by the material for a certain length of time without plastic deformation. The higher the stress, the quicker will plastic deformation occur after reaching this stress. This condition may be represented as shown in Fig. 2. The validity of this belief has been partially substantiated by preliminary tests<sup>2)</sup>. However, the equipment used in making the preliminary tests involved a solenoid to obtain the loading of the specimen, and the time required to reach the desired load was between 0.23 and 1.60 sec. Even under this condition of a relatively slow rate of loading, elastic behavior was maintained for an appreciable length of time at a stress above the static elastic limit. The results of tests might be represented in Fig. 2 by the region to the right of a time of about 0.5 sec.

From the preliminary tests it is apparent that the principles of rapid load testing and the application of results to practical structural problems are sound. However, it is clear that equipment is required which will operate more rapidly in order to investigate the left hand region of Fig. 2.

The purpose of this report is to present a design of a machine for making tensile tests on specimens with which the time to reach load can be as short as about 0.005 sec, thus permitting an extensive investigation of the effect of rapid load and time at load on the tensile properties of materials.

#### General Description of Equipment

The general character of the machine that has been designed is shown in Fig. 3. It is to have a capacity of 20000 lb which can be applied to a

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2) A preliminary study of the influence of rapid loading and time at load on the initiation of plastic deformation in tension, by P. I. Duwez, H. E. Martens, and D. S. Clark. OSRD Contract OEMsr-348. Report No. XXXVIII.

specimen within any desired time between 0.005 and 0.50 sec. Any load less than the maximum can be applied within about the same time limit and can be sustained for any desired length of time greater than 0.001 sec.

Essentially the machine is a 12 in. diameter cylinder closed at both ends, with a rigid framework built over the cylinder. A test specimen is mounted with one end attached to the rigid framework and the other end to a piston rod extending through one end of the cylinder. An aluminum alloy piston, attached to the piston rod, separates the cylinder into two chambers.

The chamber above the piston is filled with air and the chamber below the piston with water. Pressure is maintained at any desired value up to 200 lb/in.<sup>2</sup> on both sides of the piston by means of compressed air, as shown in Fig. 4. The test is initiated by closing the water line and rapidly exhausting the lower chamber to atmospheric pressure. This is accomplished by means of a quick acting valve or a bursting diaphragm. The rate of pressure drop in the lower chamber is controlled by means of a variable orifice in the exhaust line. The test is ended by rapidly closing the exhaust valve and opening the water line allowing the pressure to build up on the water side of the piston. For tests in which the time of loading is greater than about 0.030 sec, air instead of water may be used in the lower chamber in order to permit better control of the time of loading. The record of force-time and strain time will be recorded on two oscillographs.

#### Details of Design

An assembly drawing of the rapid loading machine is shown in Fig. 5. The principle part and the function of each are discussed below. The part numbers refer to the numbers on the assembly drawing (Fig. 5).

The base of the machine (-1) is made of 1-1/4 in. mild steel plate and rests on four feet made of the same material. The cylinder (-2) is made of 12 in. pipe with welded flanges. The cylinder rests in a groove in the base plate and is bolted in place with twelve bolts 7/8 in. in diameter. The upper end of the cylinder is closed with a 1-1/4 in. mild steel plate (-3) bolted to the cylinder flange, which also provides a base for the columns. The top and bottom edges of the cylinder fit into a groove in the top and bottom plates. The inside of the groove in the end plates and the edges of the cylinder wall have concentric "V" thread-like serrations 1/64 in. high which bite into a soft aluminum gasket 1/16 in. thick providing a pressure seal. The ends of the cylinder fit accurately and deep enough into the grooves to provide positive alignment. The inside surfaces of the cylinder and end plates are chromium plated to prevent corrosion.

A cast aluminum alloy piston (-5) is attached to a heat treated steel piston rod (-4). The piston assembly is guided by bronze bushings in the cylinder end plates. A pressure seal is provided in the bushings by neoprene "O" rings. Neoprene "O" rings are also used to provide a pressure seal on the piston rod inside the piston and at the outer edge of the piston. A cap in the base plate below the bushing holds a rubber disk which provides a stop for the piston rod at its lowest point of travel. The lower part of the piston rod is hollow for lightness. The piston is held in position on the rod by a steel washer (-6) and two nuts (-7).

The dome (-11) is rigidly supported by two columns (-8) which thread into bosses welded to the top plate. The columns are made of cold rolled steel. The dome structure bolted to the columns is a beam built up of welded 1/2 in. mild steel plate. The columns and dome are made very heavy so that the total deflection when loaded to 20000 lb will be less than 0.005 in.

One end of the test specimen (Fig. 6) is screwed into the dynamometer (-12) fastened to the dome and the other end is screwed into a fitting (-9, -10) on the end of the piston rod. The dynamometer consists of a hollow cylindrical section of heat treated steel onto which an electric strain gage is cemented. A sleeve (-13) protects the strain gage winding. The fittings at the upper and lower ends of the specimen are provided with spherical seats to allow for any slight misalignment of the system.

Auxiliary equipment, shown in Fig. 7, will be required to operate the machine described. These items, listed below, are all commercially available.

1. Compressor - Ingersoll-Rand No. 232C1 rated  $4.5 \text{ ft}^3/\text{min}$  at 200  $\text{lb}/\text{in.}^2$ . Unit consists of compressor, receiver, motor, and stop-start control.
2. Pressure tank - Tank 12 in. in diameter and 24 in. high for 200  $\text{lb}/\text{in.}^2$  air and water storage, with a high pressure gage glass or indicating float valve to indicate the height of the water level in the tank, and an accurate bourdon gage to indicate the air pressure.
3. Water reservoir - Tank 5 in. in diameter and 24 in. high, open at top and with a gage glass.
4. Quick acting valve - Progressive Welder Co., "Quick as a Wink" valve, normally closed type, 230 volt A.C. solenoid operated. Approximate time of operation 0.010 sec. This valve closes the water line and exhausts the cylinder in one motion of a sliding bushing.
5. Valves and piping as indicated - Piping will be  $1/2$  in. copper tubing, valves will be standard 200  $\text{lb}/\text{in.}^2$  globe valves.

The rate of loading will be controlled by the use of an orifice of the desired size in the line between the lower chamber and the exhaust valve. While not designed, it is proposed to use a fitting by which different sized orifices can be inserted with facility.

The duration of the test and the operation of the solenoid valve will be controlled by means of an electronic circuit (Fig. 8). This circuit has a 220 volt D. C. power supply with capacitor boosters to actuate the solenoid valve. A capacitor discharge circuit is used to regulate the duration of the test, i.e. the time the valve is held open. This time can be varied from 0.001 to 100 seconds. The solenoid valve can also be operated manually by means of a push button.

Two complete double channel recording units are available in this laboratory. One is a recording cathode-ray oscillograph unit with a range of single sweep time from 0.0005 to 0.050 sec. The other is a brush direct inking oscillograph. The available paper speeds on the latter are 5, 25, and 125 mm/sec which, with a 60 cycle/sec carrier wave, make it possible to record tests in which the time involved is greater than 0.10 sec.

All of the controls for the equipment can be centered in one panel, immediately adjacent to the machine, such as shown in Fig. 9, by careful arrangement of tanks and valves and by using universal joint extensions on the valve stems. With this arrangement the operator can control all of the operations of the equipment while standing in one place.

#### Operation of Equipment

A complete layout of the equipment is shown in Fig. 7 with all valves numbered. The following discussion describes the procedure of making a rapid load tensile test with this equipment.

-7-

An orifice of the proper size as dictated by the rate of loading to be used is inserted in the exhaust line. The compressor receiver is pumped up to a pressure of 200 lb/in.<sup>2</sup> which is maintained for the test. The water reservoir is filled to the desired level. Initially all valves are closed. Valves 8 and 9 are opened until the pressure tank is filled with water from the reservoir to the desired height and then closed. The lower chamber of the machine is completely filled with water by opening valves 6 and 2. Valve 2 is held open until the hydrostatic pressure in the tank fills the lower chamber and all air is removed and then it is closed. Valve 5 is opened and then the pressure on the system is raised to the desired value by means of valve 7.

The system is now under pressure and at equilibrium. To raise the piston, valve 5 is closed and a small amount of air is bled through valve 3 until the proper position is attained. To lower the piston, valve 6 is closed with valve 5 open. A small amount of water is bled through valve 2. These manipulations facilitate the installation of the specimen.

With the specimen in position and all valves except 5 and 6 closed, the pressure in the tank is adjusted to the value required to give the desired force to the specimen. The strain gage and dynamometer leads are attached to the recording lines and the control unit is set to give the desired duration of the test. The test is started by means of a push button which energizes the solenoid valve circuit opening the quick acting valve 1, exhausting the lower chamber, and closing the water line from the pressure tank, thus loading the specimen. After the proper length of time has elapsed, the time control unit causes valve 1 to shift and to admit water under pressure to the lower chamber thus unloading the specimen.

The records obtained under such a procedure when plastic deformation occurs may be of the type shown in Fig. 10. The time at which plastic strain

occurs after loading is determined from the curve shown in Fig. 10b. A series of specimens would be tested at different loads for a given rate of loading. From this data a curve such as shown in Fig. 2 may be obtained.

### Application of Results

The application of the results obtained with the equipment described in this report would seem to be of considerable practical value. As pointed out in a previous report, (3) the most common dynamic loading to which a machine part or structure is subjected, aside from fatigue, involves rapid loading and not impact. If impact is involved there are usually conditions present that reduce the shock effect and leave primarily a rapid loading condition on the structure. There is an increasing tendency to analyze the force-time condition existing in structural members in which dynamic loading is involved by the use of electric strain gages.

Suppose, for example, a structural member is loaded in the manner indicated in Fig. 11. The maximum force  $f_1$  is attained in a certain time and maintained for a time  $t_1$ . By referring to a curve such as Fig. 2 for the material, it is possible to select that stress which will not induce plastic deformation in time  $t_1$ , and thence the dimension of the member. It would also be necessary to determine if the force  $f_2$  in Fig. 11 acting for the time  $t_2$  would produce plastic deformation. With data of this type available for different materials, the one most suitable for the condition of loading could be selected.

In discussing the possible application of the results of the proposed tests, it is recognized that very complex stress conditions are involved in many structures and that the illustration employed is very simple. However, the

3) Reference 2



importance of the information to even the simple case should be recognized.

### Summary

The design of this equipment for the study of the influence of rapid loading and time at load on the initiation of plastic deformation in tension was stimulated by the encouraging results of preliminary tests made with very simple and inadequate equipment <sup>4)</sup>. A design which is believed to be workable has been prepared in detail. While this report presents the general plan of the equipment, detail working drawings have been made of all parts of the machine itself. The various parts have been carefully engineered. The problem of attaining such rapid loading has been carefully studied for this particular machine. The analysis is presented in the appendix. Some additional engineering is required since the detailed layout of piping, valves, etc., has not been made.

It is estimated that the testing machine alone would cost less than \$2000 to construct. The auxiliary equipment, exclusive of recording facilities, may cost of the order of \$1500 including installation. Hence, it may be estimated that the total cost would be around \$3500.

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4) Reference 2

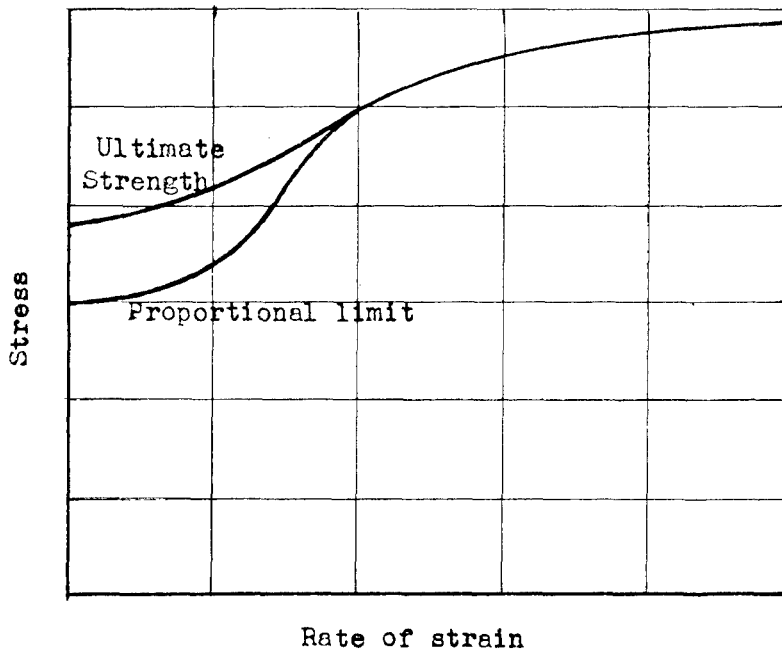


Fig. 1 Approximate relation of ultimate strength and proportional limit to rate of strain.

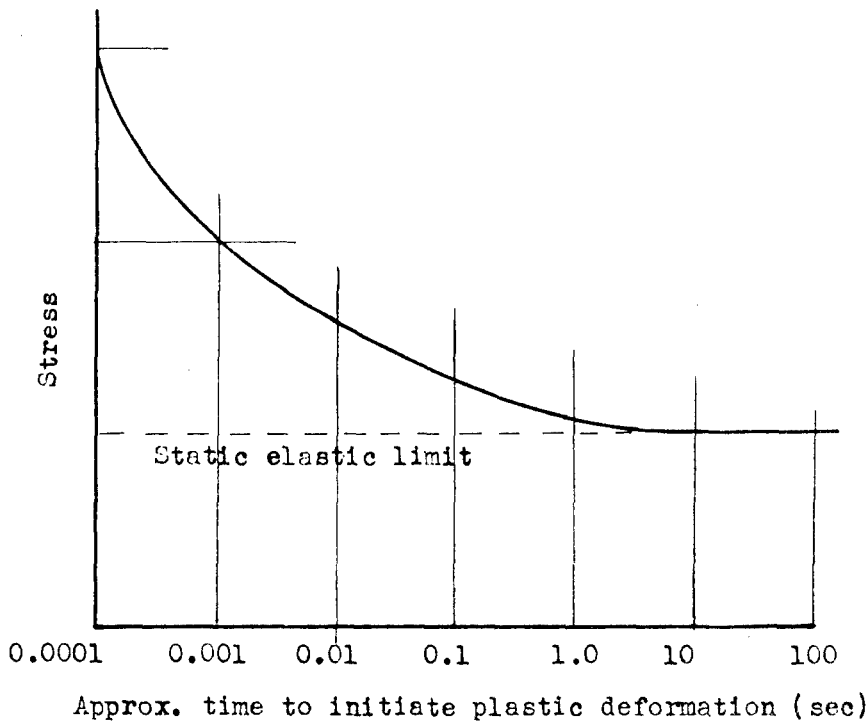


Fig. 2 Approximate relation of elastic limit to time to initiate plastic deformation.

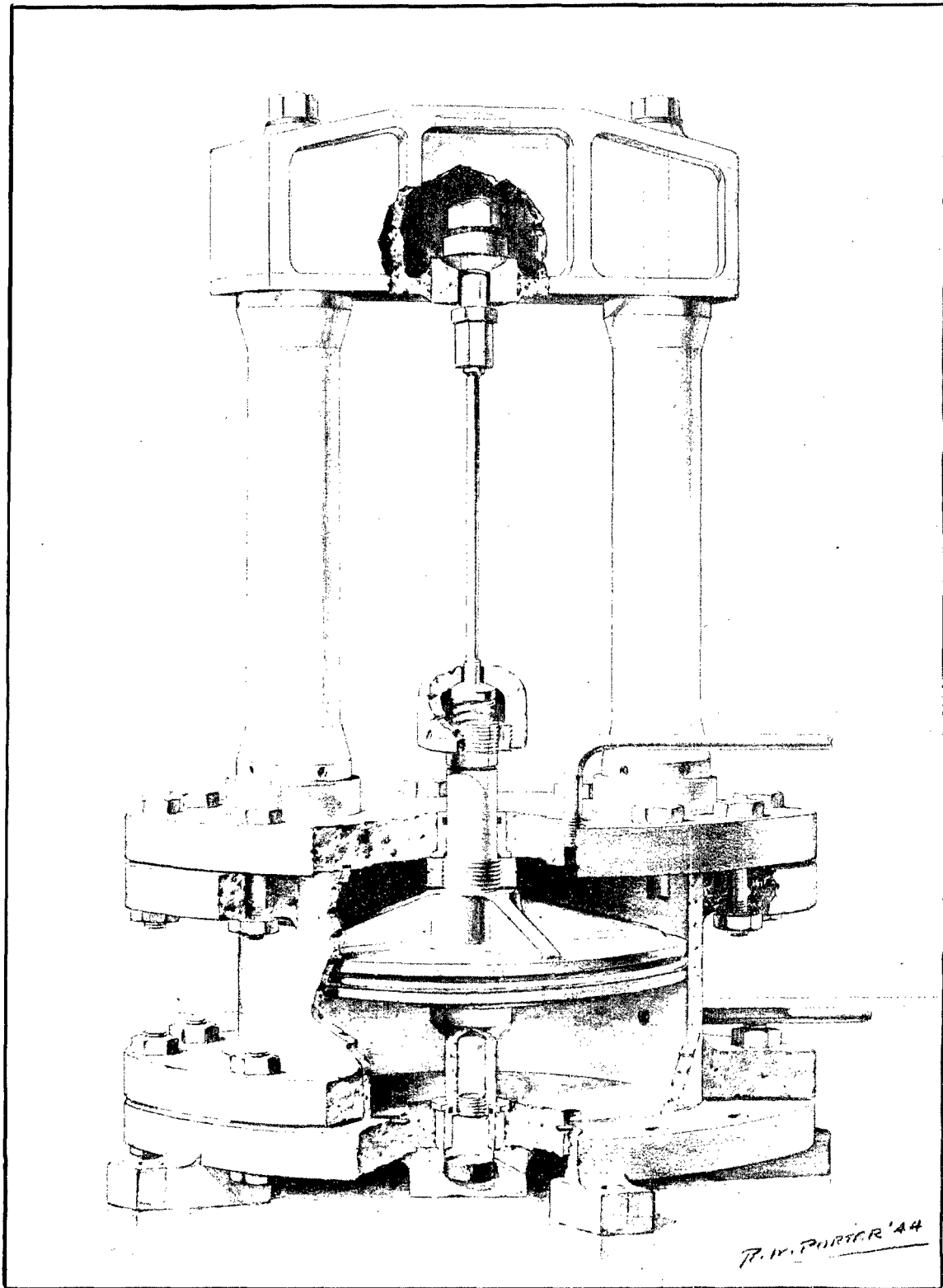


Fig. 3 Rapid load tensile testing machine.

-12-

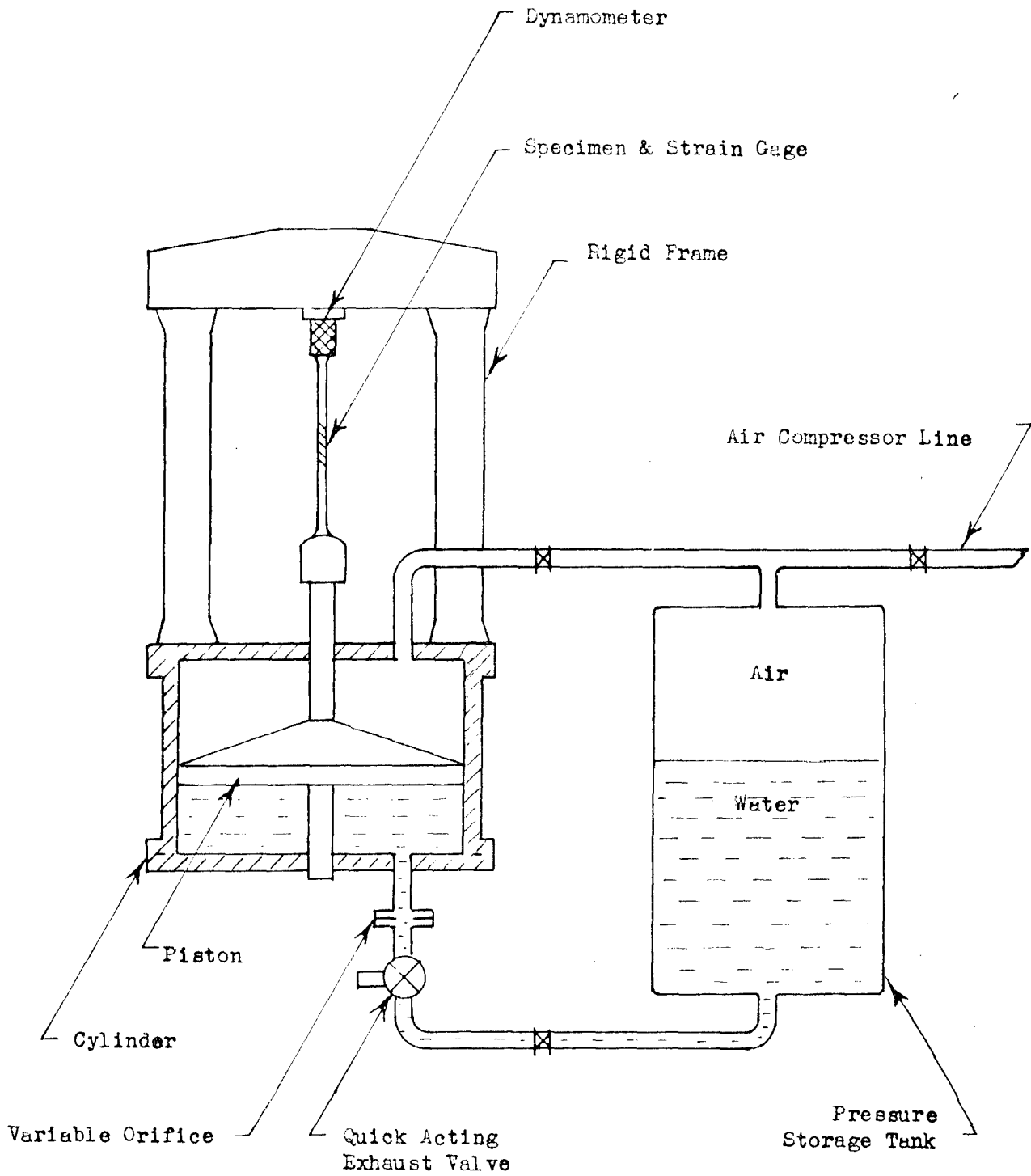
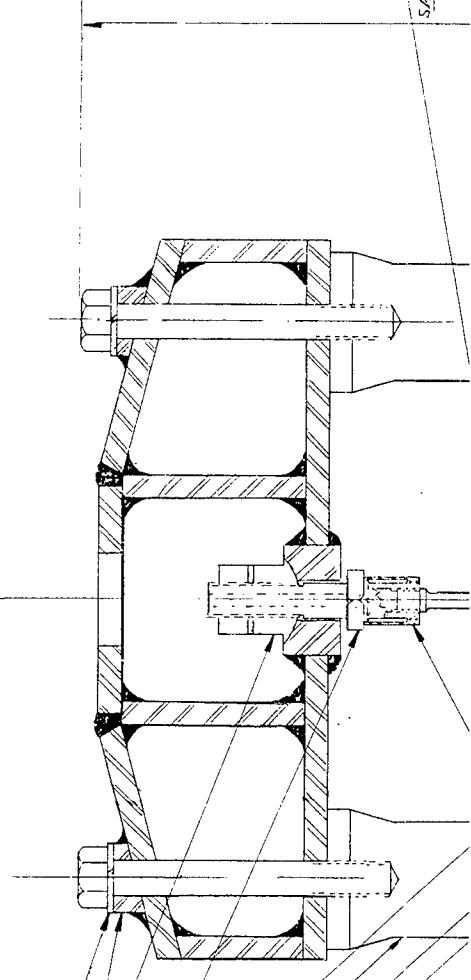
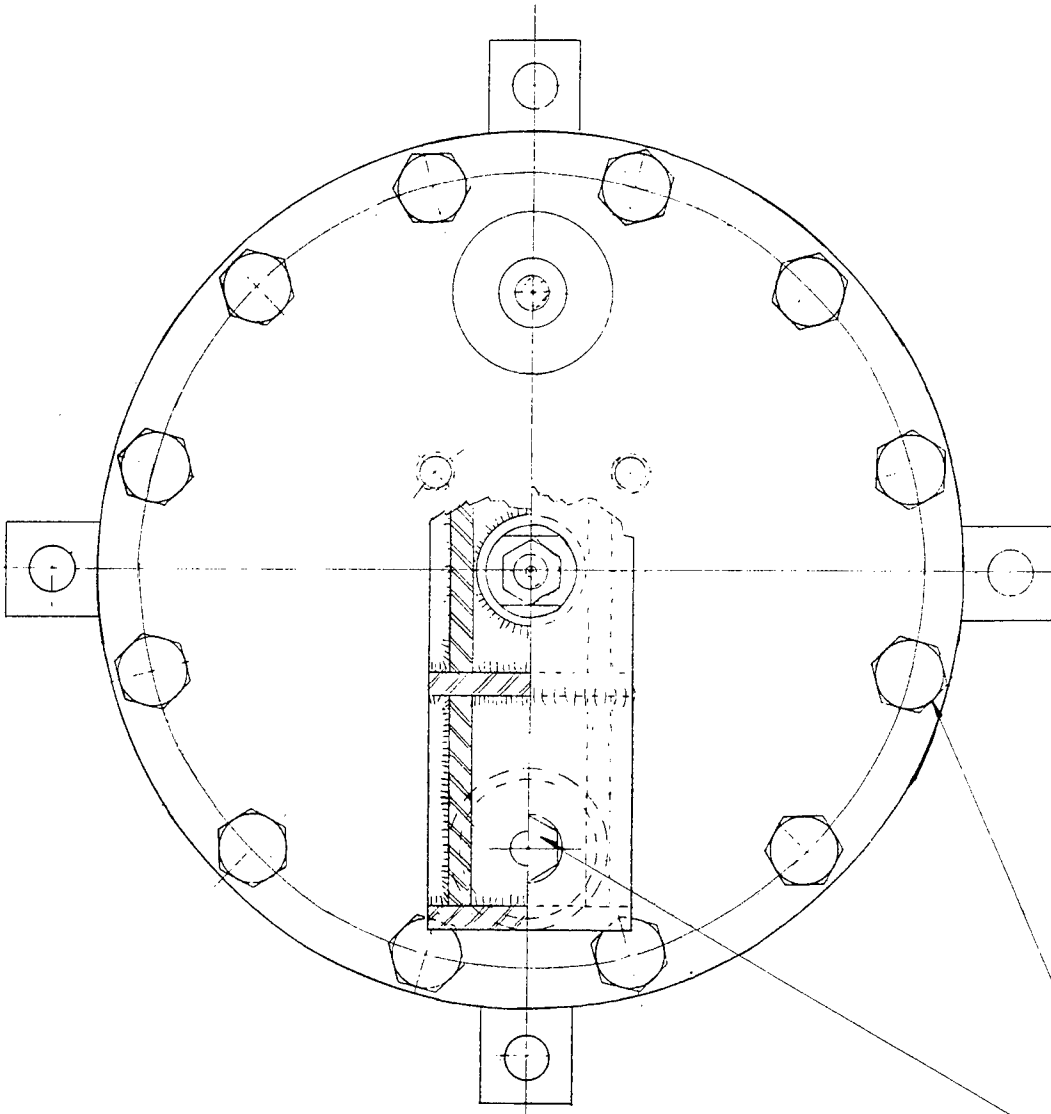


Fig. 4 Basic equipment for rapid load tensile tests.



PART	DESCRIPTION	REQ
RL-2-3-978	ASSEMBLY - MTD. LDR. TENSILE TESTING MACHINE	1
-1	PLATE ASSEMBLY - BOTTOM	1
-2	CHAMBER ASSEMBLY	1
-3	PLATE ASSEMBLY - TOP	1
-4	SHAFT	1
-5	PISTON	1
-6	WASHER	1
-7	NUT - PISTON	2
-8	COLUMN	2
-9	ADAPTER ASSEMBLY	1
-10	NUT - LOWER SPECIMEN	1
-11	DDME ASSEMBLY	1
-12	DYNAMOMETER	1
-13	SHIELD - DYNAMOMETER	1
-14	NUT - UPPER SPECIMEN	1
-15	WRENCH - SPANNER	1
-16	WRENCH - NUT SOCKET	1
-17	BUMPER - RUBBER SHORE 50 1/2" DIA. x 1/2" THK.	1
-18	BOLT - 3/8" DIA. x 3" STL HEX HD MACH. 24	24
-19	NUT - 3/8" DIA. x 3" STL HEX	24
-20	WASHER - 3/8" STL LOCK	24
-21	O'RING - AN 6227-25 APPROX. 1/2" DIA. x 1/8" THK. RUBBER CO.	2
-22	O'RING - AN 6227-27 APPROX. 3/4" DIA. x 1/8" THK. RUBBER CO.	1
-23	O'RING - AN 6227-29 APPROX. 1" DIA. x 1/8" THK. RUBBER CO.	1
-24	WASHER - STL 2 1/2" DIA. x 1/4" THK.	1
-25	GASKET - ALUM. 2.50 - 12.975 DIA. x 12.000 ID. x 1/8 THK FT70 GROOVE	2
-26	SCREW - F-20NC-3 STL SET x 1/2" LONG	1
-27	WASHER - 3/8" STL LOCK	3
-28	BOLT - 3/8" DIA. x 1 1/2" STL HEX HD MACH. 24	2
-29	SPRING - 1/16" x 1 1/4" OD. x 1 1/2" BIRTH. WIRE GA. 3/4 FREE LENGTH x 15 LB (APPROX) 1/4 DEFLECTION - ENDS FLAT	1
-30	NIPPLE - 1/8" NOM. PIPE SIZE NATIONAL GALV. PIPE x 5" LONG	1

- 12 BOLT - 1/2" REQ.
- 12 BOLT - 24 REQ.
- 11 WASHER - 9 REQ.
- 11 DDME ASSEMBLY
- 14 NUT - UPPER SPECIMEN
- 12 DYNAMOMETER
- 13 SHIELD - DYNAMOMETER
- 8 COLUMN - 2 REQ.
- 9 ADAPTER ASSEMBLY
- 10 NUT - LOWER SPECIMEN
- 10 SCREW - SET
- 4 SHAFT
- 5 O'RING - 2 REQ.
- 3 PLATE ASSEMBLY - TOP
- 9 GASKET - 2 REQ.
- 2 CHAMBER ASSEMBLY

SP 2-3-968  
SPECIMEN (REF.)

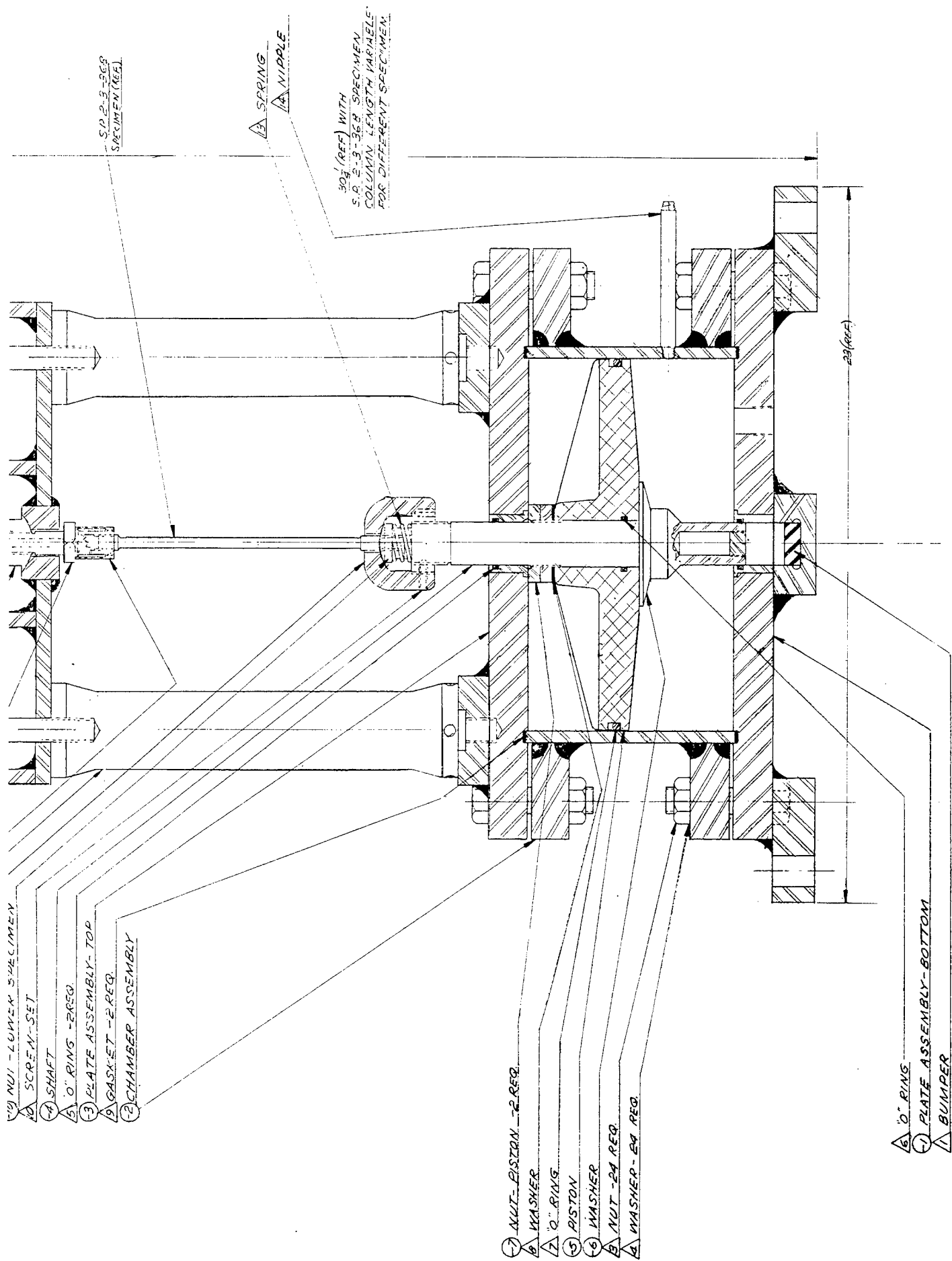


Fig. 5 Assembly, rapid load tensile testing machine.  
Half scale.

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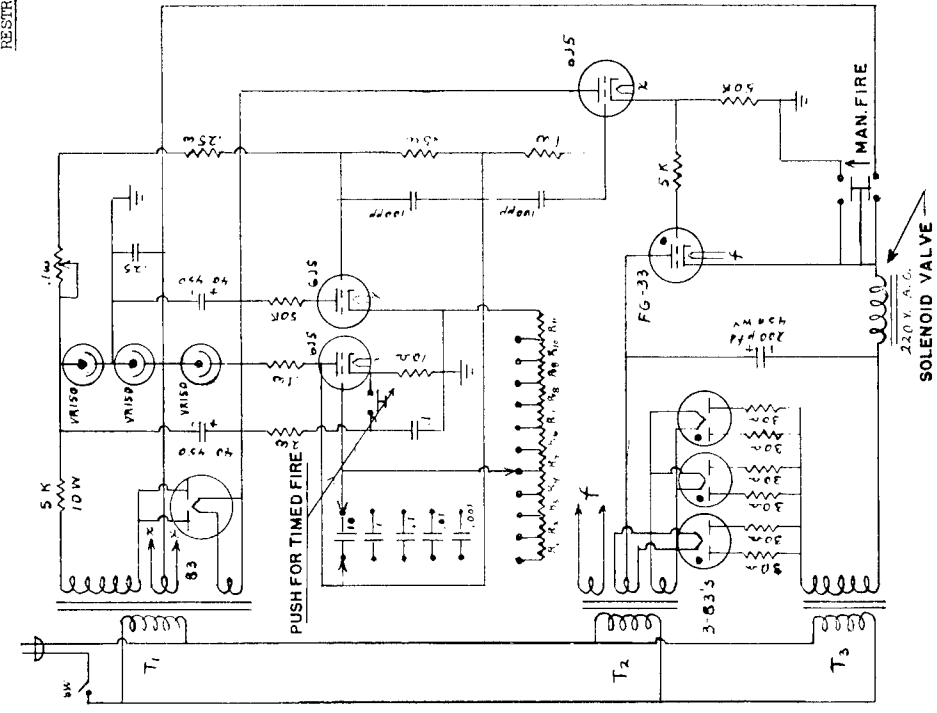


Fig. 8 Diagram of electronic control unit

Notes: Transformers, all 110v pri.; T1, 750v C.T.- 40a, 6v-2a.  
 T2, 5v-3a, 5v-3a, 5v-6a.  
 T3, 220v isolated, 150 W.  
 Resistors R1, R2, R3, etc. to be determined experimentally.

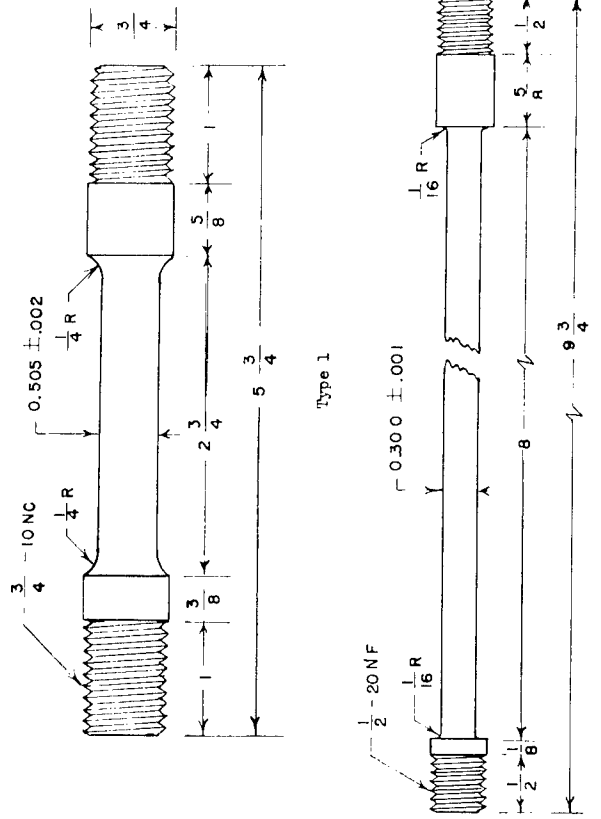


Fig. 6 Test specimens.

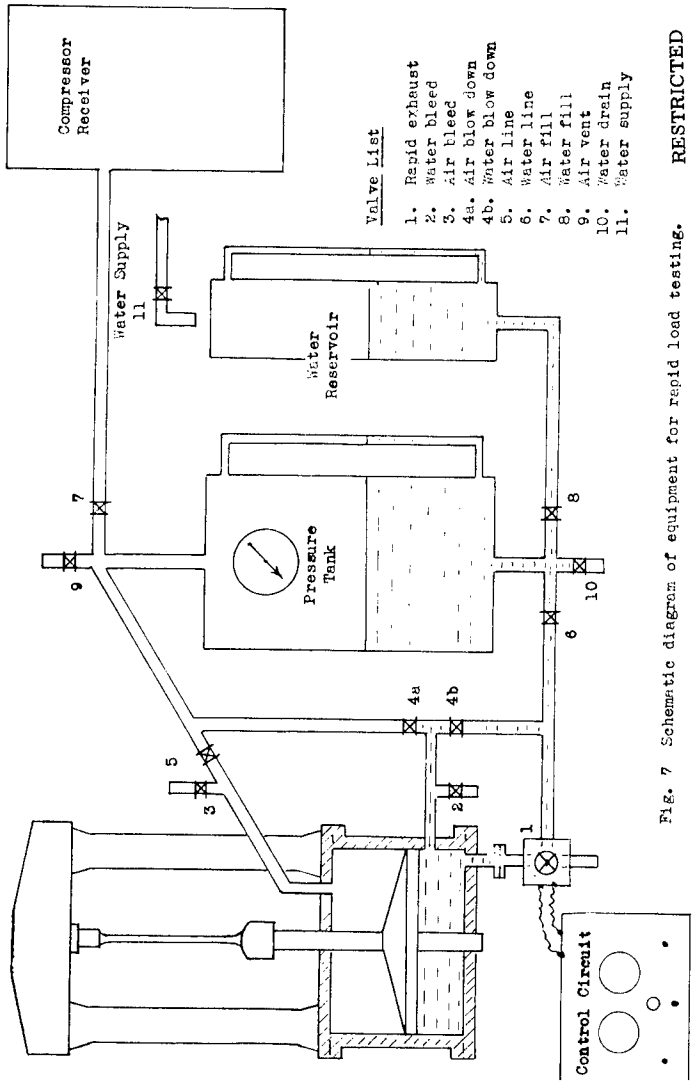


Fig. 7 Schematic diagram of equipment for rapid load testing.

- Valve List
1. Rapid exhaust
  2. Water bleed
  3. Air bleed
  - 4a. Air blow down
  - 4b. Water blow down
  5. Air line
  6. Water line
  7. Air fill
  8. Water fill
  9. Air vent
  10. Water drain
  11. Water supply

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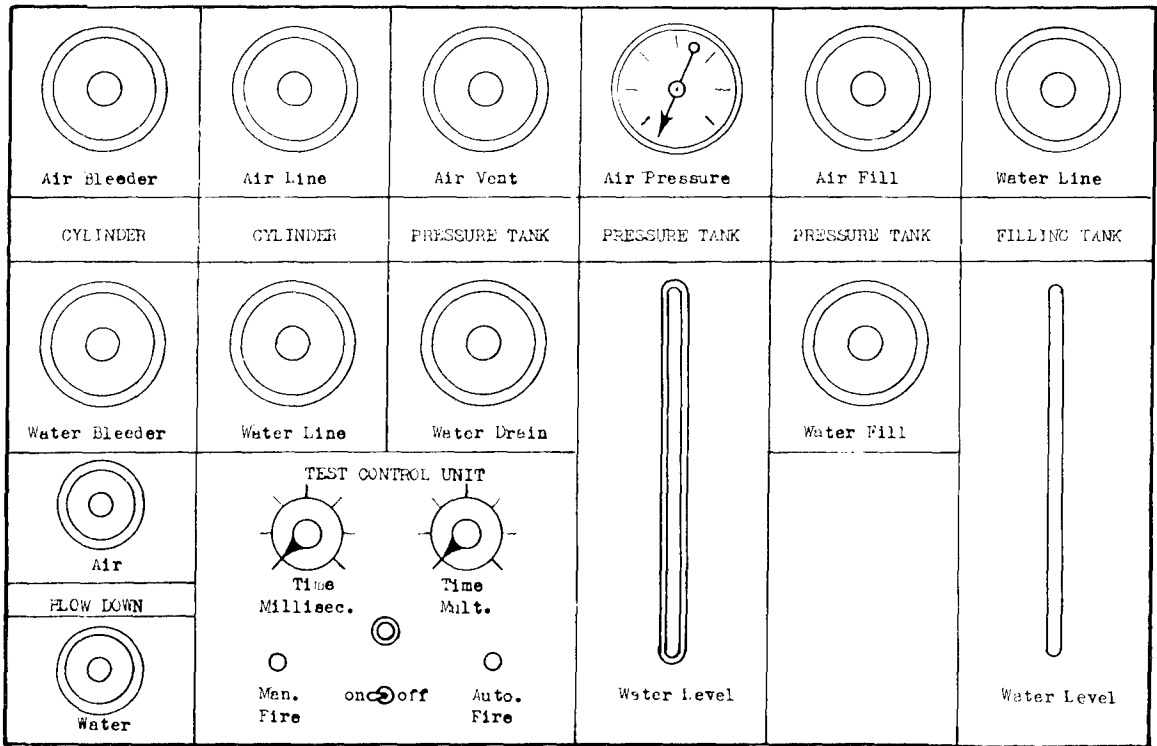
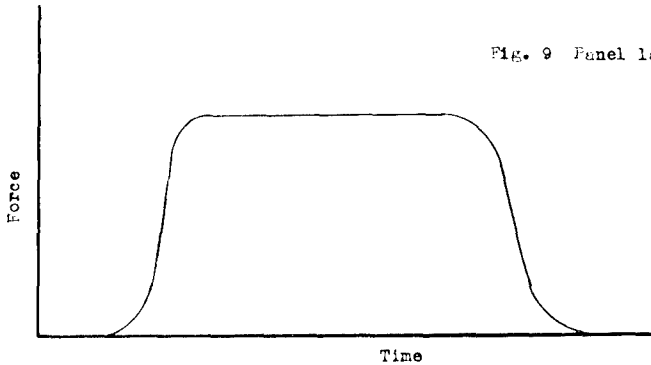
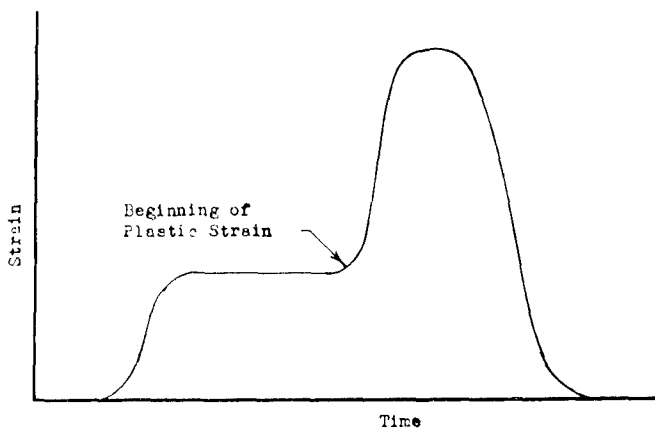


Fig. 9 Panel layout for rapid load testing.



(a) Typical force-time diagram.



(b) Typical strain-time diagram.

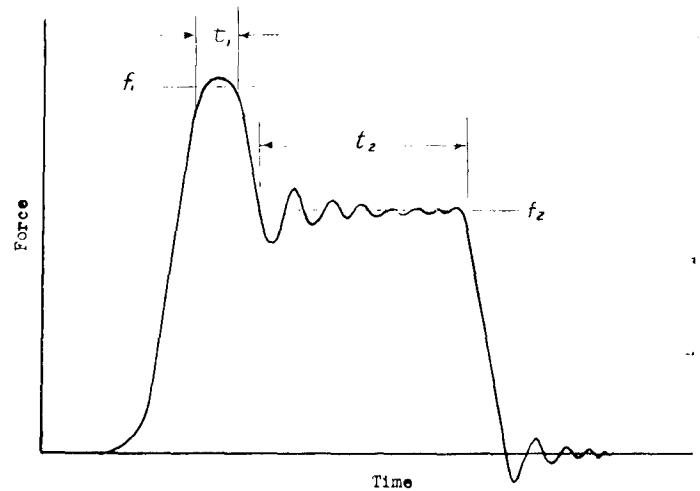


Fig. 11 Typical force-time diagram for a structural member.

Fig. 10 Typical force-time and strain-time diagrams.



## APPENDIX

A Mathematical Analysis of the  
Hydro-pneumatic Testing Machine

by D. H. Myers

The Analysis

A mathematical analysis of the rapid loading testing machine has been made as a guide in the design of the equipment described in the body of this report and, in particular, to obtain an estimate of the times required to load the specimen for different sized orifices. The following simplifying assumptions were made. The water was considered incompressible and fluid friction was neglected. The orifice was assumed to be opened instantaneously at the time  $t = 0$ . The tensile specimen was assumed to remain elastic. Finally, the horizontal components of the velocity of the water were neglected in comparison to the vertical component. This last assumption is certainly correct when the water chamber tapers slowly towards the orifice, as shown in Fig. A-1. When the chamber is cylindrical as in the design proposed in the body of this report, the time required to load the specimen may actually be somewhat greater than indicated by the results of this analysis.

The test specimen, of length  $l_s$  and cross-sectional area  $A_s$ , is shown at the top of Fig. A-1. Denote the (constant) air pressure on the top of the piston by  $P_0$  and the variable water pressure on the bottom face of the piston by  $P$ . Since the (downward) force on the piston is

$A_p(P_0 - P)$ , where  $A_p$  is the area of the piston, the equation of motion of the piston is:

$$m \frac{d^2x}{dt^2} = A_p (P_0 - P) - kx, \quad (1)$$

where  $m$  = mass of piston and connecting rod,  $x$  = displacement of piston, and  $k = A_s E / l_s$  is the spring constant of the specimen. The unknown pressure  $P$  can be found in terms of the velocity  $dx/dt$  of the piston by investigating the flow of water out of the orifice at B in Fig. A-1.

Since the horizontal components of the flow are neglected, and the force of gravity may also be neglected in comparison to the large forces involved, the equation of motion for the water is

$$-\frac{1}{\rho} \frac{\partial p}{\partial z} = W \cdot \frac{\partial W}{\partial z} + \frac{\partial W}{\partial t} \quad (2)$$

where  $\rho$  is the density of the water,  $p$  is the pressure at time  $t$  and at distance  $z$  from the initial position of the piston, and  $W$  is the velocity of the water at the same time and place.

The condition of continuity is

$$A_p \frac{dx}{dt} = A(z) \cdot W = A_0 \cdot W_0, \quad (3)$$

where  $A(z)$  is the cross-sectional area of the chamber  $z$  in. below the initial position of the piston,  $A_0$  is the area of the orifice, and  $W_0$  is the velocity of the water as it flows out of the orifice.

The boundary conditions, besides Eq. (1), are as follows:

at the orifice,  $z = h$ ,  $W = W_0$  and  $p = p_a$ , where  $p_a$  = atmospheric pressure; when  $t = 0$ ,  $p = P_0$  and  $W = 0$ .

By using the boundary conditions together with Eqs. (2) and (3), the unknown pressure  $P$  can be eliminated from Eq. (1). The result of this elimination is:

$$M \frac{d^2 x}{dt^2} = F_0 - \rho A_p \alpha^2 \left( \frac{dx}{dt} \right)^2 - kx, \quad (4)$$

where  $F_0 = A_p (P_0 - P_a) =$  final force on piston,

$$\alpha^2 = \frac{1}{2} \left( \frac{A_0^2}{A^2} - 1 \right),$$

and where  $M = \rho_0 + \rho A_p b$ . Here  $b$  denotes a constant

$$b = A_p \int_0^h \frac{dV}{A(z)},$$

which depends on the shape of the chamber.

The solution of Eq. (4), subject to the initial conditions that  $x = dx/dt = 0$  for  $t = 0$  can be expressed in the form

$$\tau = \int_0^s \frac{ds}{\sqrt{1 - e^{-2s} - s}}, \quad (5)$$

where

$$\tau = t \sqrt{\frac{2k}{M}},$$

$$s = \left( \frac{2\rho d^2 k A_p}{2\rho d^2 A_p F_0 + kM} \right) x,$$

$$B = \frac{2\rho d^2 A_p F_0}{kM} + 1,$$

are dimensionless parameters.

The general aspect of the relationship, Eq. (5), between the dimensionless parameters  $s$  and  $\tau$  during the loading period is shown in Fig. A-2. The slope is zero to start with, and again at  $\tau = \tau_c$ , when the value of  $s$  reaches a maximum. The stress in the specimen at the time  $t_0$  corresponding to  $\tau = \tau_c$  will be somewhat greater than the "steady state" stress  $F_0/A_s$ , due to the vibrational phenomenon of "overshooting". For

small orifices ( $\beta^2$  and  $\beta$  large) the amount of overshooting will be small, due to the large damping factor in Eq. (4). As the diameter of the orifice increases, so will the amount of overshooting, until it reaches 100% when the orifice diameter is equal to the piston diameter, and the motion becomes undamped simple harmonic.

The time  $t_0$  for the specimen to become fully loaded is given by

$$t_0 = T_0 \sqrt{\frac{BM}{2k}} \quad (6)$$

where

$$T_0 = \int_0^{s_0} \frac{ds}{1 - e^{-\beta s} - s} \quad (7)$$

and  $s_0$  is the positive root of the equation

$$1 - e^{-\beta s_0} - s_0 = 0 \quad (8)$$

Calculation of the Time of Loading

When the parameter  $\beta$  is large ( $\beta \geq 14$ ), corresponding to rather small orifices, the value of  $s_0$  in Eq. (8) is practically one, and the integral of Eq. (7) may be approximated to within about 1% by the formula  $T_0 = 2 + \frac{\ln 4}{\beta}$ , while the corresponding value of  $t_0$  is given by

$$t_0 = \sqrt{\frac{BM}{2k}} \left[ 2 + \frac{\ln 4}{\beta} \right] \quad (9)$$

On the other hand, when  $1 < \beta < 14$  the calculation of  $t_0$  is more difficult, and involves numerical integration. To facilitate this computation for this range a graph of the relationship between  $T_0$  and  $\beta$

was computed on the basis of Eq. (7). This graph is shown in Fig. A-3.

It is evident from Eq. (9) and the definition of  $\beta$  (following Eq. (5)) that for small orifices,  $t_0$  is nearly inversely proportional to the area  $A_0$  of the orifice.

The limiting value of  $T_0$  as  $\beta \rightarrow 1$  may be obtained by putting  $\alpha = 0$  in Eq. (4). This corresponds to taking the area of the orifice equal to the area of the piston. When  $\alpha = 0$ , Eq. (4) reduces to the equation of undamped simple harmonic motion, and the loading time  $t_0$  is equal to a half-period, so that

$$t_0 = \pi \sqrt{\frac{m}{k}} \quad (10)$$

This value of  $t_0$  will be called the "theoretical lower limit of  $t_0$ ".

The corresponding value of  $\beta_0$ , obtained from Eq. (6) by putting

$$\beta_0 = \frac{A_0}{A_p} = \frac{m}{k} \frac{g}{h} = 1 \quad \beta_0 = 11.72 = 4.45$$

Results of the Calculations

In calculating the loading curves and the time to get to load for the design proposed in the body of this report, the dimensions of the machine were taken in accordance with Fig. 5. The total weight  $mg$  of the piston and connecting rod was estimated at 20 lb. The "shape" constant  $b$  [defined after Eq. (4)] was taken equal to  $h = 3$  in. The most complete computations were carried out for steel specimens (Fig. 6) of length  $l_s = 2$  in. and cross-sectional area  $A_s = 0.2 \text{ in.}^2$ , where the steady state force  $F_0$  on the piston was taken large enough to produce a stress of 60000 lb/in.<sup>2</sup> in the specimen.

The computed loading curve for this case, and for the maximum designed orifice diameter of 0.75 in., is shown in Fig. A-4. Notice that the "overshoot" is 1800 lb/in.<sup>2</sup> in this case, since the maximum stress is 61800 lb/in.<sup>2</sup> instead of 60000 lb/in.<sup>2</sup>. This indicates that there will be an oscillation of small amplitude in the stress after the time  $t = t_0$ .

A curve showing the computed relationship between the loading time  $t_0$  and the orifice diameter is given in Fig. A-5, again for the 2 in. specimen, at a steady state stress of 60000 lb/in.<sup>2</sup>. For small orifices (less than 0.5 in. in diameter) the value of  $t_0$  is nearly inversely proportional to the area of the orifice. The "theoretical lower limit" of  $t_0$  in this case, obtained by allowing the area of the orifice to approach the area of the piston, is 0.524 millisecc (see Eq. (10)).

The effect of raising the steady state stress or of decreasing the spring constant  $k$  is to increase the time  $t_0$  for a given orifice diameter. For example, if the steady state stress is increased from 60000 to 100000 lb/in.<sup>2</sup> the value of  $t_0$  corresponding to the orifice diameter of 0.75 in. is raised from 1.4 millisecc to 1.8 millisecc. In the case of the 8 in. specimen (Fig. 6), taking a steady state stress of 150000 lb gives  $t_0 = 4.6$  millisecc for the same 0.75 in. orifice.

### Conclusions

The above calculations indicate that the minimum time  $t_0$  of which the machine is capable varies from about 2 millisecc to about 5 millisecc depending on the type of specimen used, assuming an instantaneously acting valve. The type of response curve is shown in Fig. A-4, while a typical graph of the time  $t_0$  as a function of orifice size is shown in Fig. A-5.

If it is desired to decrease the time  $t_0$  for a given specimen to be loaded, the most practical method would be to decrease the dimensions of the machine, and increase the working pressure  $P_0$  accordingly. For example, if the working pressure is increased from 200 lb/in.<sup>2</sup> to 1400 lb/in.<sup>2</sup>, permitting the area of the piston to be reduced by a factor of 1/7, then the theoretical lower limit of  $t_0$  (see Eq. 10) is reduced by a factor of  $(1/7)^{3/4} = 0.23$ . Thus it should be possible to decrease the minimum time of which the machine is capable to about one fourth of its value by increasing the working pressure from 200 to 1400 lb/in.<sup>2</sup>, and decreasing the dimensions accordingly. If the water chamber were made in a more streamlined shape, tapering towards the orifice, this would also tend to reduce the time, and incidentally make the above analysis more reliable, since horizontal components of the velocity were neglected. However, such a change in the design might make the construction of the machine unduly expensive.

A third method of reducing the time would be to increase the area of the orifice. This would not be very effective if the orifice diameter already corresponded to a point in the flat portion of the curve (Fig. A-5). It would also tend to make the problem of a rapid opening of the valve more difficult to solve. However, if the overall dimensions were decreased, the orifice size might be increased somewhat in proportion to the piston size. Another objection to very much increase in the ratio of orifice area to piston area, thereby reducing the damping effect, would be the increased amount of "overshoot", which might result in vibrations of excessive amplitude during the test.

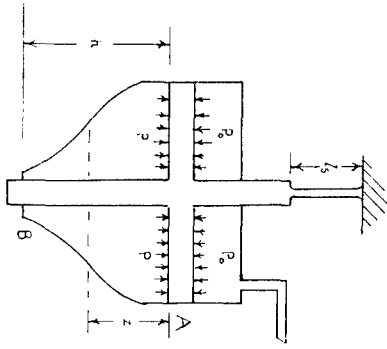


Fig. A-1 Schematic diagram of rapid load testing machine.

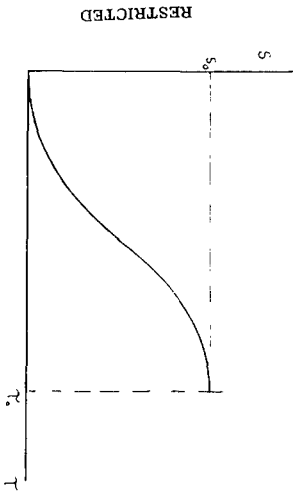


Fig. A-2 Schematic graph of  $S$  vs.  $t$  during the loading period.

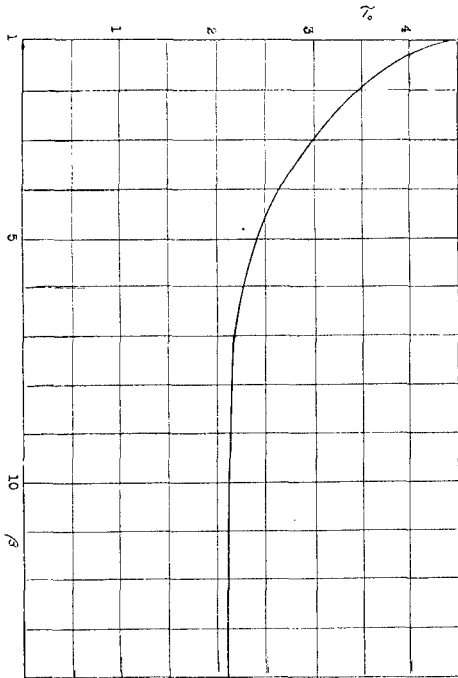


Fig. A-3 Relation between the dimensionless parameters  $\beta_0$  and  $\beta$ .

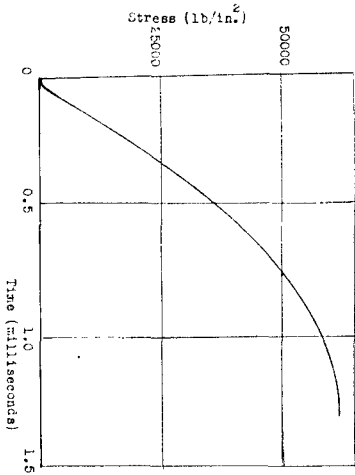


Fig. A-4 Stress vs. time for G-505 in. diameter steel specimen. Orifice diameter 0.75 in.

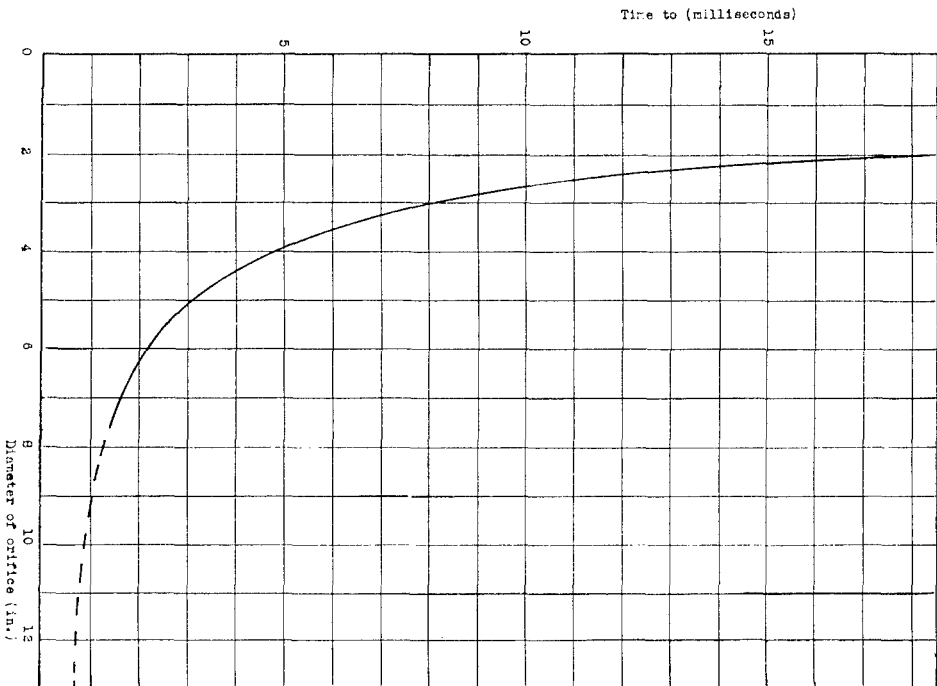


Fig. A-5 Time to vs. orifice diameter for G-505 in. diameter specimen. Maximum stress 60000 lb./in.<sup>2</sup>



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ABSTRACT

This report presents the design of equipment for studying the influence of rapid loading and time at load on the initiation of plastic deformation of metals in tension. The machine consists of a cylinder and platen actuated by a hydro-pneumatic system which permits a load as high as 20,000 lb to be applied to a specimen 0.500 in. in diameter in an interval of 0.005 sec or more. This load or a smaller load can be maintained on the specimen for any desired length of time greater than 0.001 sec. The application of the data obtained from tests with this equipment is considered.

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