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LIST OF SYMBOLS

M	Mass of Fluid
V	Volume
L	Length of each feed pipe
A	Cross Sectional Area
P	Pressure in pounds per square inch
d	Average weight density of fluid
ρ	Mass density of fluid
ρ_o	Mass density at zero pressure
f	Friction factor
v	Velocity of fluid in pipe
D	Inside diameter of pipe in inches
g	Acceleration due to gravity
t	Time in seconds from opening of feed valve
k	Compressibility of fluid
$(\ )_s$	Specimen
$(\ )_v$	Valve
$(\ )_a$	Accumulator
$(\ )_p$	Pipes
$(\ )_o$	Initial condition

A NEW CONCEPT FOR STUDYING PRESSURE VESSEL CONFIGURATIONS UNDER VERY HIGH PRESSURES AND LOADING RATES

Abstract

This report discusses the design and operation of a newly developed testing system for hydrodynamically simulating the pressure-time response typical of large caliber cannon. The system is capable of producing peak pressures in excess of 44,000 pounds per square inch in rise times of about three milliseconds. It is also capable of producing a wide variety of pressure-time curves at lower pressures and/or longer rise times. This system may be used for the study of a wide variety of pressure vessel configurations under high loading rates.

Cross-Reference Data

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Pressure vessels

Thick-wall cylinders

Industrial production

Stress analysis

DO NOT REMOVE THIS ABSTRACT FROM THE REPORT

CONCLUSIONS

The pressure-time response associated with the firing of a large caliber weapon has been accurately reproduced on full-scale cannon components using a hydrodynamic pressure system based on the rapid transfer of fluid from a liquid accumulator. This system, can be utilized for a wide variety of high loading rate thick-wall cylinder studies, and is capable of producing pressures in excess of 44,000 pounds per square inch in times as low as 3.2 milliseconds. The elapsed time at maximum pressure can be as low as 1 millisecond and varied over a wide range. The total cycle time, measured from zero pressure through the maximum to a point where the pressure has decayed to approximately 75 per cent of the maximum value, can be as low as 6 milliseconds. If desired, the lower portion of the pressure-time decay curve can also be accurately reproduced with a slight system modification.

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INTRODUCTION

It is often desirable, and in some cases necessary, to evaluate design parameters, study materials, and determine the operational characteristics of components in the laboratory under simulated service conditions. The desirability of such a procedure increases both with the complexity of the component and the extremeness of the service conditions where accurate theoretical solutions are impractical. This report discusses the design and functioning of a hydrodynamic pressure system capable of producing internal pressures, in a wide variety of heavy walled pressure vessels, in excess of 44,000 pounds per square inch in approximately 3 milliseconds. The intent of this system is to simulate the pressure and pressure-rise time in cannon type weapons, thus permitting the laboratory evaluation of the dynamic stress conditions, strength, and low cycle fatigue life characteristics, of components and materials associated with new weapon concepts, without the high expense of actual firing. This type of system also lends itself to a wide variety of studies into the dynamic stress-strain conditions and fatigue characteristics of pressure vessel type configurations subjected to high loading rates and high pressures.

RESULTS AND DISCUSSION

A system for producing high loading rates in pressure vessels must consist of two basic segments; a means for storing the required amount of energy and a means for rapidly releasing this energy and transferring it to the interior of the specimen. The following three basic methods for the storing of energy were considered:

1. A moving mass, accelerated by a prime mover and/or gravity, which has the advantage of being capable of high cyclic loading rates;

2. Gas charged accumulator which has the advantage that an accumulator of moderate volume need only be pressurized to slightly more than that required in the specimen;

3. Liquid charged accumulator.

The latter approach, consisting of a liquid charged accumulator with a high velocity release and fluid transfer system, was chosen based on the following considerations:

a. Flexibility

Since it is a purely hydraulic system, it can be used to test virtually any component under hydrodynamic loading conditions by simple piping modifications. The peak pressure can be varied over a wide range of values by simply changing the accumulator charging pressure. The rate of loading can be varied by changing the viscosity of the fluid used or by changing orifice or pipe sizes. High or low temperature capabilities can easily be added by heating or cooling the specimen without greatly affecting the remainder of the system.

b. Control

Very accurate control and reproducibility of peak pressure is inherent in this type of system since the peak pressure is directly a function of the compressibility of the liquid and the charging pressure. The former is very nearly a constant at any given pressure and the latter involves measuring and controlling a constant hydrostatic pressure; a rather simple instrumentation problem.

c. Safety

The hydraulic system offers the advantages over the gas system of much greater safety, due to lower stored energy, and fewer seal leakage problems.

Fluid Transfer Analysis

Prior to discussing the design and functioning of the individual components of the system, it is important to consider the controlling parameter--the fluid transfer from the feed accumulator to the specimen. The following approximate solution to the fluid transfer problem will be based on the testing of the specimen shown in figure 1, which is a new type of chamber section for a large caliber weapon.

To determine the pipe and orifice size requirements, a trial and error approach must be taken. However, only the solution for the $\frac{3}{4}$ inch internal diameter pipe and 44,000 pounds per square inch accumulator pressure will be shown since that was the condition finally chosen.

The following assumptions were made in the solution of the problem:

1. The compressibility of the fluid (water) is a constant between 0 and 50,000 pounds per square inch and is given by

$$\frac{\Delta V}{V_o} = 2.22 \times 10^{-6} P = k \qquad (1)$$

Equation (1) is a linear approximation of the compression curve found by Bridgman[1].

2. Any volume changes due to dilation of the accumulator and the piping are neglected.

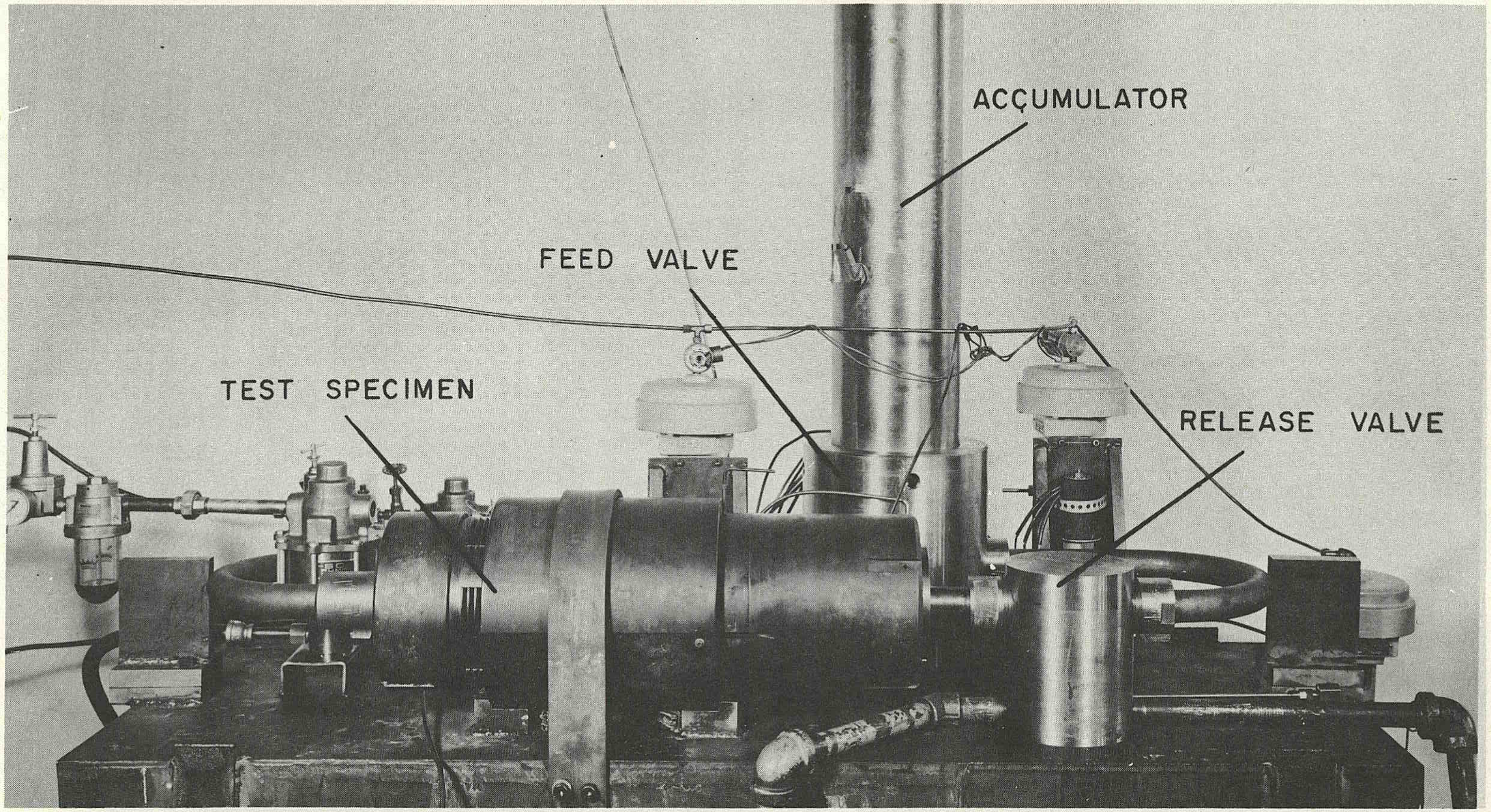

Figure 1. Specimen Mounted In Test System

3. The Reynold's number for the pipe flow exceeds 4×10^5. Therefore, the friction factor may be considered a constant and equal to 0.024 for $\frac{3}{4}$ inch diameter pipe[2].

4. The pressure in the pipe is equal to the pressure in the specimen.

The change in mass of fluid in the accumulator during fluid transfer is given by

$$\Delta M_a = \rho_o k (P_{ao} - P_a) V_a \tag{2}$$

The change in mass in the pipe is

$$\Delta M_p = \rho_o k P_s V_p \tag{3}$$

The change in mass in the specimen due to compression is

$$\Delta M_{sc} = \rho_o k P_s V_{so} \tag{4}$$

and that due to dilation is

$$\Delta M_{sd} = \rho_o (1 + k P_s) k_s P_s V_{so} \tag{5}$$

where k_s is defined as the change in volume of the specimen due to dilation divided by the initial volume.

The total change in mass in the specimen thus becomes

$$\Delta M_s = \rho_o P_s V_{so} \left[k + (1 + kP_s) k_s \right] \tag{6}$$

Since the change in mass in the accumulator must equal the change in mass in the pipes and specimen

$$k(P_{ao} - P_a)V_a = P_s \left[k (V_p + V_s) + k_s V_s (1 + kP_s) \right] \tag{7}$$

For the conditions of this test we may neglect the kP_s of the $(1 + kP_s)$ term in equation (7). This will result in an error of less than five per cent and will thus allow the following constants to be defined:

$$C_a = k V_a \tag{8}$$

$$C_s = k(V_p + V_s) + k_s V_s \tag{9}$$

Equation (7), therefore, becomes

$$C_a(P_{ao} - P_a) = C_s P_s \tag{10}$$

or

$$P_a - P_s = P_{ao} - (1 + \frac{C_s}{C_a}) P_s \tag{11}$$

The pressure difference between the accumulator and the specimen may be considered to be composed of three factors, namely: (1) velocity head loss in the feed valve, (2) a frictional head loss in the pipe, and (3) a velocity head loss on entry into the specimen. These are given by the following equation

$$P_a - P_s = \frac{d}{2g}\left[\frac{fL}{D_p} v_p^{\,2} + v_p^{\,2} + F_v\, v_v^{\,2}\right] \tag{12}$$

where F_v is a velocity loss coefficient for the valve.

From continuity of flow requirements

$$v_v^{\,2} = \left(v_p \frac{2A_p}{A_v}\right)^2 = v_p^{\,2}\, 4 \left(\frac{D_p}{D_v}\right)^4 \tag{13}$$

Therefore

$$P_a - P_s = \frac{d}{2g}\left[\frac{fL}{D_p} + 1 + 4F_v \left(\frac{D_p}{D_v}\right)^4\right] v_p^{\,2} \tag{14}$$

Letting

$$\frac{d}{2g}\left[\frac{fL}{D_p} + 1 + 4F_v \left(\frac{D_p}{D_v}\right)^4\right] = F \tag{15}$$

in equation (14) yields

$$P_a - P_s = F\, v_p^{\,2} \tag{16}$$

From continuity of flow entering the pipes

$$\frac{1}{\rho}\frac{dM}{dt} = 2A_p\, v_p \tag{17}$$

Combining equations (16) and (17) yields

$$P_a - P_s = \frac{F}{4\rho^2 A_p^{\,2}}\left(\frac{dM}{dt}\right)^2 \tag{18}$$

and combining equations (11) and (18) yields

$$\frac{dM}{dt} = \frac{2\rho A_p}{F}\sqrt{P_{ao} - \left(1 + \frac{C_s}{C_a}\right)P_s} \tag{19}$$

From equations (3), (6) and (9) the mass of fluid entering the pipes is given by

$$M = \rho_o\, C_s\, P_s \tag{20}$$

therefore, since $\rho = \rho_o\,(1 + kP_s)$

$$\frac{dP_s}{dt} = \frac{2A_p\,(1 + kP_s)}{C_s\sqrt{F}}\sqrt{P_{ao} - (1 + \frac{C_s}{C_a})P_s} \tag{21}$$

Again neglecting the kP_s term, separating variables and integrating from 0 to P_s and 0 to t gives the following equation for the pressure time curve in the specimen

$$t = \frac{\sqrt{P_{ao}} - \sqrt{P_{ao} - (1 + \frac{C_s}{C_a})\,P_s}}{\frac{A_p}{C_s\sqrt{F}}\;(1 + \frac{C_s}{C_a})} \tag{22}$$

The constants in the above equation are determined as follows:

C_s due to compression, assuming a combined specimen and pipe volume of 72 cubic inches, is equal to (2.22) (72) X 10^{-6}

C_s due to dilation, assuming a straight, closed end cylinder, is equal to 1.18 X 10^{-4}

Therefore, C_s = .278 X 10^{-3}

Assuming an accumulator volume of 522 inches, C_a = 2.22 X 522 X 10^{-6}

Therefore C_a = 1.16 X 10^{-3}.

An examination of the probable flow pattern through the feed valve indicates that most, but not all, of the kinetic energy of the fluid at the seat is lost passing through the valve. Based on this assumption, a value of F_v equal to 0.75 was used. Therefore

$$F = \frac{.0362}{773}\left[\frac{(.024)(56)}{0.75} + 1 + (0.75)(4)\left(\frac{.75}{.5}\right)^4\right]$$

$$F = 8.42 \text{ X } 10^{-4}$$

Using these constants and an initial accumulator pressure of 44,000 pounds per square inch in equation (22) gives the computed pressure-time curve shown in figure 2. Figure 2 also shows the experimentally obtained pressure-time curve. As can be seen, the computed and experimental curve

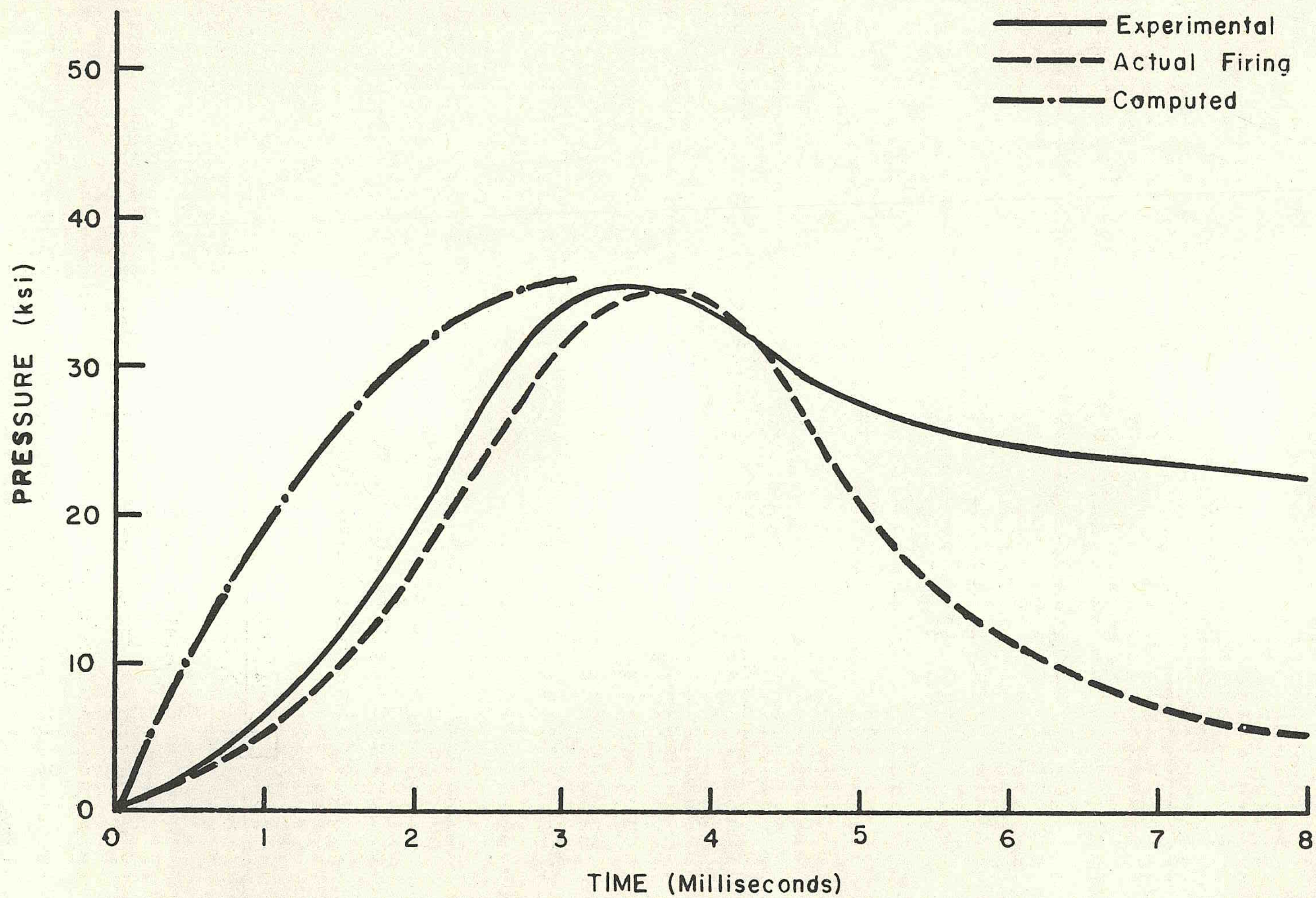

Figure 2. Computed, Actual Firing, Experimental Pressure-Time Curve

are in very good agreement with respect to total rise time. The difference in the shape of the curves is probably due to the inertial forces involved with accelerating and decelerating the fluid which were neglected in the above analysis.

High Speed Valve

In order to release the energy stored in the accumulator into the test specimen in the required loading time, a valve which has the following operating characteristics is required: (1) Opening time must be less than one millisecond. (2) Time of opening must be controllable within one-half millisecond. (3) Must be capable of withstanding 50,000 pounds per square inch pressure both in the closed and open position. (4) Must have a volumetric flow capacity of in excess of 2,000 gallons per minute.

In order to obtain these requirements, a pilot operated valve utilizing the differential area principle was designed. A schematic diagram of this valve is shown in figure 3. The operation of the valve is as follows: A pilot pressure of about 5,000 pounds per square inch is introduced into the area under the plunger. This is sufficient to hold the valve closed against the seat with an accumulator pressure of 50,000 pounds per square inch. The valve may now be opened by either of two methods. The pressure under the plunger may be released or a pressure equal to or greater than the pilot pressure may be introduced above the plunger through the trigger port. Either of these actions will cause the net force on the plunger to assume a downward direction. As the plunger begins to move, the high pressure fluid at the seat will move into the area above the plunger, thereby exerting a very large downward force on the plunger, which will accelerate rapidly downward, compressing the fluid beneath it. The volume of fluid under the plunger is enough so that the compression of this fluid will allow the plunger to move far enough to uncover the outlet ports. This means that a high flow rate out of the bottom of the valve is not required, allowing the use of small orifice tubing and valves in the pilot pressure system. In order to prevent premature firing of the valve in case of leakage at the seat, the trigger port is vented to the atmosphere through an air operated dump valve which is closed just before the valve is fired.

Pressure Release Valve

Along with the requirement for building up a maximum pressure in a given period of time, it is also necessary to relieve this pressure after a specified time and at a given rate. For the particular test described

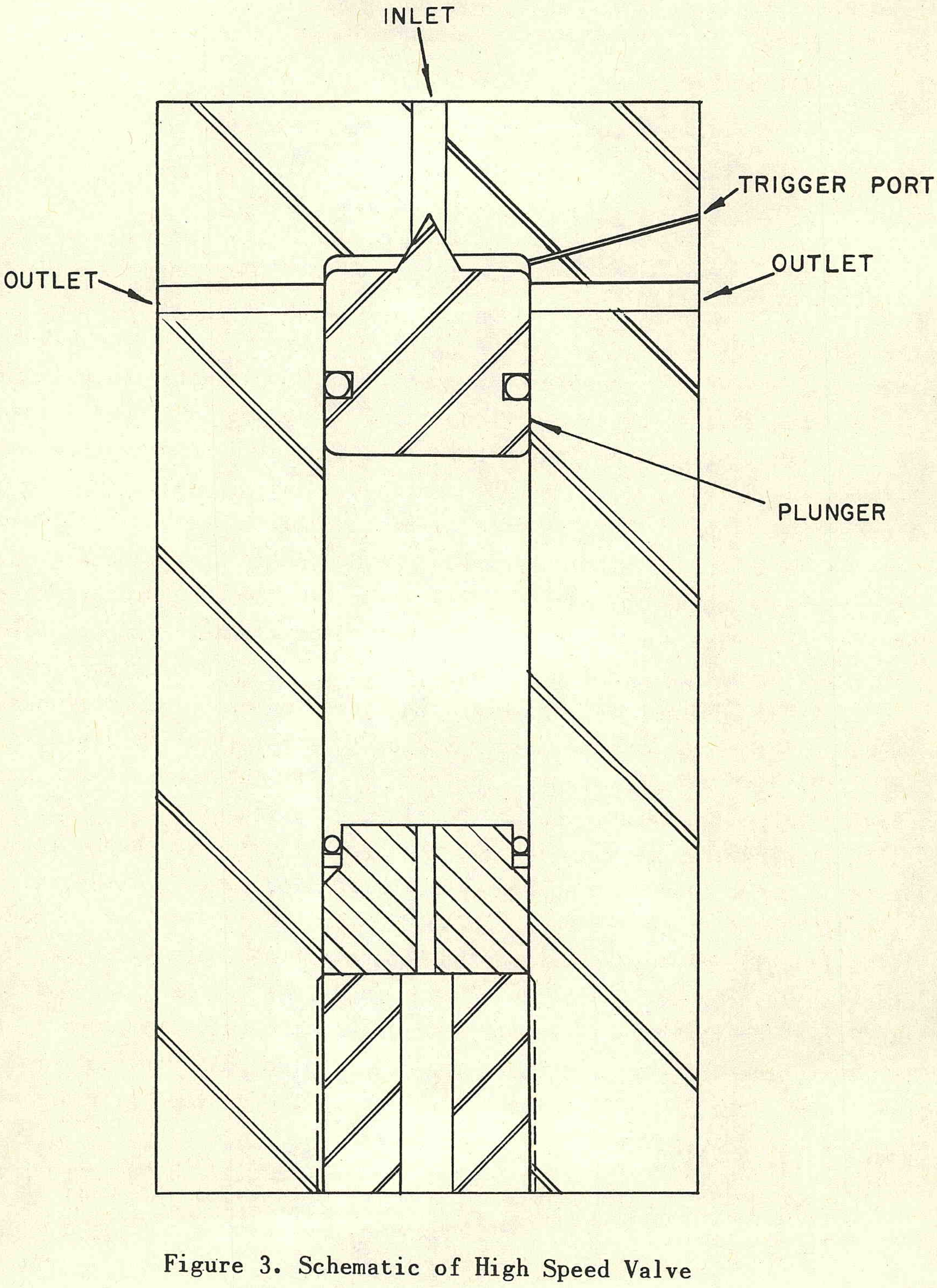

Figure 3. Schematic of High Speed Valve

herein, the time of pressure and decay rate should closely approximate that associated with the actual firing of the weapon involved near the maximum pressure.

The pressure release valve is identical in principle and operation to the feed valve. It is opened by a hydraulic "feed back" circuit connecting its trigger port to the feed valve trigger port. Due to the fluid trapped under the plunger, the release valve will reclose at approximately 10,000 pounds per square inch in the system. Although this condition is not considered harmful, as will be subsequently discussed, it is possible to drop the residual pressure to zero.

Description of Operation

The system consists of three primary segments, (1) an accumulator with a high speed valve, (2) piping for controlled fluid transfer to the specimen and (3) a high speed pressure release valve. A schematic of these components assembled into the final system, along with the test specimen, is shown in figure 4.

It may be helpful now to consider the sequence of functions that occur when an attempt is made to simulate the pressure-time curve for firing in the test specimen. Initially the accumulator is charged to a pressure of approximately 44,000 pounds per square inch using an intensifier pumping system. The exact accumulator pressure, of course, is controlled by the desired maximum pressure in the specimen. The high speed feed valve, which is located just below the accumulator and built integral with it, is fired by releasing the pressure under the plunger. Fluid now flows out of both sides of the feed valve and into both ends of the test chamber until an equilibrium pressure, in this case about 35,000 pounds per square inch, is attained. The volume of fluid transferred during this process is kept to a minimum by steel filler bars which occupy most of the volume of the test chamber. The remaining volume is pre-filled with fluid at atmospheric pressure. The fluid is piped into both ends of the specimen to assure uniform pressure distribution and to reduce external reaction forces produced by the acceleration of the fluid.

When the test pressure is reached in the chamber, the high-speed release valve is opened by the flow of fluid through the controlled orifice line connecting the trigger port of the feed valve with the trigger port of the release valve. As has been previously mentioned, the release valve will reclose at about 10,000 pounds per square inch pressure in the chamber.

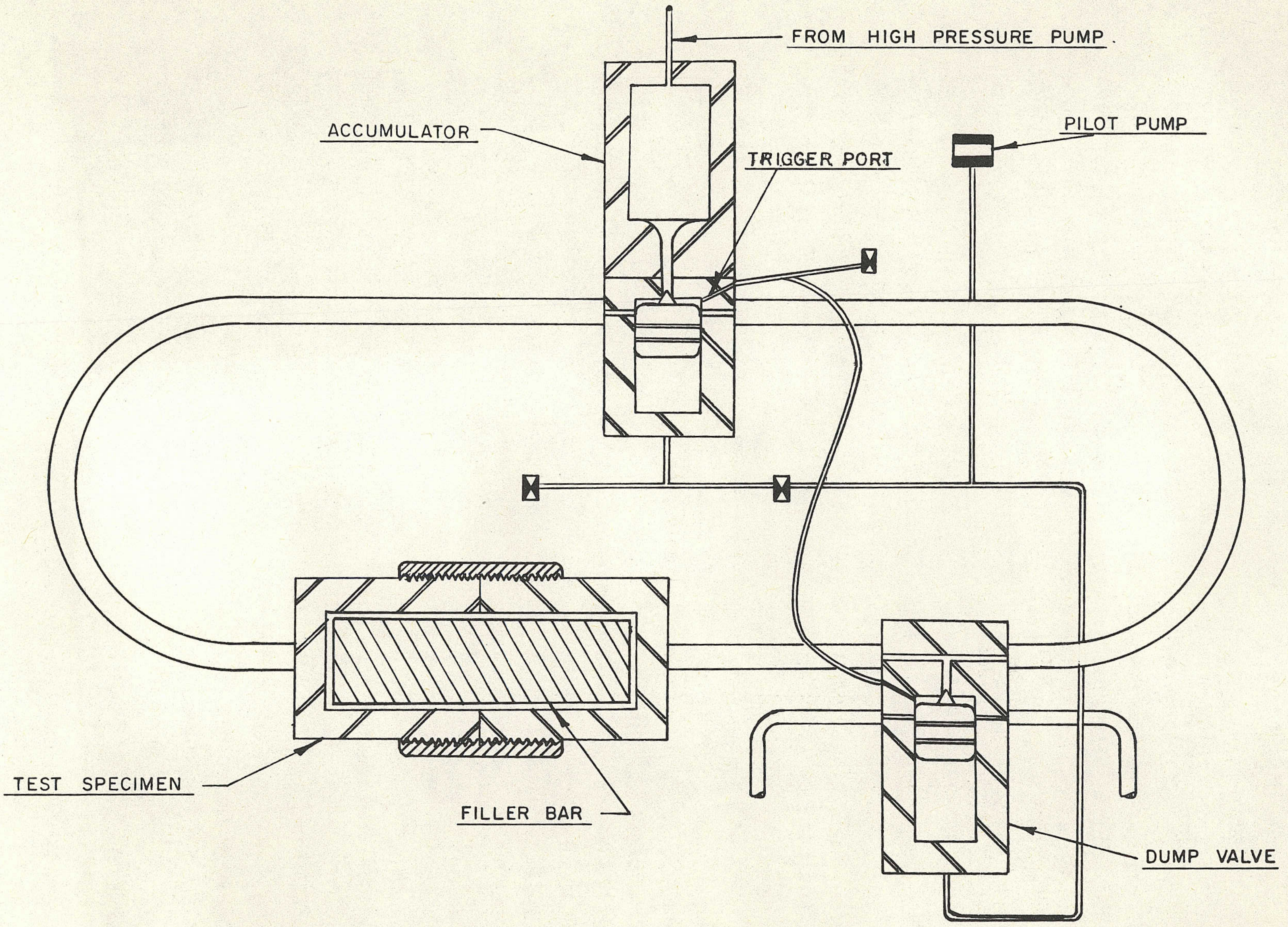

Figure 4. Schematic of Test System

This pressure will drop to zero when the valve connected to the trigger ports is opened due to leakage through the outlet ports of the feed valve, past the plunger and out the trigger port.

Instrumentation

Initial pressure-time data was obtained from SR-4 strain gages mounted on the specimen. These gages were connected to a Honeywell Visicorder with a 3300 cycle, fluid damped, galvanometer. Amplification was provided by an Ellis BAM-1 strain gage amplifier and a transistorized, cathode-follower-type, driver-amplifier. This measuring system was calibrated by building up a static pressure inside the specimen and calibrating the strain gages against a Manganin pressure cell and Wheatstone bridge. Subsequent calibration of the amplifier system was accomplished by shunting an accurate resistance across one arm of the strain gage bridge.

Pressure-time curves were also obtained using a Kistler quartz crystal ballistic pickup, "Piezo Calibrator" and oscilloscope. The pressure-time curves obtained by both systems agreed very closely.

Results

As was previously mentioned, this system was initially designed to reproduce the pressure-time curve associated with the firing of a large caliber weapon. Figure 2 depicts the general type of pressure-time curve of concern along with a sample of that obtained in the laboratory. For a pressure of 35,000 pounds per square inch, the obtained rise time is 3.2 milliseconds with a time at peak pressure of 1 millisecond. As can be seen, there is very close correlation between the experimental and actual firing curves in the rise and time at pressure portion.

It can also be seen from figure 2 that there is good agreement in the initial portion of the decay curve to a pressure of approximately 75 per cent of the peak value, with a wide deviation at the lower pressures due to the reclosing of the pressure release valve. For the testing program involved, this deviation was not considered harmful since it did not become serious until the pressure had reached a negligible level. A lower pressure system, currently being installed, will have a different type of pressure release valve which will allow the reproduction of the complete decay portion of the pressure-time curve.

Figure 5 shows the pressure-time curve associated with a peak pressure of 44,000 pounds per square inch. In this case also, the pressure rise time was 3.2 milliseconds.

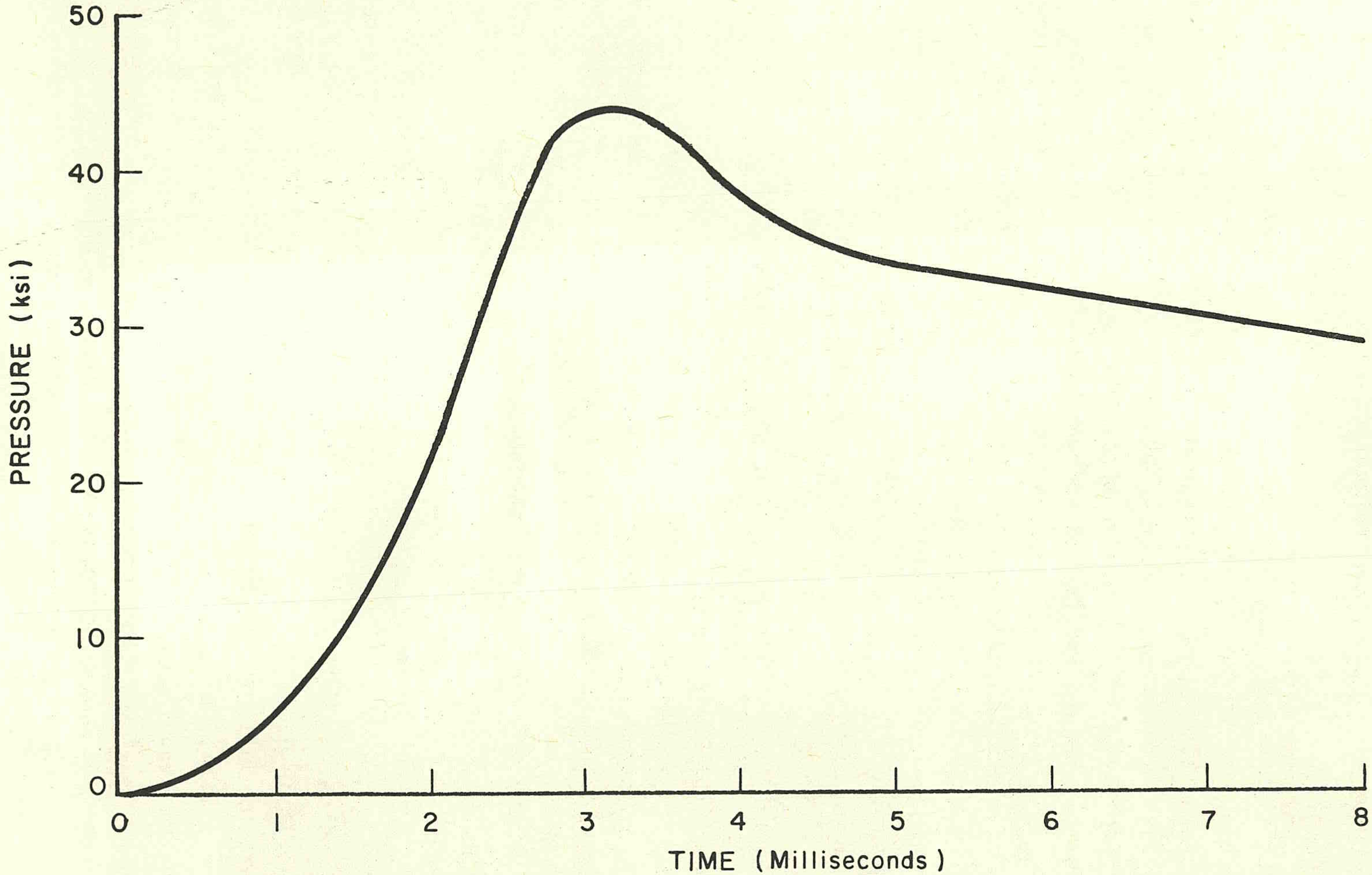

Figure 5. Pressure-Time Curve For 44,000 Pounds Per Square Inch Peak Pressure

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