A New Concept for Studying Pressure Vessel Configurations Under High Pressures and Loading Rates

By T. E. DAVIDSON and D. P. KENDALL

This paper 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 above 44,000 psi in rise times of about 3 millisec and a wide variety of pressure-time curves at lower pressures and longer rise times. It may be used for a wide variety of pressure-vessel configurations under high loading rates.

LIST OF SYMBOLS

M = mass of fluid, lb mass,
V = volume, cu in.,
L = length of each feed pipe, in.,
A = cross sectional area, sq in.,
P = pressure, psi,
d = average weight density of fluid, lb per cu in.,
ρ = mass density of fluid, lb mass per cu in.,
ρ_0 = mass density at zero pressure,
f = friction factor,
v = velocity of fluid in pipe, in. per sec,
D = inside diameter of pipe, in.,
C = constant, cu in. per psi,
F = velocity loss coefficient, psi sec^2 per in.^{2},
g = acceleration due to gravity, in. per sec^2,
t = time from opening of feed valve, sec,
k = compressibility of fluid, sq in. per lb,
$(\ )_s$ = specimen,
$(\ )_v$ = valve,
$(\ )_a$ = accumulator,
$(\ )_p$ = pipes, and
$(\ )_o$ = initial condition

It is often desirable and sometimes necessary to evaluate design parameters, study materials, and determine the operational characteristics of components in the laboratory under simulated service conditions, particularly when the complexity of the component or of service conditions render accurate theoretical solutions impractical. This paper discusses the design and operation of a hydrodynamic pressure system capable of producing internal pressures in a wide variety of heavy-wall pressure vessels above 44,000 psi in approximately 3 millisec. By simulating the pressure and pressure-rise time in cannon, this system permits laboratory evaluation of the dynamic stress conditions, strength, and low-cycle fatigue characteristics of components and materials, 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 vessels subjected to high loading rates and high pressures.

A system for producing high loading rates in pressure vessels must be able to (1) store the required amount of energy, and (2) rapidly release this energy and transfer it to the interior of the specimen. The following three basic methods for the storing of energy were considered:

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1. A moving mass, accelerated by a prime mover or gravity, which is capable of high cyclic loading rates,

2. A gas-charged accumulator, of moderate volume, which need be pressurized to only slightly more than that required in the specimen,

3. A liquid-charged accumulator.

Liquid-charged Accumulator

A liquid-charged accumulator with a high velocity release and fluid transfer system was chosen for the following reasons:

1. *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.

2. *Control*.—Accurate control and reproducibility of peak pressure are inherent in this type of system since the peak pressure is a direct function of the compressibility of the liquid and the charging pressure. The compressibility is very nearly a constant at any given pressure, and it is only a simple instrumentation problem to measure and control a constant hydrostatic pressure.

3. *Safety*.—In contrast to the gas system, the hydraulic system offers much greater safety, due to lower stored energy, and fewer seal-leakage problems.

Fluid Transfer Analysis

Before 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 is based on the testing of a new type of chamber section for a large-caliber weapon (Fig. 1).

Although a trial and error approach must be taken to determine the pipe and orifice sizes, only the final solution for the $\frac{3}{4}$-in. internal diameter pipe and 44,000 psi accumulator pressure will be shown.

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

1. The compressibility of the fluid (water) is a constant between 0 and

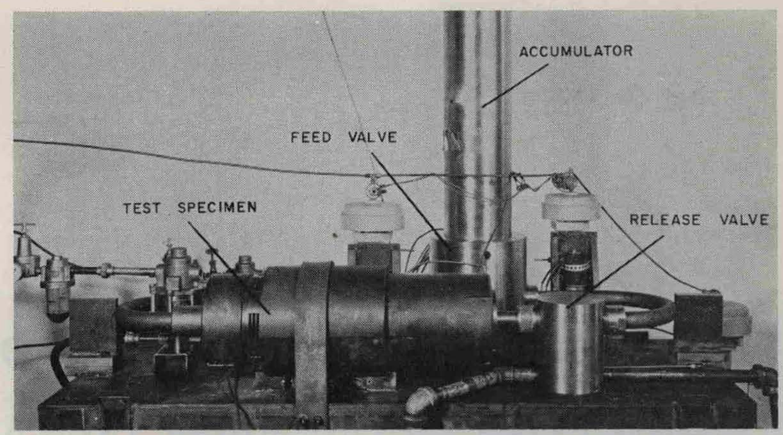

Fig. 1.—Specimen mounted in test system.

50,000 psi and is given by

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

Eq 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.

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 $\frac{3}{4}$-in. diam 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

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

The change in mass in the pipe is

$$\Delta M_p = \rho_o k P_s V_p \ldots\ldots\ldots (3)$$

The change in mass in the specimen due to compression is

$$\Delta M_{sc} = \rho_o k P_s V_{so} \ldots\ldots\ldots (4)$$

The change in mass in the specimen due to dilation is

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

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

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

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

$$k(P_{ao} - P_a)V_a = P_s[k(V_p + V_{so}) + k_s V_{so}(1 + kP_s)] \,..(7)$$

For the conditions of this test we may neglect the kP_s of the $(1 + kP_s)$ term in Eq 7. This will result in an error of less than 5 per cent and will allow the following constants to be defined:

$$C_a = kV_a \ldots\ldots\ldots\ldots (8)$$

$$C_s = k(V_p + V_s) + k_s V_{so} \ldots (9)$$

Equation 7 therefore becomes

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

or

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

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

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

From continuity of flow requirements

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

Therefore

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

Letting

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

in Eq 14 yields

$$P_a - P_s = F v_p^2 \ldots\ldots\ldots (16)$$

From continuity of flow entering the pipes

$$\frac{1}{\rho}\frac{dM}{dt} = 2A_p v_p \ldots\ldots\ldots (17)$$

Combining Eqs 16 and 17 yields

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

and combining Eqs 11 and 18 yields

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

[1] P. W. Bridgman, "The Physics of High Pressure," G. Bell and Sons Ltd., London, (1949).

[2] G. V. Shaw and A. W. Loomis, "Cameron Hydraulic Data," Ingersoll-Rand Co., New York, (1951).

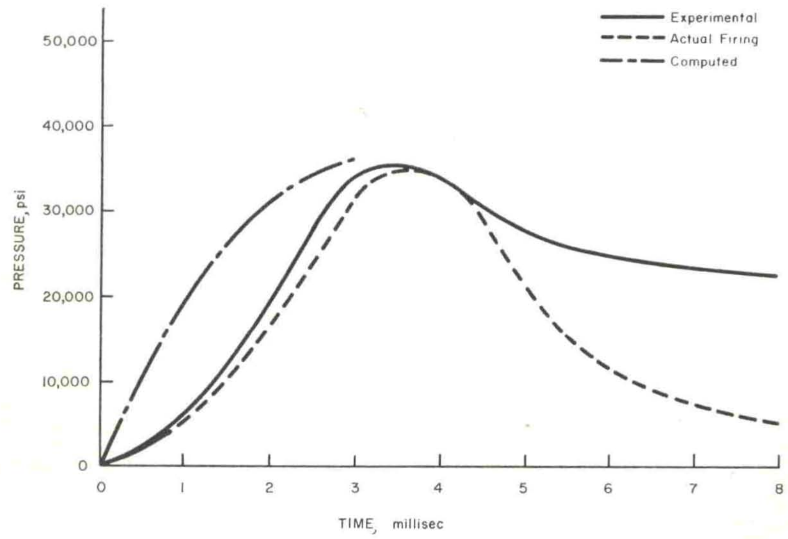

Fig. 2.—Experimental, actual firing, and computed pressure-time curves.

From Eqs 3, 6, and 9, the mass of fluid entering the pipes is given by

$$M = \rho_o C_s P_s \qquad (20)$$

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

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

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:

$$t = \frac{\sqrt{P_{ao}} - \sqrt{P_{ao} - \left(1 + \frac{C_s}{C_a}\right) P_s}}{\frac{A_p}{C_s\sqrt{F}}\left(1 + \frac{C_s}{C_a}\right)} \qquad (22)$$

The constants in the above equation are determined as follows:

That part of C_s due to compression, assuming a combined specimen and pipe volume of 72 cu in., is equal to (2.22) (72) $\times 10^{-6}$.

That part of C_s due to dilation, assuming a straight, closed-end cylinder, is equal to 1.18×10^{-4} from the well-known Lamé equations.[3]

Therefore:

$$C_s = 0.278 \times 10^{-3} \text{ cu in. per psi}$$

Assuming an accumulator volume of 522 cu in., $C_a = 2.22 \times 522 \times 10^{-6}$

Therefore:

$$C_a = 1.16 \times 10^{-3} \text{ cu in. per psi}$$

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:

$$F = \frac{0.0362}{773} \left[\frac{(0.024)(56)}{0.75} + 1 + (0.75)(4)\left(\frac{0.75}{0.5}\right)^4\right]$$

$$F = 8.42 \times 10^{-4} \text{ psi sec}^2 \text{ per in.}^2$$

Using these constants and an initial accumulator pressure of 44,000 psi in Eq 22 gives the computed pressure-time curve shown in Fig. 2. Also shown in Fig. 2 is the experimentally obtained pressure-time curve. The computed and experimental curves are in 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

To release the energy stored in the accumulator into the test specimen in the required loading time, a valve having the following operating characteristics is required: (1) opening time must be less than 1 millisec, (2) time of opening must be controllable within $\frac{1}{2}$ millisec, (3) the valve must be capable of withstanding 50,000 psi pressure in the closed and open positions, and (4) it must have a volumetric flow capacity greater than 2000 gal per min.

To obtain these requirements, a pilot-operated valve using the differential-area principle was designed (Fig. 3). A pilot pressure of about 5000 psi, introduced into the area under the plunger, is sufficient to hold the valve closed against the seat with an accumulator pressure of 50,000 psi. The valve may now be opened either by releasing the pressure under the plunger or by introducing a pressure equal to or greater than the pilot pressure above the plunger through the trigger port. Either of these actions will cause the net force on the plunger to become downward. As the plunger begins to move, the high-pressure fluid at the seat will move into the area above the plunger and exert a large downward force on the plunger,

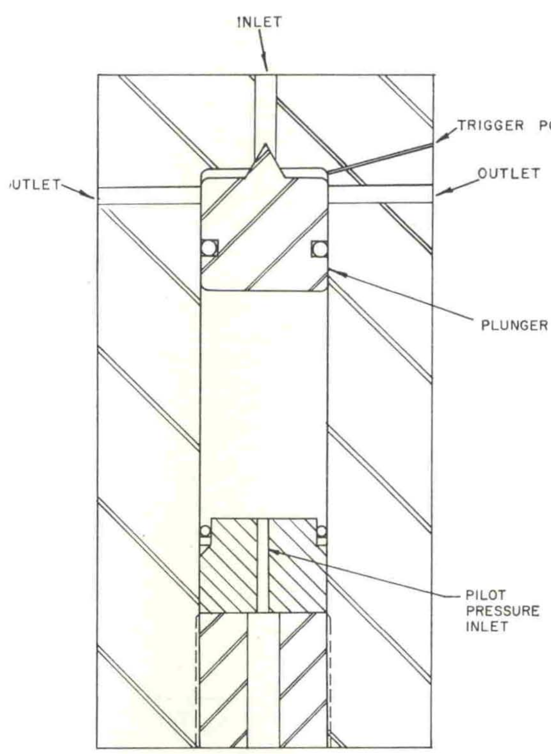

Fig. 3.—Schematic of high-speed valve.

[3] S. Timoshenko, "Strength of Materials, Part II," D. Van Nostrand, Inc., New York, (1942).

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. Since a high flow rate out of the bottom of the valve is not required, small orifice tubing and valves may be used in the pilot pressure system. To prevent premature firing of the valve in case of leakage at the seat, the trigger port is vented to atmosphere through an air-operated dump valve which is closed just before the valve is fired.

Pressure-Release Valve

In addition to 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 this particular test, the time at 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 feedback circuit connecting its trigger port to the feed-valve trigger port. Owing to the fluid trapped under the plunger, the release valve will reclose at a pressure of approximately 10,000 psi in the system. Although this condition is not considered harmful, it is possible to drop the residual pressure to zero.

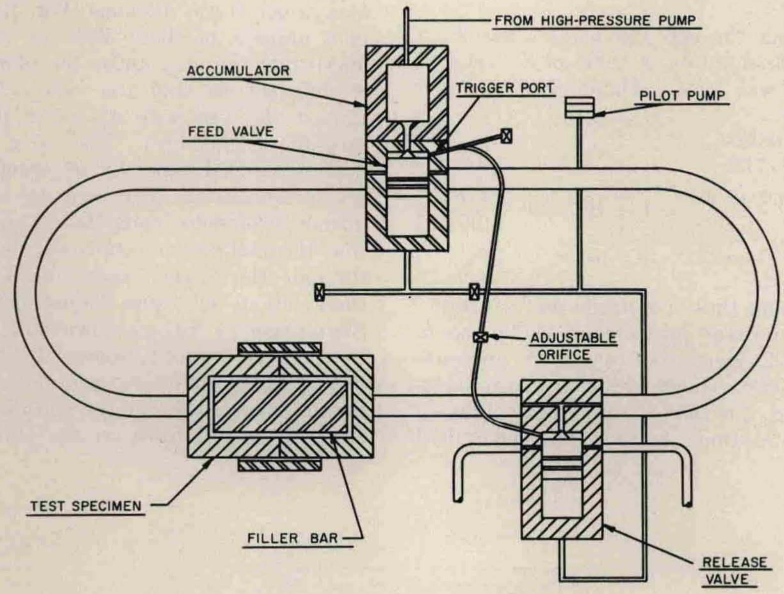

Fig. 4.—Schematic of test system.

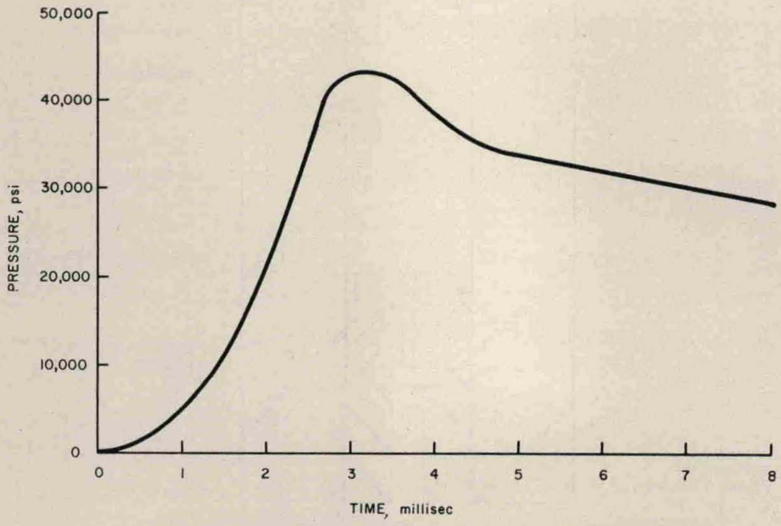

Fig. 5.—Pressure–time curve for 44,000 psi peak pressure.

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 Fig. 4.

Following is the sequence of events 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 psi 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, located just below the accumulator and built integrally 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 psi, 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 pressure introduced above the plunger through the line connecting the trigger port of the feed valve with the trigger port of the release valve. The required time delay is obtained by throttling this flow with an adjustable orifice. The release valve will reclose at about 10,000 psi pressure in the chamber. 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 were obtained from SR-4 strain gages mounted on the specimen. These gages were connected to an optical oscillograph with a 3300-cycle, fluid-damped, galvanometer. Amplification was provided by a d-c 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 wire pressure cell and Wheatstone bridge. The amplifier system was calibrated by shunting an accurate resistance across one arm of the strain-gage bridge.

Pressure-time curves were also obtained using a piezoelectric quartz crystal ballistic pickup and oscilloscope. The pressure-time curves obtained by both systems agreed closely.

Results

This system was designed to reproduce the pressure-time curve associated with the firing of a large-caliber weapon. Figure 2 shows the general type of pressure-time curve of concern along with a sample of the curve obtained in the laboratory. For a pressure of 35,000 psi, the rise time obtained was 3.2 millisec with a time at peak pressure of 1 millisec. There is close correlation between the experimental and actual firing curves in the rise and time-at-pressure portion. Figure 5 shows the pressure-time curve associated with a peak pressure of 44,000 psi. In this case also, the pressure-rise time was 3.2 millisec.

Figure 2 shows 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 owing 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.

Summary

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, which can be used for a wide variety of high-loading-rate, thick-wall cylinder studies, can produce pressures in excess of 44,000 psi in times as low as 3.2 millisec. The elapsed time at maximum pressure can be as low as 1 millisec 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 millisec. If desired, the lower portion of the pressure-time curve can also be accurately reproduced with a slight system modification.