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.,
 = mass density at zero pressure,
 = 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).

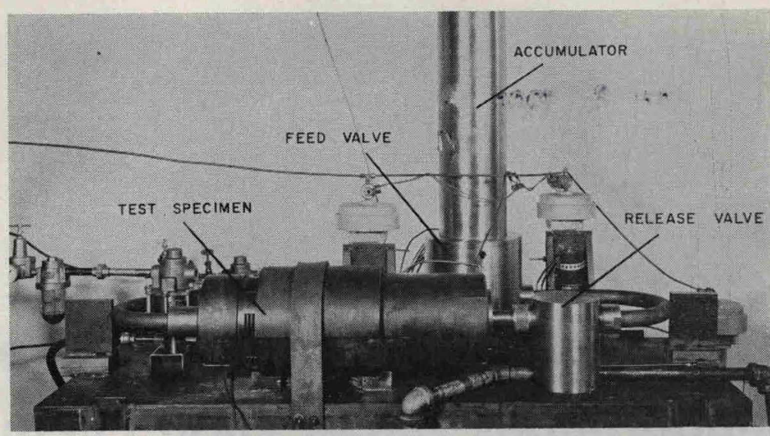

Fig. 1.—Specimen mounted in test system.

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 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

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

The change in mass in the pipe is

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

The change in mass in the specimen due to compression is

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

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

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

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

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

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:

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

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

Equation 7 therefore becomes

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

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:

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

From continuity of flow requirements

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

Therefore

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

Letting

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

in Eq 14 yields

$$P_a - P_s = Fv_p^2 \ldots\ldots.(16)$$

From continuity of flow entering the pipes

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

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

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

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[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).