Performance Study of a Ballistic Compressor

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An experimental and theoretical investigation of the performance of a ballistic compressor is described. Comparison of the computed ideal behavior of the ballistic compressor with experimental test pressures show that the ideal theory overestimates the generated pressure by as much as 30%. An assessment of the factors responsible for the nonideal behavior of the ballistic compressor is given. These factors include: valve losses, external and internal (blow-by) leakage, friction of the inertial piston, nonideal behavior of the driver and test gases, shock wave formations, and heat losses. The valve losses and both the internal and external leakages have been measured experimentally. A parametric study shows that the choice of equation of state of the test gas is not very critical. The heat loss experienced by the test gas was estimated to be 1% of the total energy imparted to the test gas. Shock-wave formations have negligible effect on the performance of the ballistic compressor for the mass of the inertial piston (1.34 kg) used.

Nomenclature

A_c = cross-sectional area of compression chamber
C_D = discharge coefficient
F = friction of the inertial piston
h = enthalpy
h = radial clearance of the piston
K = thermal conductivity
L_c = effective length of compression chamber
L_p = length of the piston
m = mass
m_L = mass of blow-by gas
m_p = mass of piston
P = pressure
Q_c = conductive heat loss in the test section
Q_R = radiative heat loss in the test section
q_H^D = heat loss in driver section
q_H^T = heat loss in test section
q_L^D = energy transfer by blow-by driver gas
q_L^T = energy transfer by blow-by test gas
R = gas constant
t = time
T = temperature
u = internal energy
u° = internal energy of ideal gas
v = velocity
V = volume
V_p = velocity of piston
x = distance along the compression chamber
y = radial distance
W_D = total work done by the driver gas
α = thermal diffusivity
γ = ratio of specific heats
ρ = density
μ = viscosity
σ = Stefan-Boltzmann constant

Subscripts

1 = initial conditions
2 = final conditions
D = driver gas conditions
R = reservoir conditions
T = test gas conditions
W = bore surface conditions

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Introduction

THE free-piston ballistic compressor has been used by a number of investigators to study the physical and chemical properties of gases under high-pressure and high-temperature conditions. The transient compression of the test gas achieved in this apparatus circumvents the temperature limitation of the static compression techniques and enables test gas temperature levels of up to 10,000°K and higher to be realized. Personnel at the U.S. Naval Ordnance Laboratories[1-4] have engaged in ballistic compressor development and its uses over the past 20 years. Among other investigators who have studied the free-piston-type ballistic compressor as a source of high-pressure and high-temperature gases are Longwell et al.,[5] Ryabinin,[6] and Minardi.[7] A comprehensive review of the ballistic compressor development and applications has been written recently by Lalos and Hammond.[8]

A major problem in the utilization of the ballistic compressor as a scientific tool is the determination of the state of the test gas as a function of time during the cycle of compression-rarefaction. Pressure measurements are made readily, but the temperature is not measured as easily. Spectroscopic temperature measurements of the test gas in the ballistic compressor have been made at NOL.[8,9] The "brightness-emissivity" method, which is based on the solution of the radiative transfer equation for a homogeneous gas in local thermodynamic equilibrium, was used. The measured temperatures, using helium as the test gas, are about 25% lower than the temperatures calculated based on ideal isentropic relations in the 3000°-6000°K temperature range. Correspondingly, the experimentally obtained peak pressures are less than the predicted pressures. Typically, for a reservoir pressure of 3.04 MN/m^2 and initial test gas pressure of 0.1 MN/m^2, the predicted peak pressure was 204 MN/m^2, whereas the experimentally obtained peak pressure was 101 MN/m^2. A number of the factors responsible for the deviation of the ballistic compressor performance from the ideal expectations have been accounted for analytically by Longwell et al.[5] and they assumed that the friction acting on the piston is proportional to the velocity of the piston. In their theoretical analysis, Takeo et al.[10] took into account gas leakage, viscous effects, and heat losses. For the monatomic gases used in the experiments, the Van der Waal equation of state, with coefficients varying with temperature, was assumed to be applicable. The effective radial clearance of the piston and the heat-transfer coefficient [h = heat transfer to the wall/(gas temperature − wall temperature)] were adjusted so that the computed peak pressure and minimum volume of the test gas agreed with the experimentally measured values. Predictions were obtained for a range of peak pressures from 48-168 MN/m^2. These nonideal calculations showed that the ideal theory predicts final temperatures that are greater, by a factor of 2, than the temperature obtained by nonideal calculations.

This paper presents an experimental and analytical study of the performance of the Princeton University ballistic compressor. The experimental performance curve of the ballistic compressor (test gas peak pressure vs driver gas initial pressure) was obtained for the range of test gas peak pressure from 50 to 427 MN/m^2 (i.e., about 2.5 times the pressure attained by Takeo et al.[10]). Nitrogen and air were used as the test gases for the results reported herein. It was not feasible, within the scope of this research, to measure the temperature directly by use of a method such as the brightness-emissivity method. It was felt that an evaluation of nonideal behavior based on some relatively simple diagnostic tests would serve the purpose of the present research more practically than engaging in a more elaborate experimental program of direct measurement of the temperature. It is felt that this approach has been successful within the required dependability of many engineering studies.

The analytical study consisted of the development of a theoretical analysis of the dynamic behavior of the ballistic compressor, which accounts for real-system effects (valve losses, leakage, friction, etc.). The factors that may be responsible for the deviation in the performance of the ballistic compressor from ideal are: a) behavior of the driver and test gases which deviate from that of a perfect gas, b) valve losses due to viscosity (pressure drop experienced by the driver gas flowing through the actuating valve and transition section), c) internal leakage (piston blow-by), d) external leakage, e) friction of the piston, f) generation of shock waves, and g) heat losses to the tube walls. The importance of each of the preceding factors has been assessed. For the evaluation of the effect of valve losses and external and internal leakage, the effect of each was estimated based on experimentally measured quantities at several conditions.

The ballistic compressor discussed in this paper was developed for studies on the erosive effects of high-temperature, high-pressure gases on metals. In order to allow meaningful correlations of the erosion results, it is important to know the state of the test gas. Results of erosion studies will be reported in a separate paper. The focus of this paper is on the methods used to determine the test gas state, how the various losses were evaluated, and how the various loss mechanisms combine to yield a test gas state that is quite different from what is expected based on ideal conditions. For the simulation of conditions desired in this study, pressure pulse durations of the order of 1 msec were desired, and peak pressures of over 400 MN/m^2 were required. Hence, the conditions are more severe than those studied previously.[10]

Ballistic Compressor Apparatus

The ballistic compressor under consideration utilizes a reservoir of gas to drive an inertial piston which, in turn, compresses the desired test gas. A schematic diagram of the ballistic compressor apparatus is shown in Fig. 1. It consists, mainly, of a reservoir in which the test gas is initially situated, an inertial piston, and the test section, which serves as a holder for test specimens used in erosion studies and as an exhaust port for the test gas flowing through the test orifice.

The driver gas in the reservoir is allowed to flow through the transition section to the compression chamber; flow is initiated by opening an actuating valve. Initially, the actuating chamber is pressurized above the pressure of the driver gas in the reservoir, so that the actuating piston shuts the

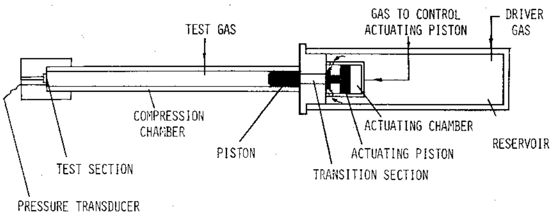

Fig. 1 Schematic diagram of the ballistic compressor apparatus.

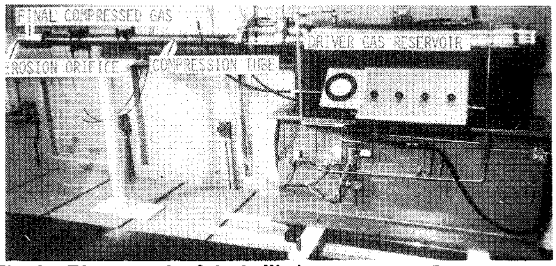

Fig. 2 Photograph of the ballistic compressor (length ~ 3 m).

passage to the compression chamber. By discharging the gas in the actuating chamber, the pressure differential thus produced drives the actuating piston back (to the right, Fig. 1) and opens the transition passage. The discharge of the gases from the actuating chamber is accomplished through a fast solenoid valve. Typical time of opening of the transition passage is 1 msec. The driver gas that flows through the transition section drives the inertial piston, which compresses the test gas in front of it.

A photograph of the ballistic compressor apparatus is shown in Fig. 2. Major dimensions of interest are: reservoir volume = 0.0137 m^3; diameter of compression chamber = 3.871 cm; length of compression chamber = 152.4 cm; diameter of transition section = 3.334 cm; length of transition section = 21.0 cm. The entire assembly is mounted on wheels and moves slightly during a firing. For the present purposes, this effect was not important.

The inertial pistons are made in the shape of simple right circular cylinders, with a central undercut section. Some of the pistons have o-rings at the forward and aft ends. The pistons with o-rings were manufactured from type-304 stainless steel, whereas the pistons without o-rings were made of phosphor-bronze, a material that has good sliding characteristics in contact with steel. The length of the pistons was 15.2 cm; typical mass of the pistons was 1.34 kg. The minimum radial clearance of the piston is of the order of 25 μm.

The pressure history of the test gas was monitored by a high frequency Kistler model 607 piezoelectric pressure transducer. Typical pressure traces are shown in Fig. 3. These traces were obtained in the case for which air was the driver, as well as the test gas. For diatomic gases, the ballistic compressor was tested up to 520 MN/m^2.

Theoretical Analysis

Figure 4 shows the three sections (reservoir, driver section, and test section) considered in the development of the governing equations describing the dynamic behavior of the ballistic compressor. The conservation of energy equations applicable to the three sections are

$$\frac{d}{dt}(m_R u_R) = h_R \frac{dm_R}{dt} \tag{1}$$

$$\frac{d}{dt}(m_D u_D) = -h_R \frac{dm_R}{dt} - \frac{1}{2}\frac{d}{dt}\left[m_D\left(\frac{dx}{dt}\right)^2\right] - P_D A_c \frac{dx}{dt} \pm q_L^D - q_H^D \tag{2}$$

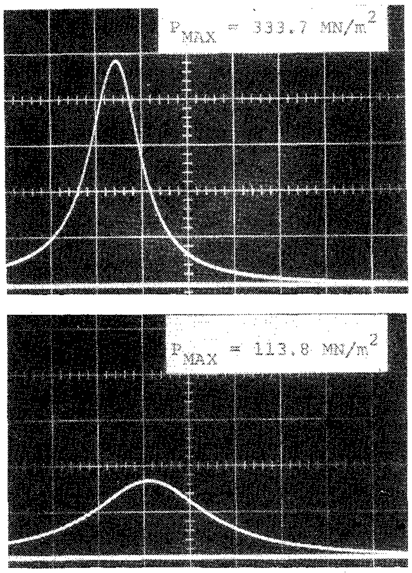

Fig. 3 Typical pressure-time traces of ballistic compressor; horizontal scale = 0.2 msec/div, vertical scale = 68.95 MN/m^2/div.

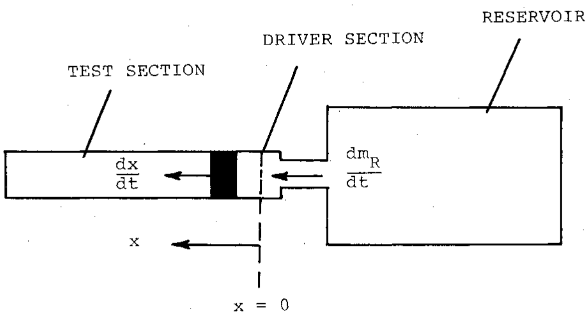

Fig. 4 Schematic diagram of the analytical system.

$$\frac{d}{dt}(m_T u_T) = P_T A_c \frac{dx}{dt} + \frac{1}{6}\frac{d}{dt}\left[m_T\left(\frac{dx}{dt}\right)^2\right] \mp q_L^T - q_H^T \tag{3}$$

where, from continuity,

$$\frac{dm_R}{dt} = -\frac{dm_D}{dt} \tag{4}$$

The definitions of the symbols are given in the Nomenclature. The energy associated with the internal leakage (piston blow-by) q_L is assigned both positive and negative signs, because internal leakage, depending on the conditions in the driver and test sections, may occur in either direction during the experimental time interval. The gas flow within the driver and test sections was assumed to be inviscid. The density of the gas in the particular section was considered to be uniform. This assumption requires that the velocity along the driver section be uniform and equal to the velocity of the piston dx/dt. This assumption appears to be justified since shock-like pressure rises were never observed in the pressure traces. In the test section, this assumption [i.e., $\rho = \rho(t)$] provides a linear velocity profile. In order to avoid space-dependent properties in the energy equation [Eq. (3)], an average kinetic energy of the test gas was considered. This average approximation results in the 1/6 multiplier in Eq. (3).

The velocity and position of the piston can be obtained from the equation of motion of the piston

$$m_p \frac{d^2x}{dt^2} = (P_D - P_T)A_c - F \tag{5}$$

where F is the friction force acting on the piston.

Assuming the flow to be quasisteady, the mass flowrate dm_R/dt is given by the compressible isentropic relation

$$\frac{dm_R}{dt} = C_D \frac{A_c P_R}{(R_D T_R)^{1/2}} \left\{ \frac{2\gamma_D}{\gamma_D - 1} \left[\left(\frac{P_D}{P_R}\right)^{2/\gamma_D} - \left(\frac{P_D}{P_R}\right)^{(\gamma_D+1)/\gamma_D}\right]\right\}^{1/2} \tag{6}$$

Equation (6) is derived based on an ideal gas equation of state. The discharge coefficient C_D was introduced in Eq. (6) to account for the valve losses and viscous losses in the transition section.

The previous equations, in conjunction with the equations of state of the test gas and driver gas, and the associated caloric equations [$u = f(T)$] constitute the governing equations of the ballistic compressor. The initial conditions of these equations are; at $t = 0$,

$$T_R = T_{IR} \quad P_R = P_{IR} \quad T_D = T_{ID} \quad P_D = P_{ID}$$

$$T_T = T_{IT} \quad P_T = P_{IT} \quad x = 0 \quad dx/dt = 0 \tag{7}$$

Ideal Performance of a Ballistic Compressor

An ideal compressor is one that allows instantaneous inflow of the driving gas (hence, no loss of impulse to the piston), has no friction between the inertial piston and the compression chamber walls, has no pressure drop across the inlet-valve assembly, exhibits no gas leakage either in the forward direction during the accelerating phase or in the backward direction during the decelerating phase of the piston motion, and has zero heat transfer between the compressed hot gas and the cylinder or piston walls.

A comparison between the actual pressure attained by the ballistic compressor and its ideal theory prediction is given in Fig. 5. The deviation of the experimental results from ideal predictions increases as the driver pressure increases. At $P_{ID} = 1.72$ MN/m^2, the pressure difference is 41.4 MN/m^2 (37% pressure loss) whereas, at $P_{ID} = 2.75$ MN/m^2, the pressure difference is 138 MN/m^2 (32% pressure loss). As mentioned earlier, the factors believed to be responsible for the loss in performance of the ballistic compressor are: nonideal behavior of the driver and test gases, valve losses, internal and external leakage, friction of the piston, shock-wave generation, and heat losses.

In the next section, the various factors that influence the real performance of the ballistic compressor will be examined. The analysis of the loss mechanisms involved examination of various equations of state of the test gas, experimental determination of the valve losses, measurement of internal and external leakage, experimental assessment of the effect of friction of the piston, and an assessment of the effect of shock-wave generation in the test gas on the performance of the ballistic compressor. In addition, the magnitude of the heat loss to the tube walls from the test gas has been estimated.

Factors Responsible for the Nonideal Behavior of the Ballistic Compressor

Equation of State

To be able to predict the performance of the ballistic compressor (i.e., the final state of the compressed test gas), the equation of state of the gas must be known throughout the compression sequence. In the present work, four of the well-known equations of state have been considered in the numerical calculations. The objective was to evaluate the influence of the assumed equation of the state of gases on the performance of the ballistic compressor. Because of the wide range of pressure and temperature experienced by the test gas undergoing compression, none of the considered equations of state are expected to describe correctly these large changes of state. However, this parametric study may provide information regarding the analytical errors occurring because an inaccurate equation of state was used. The equation of state considered, along with the associated caloric equations, are:

Perfect gas equation

$$pV = mRT \tag{8}$$

$$u = u^{\circ}(T) \tag{9}$$

Abel's equation

$$p(V-b) = mRT \tag{10}$$

$$u = u^{\circ}(T) \tag{11}$$

Van-der Walls equation

$$\left(p + \frac{a}{V^2}\right)(V-b) = mRT \tag{12}$$

$$u = u^{\circ}(T) - \frac{a}{V} \tag{13}$$

Beattie-Bridgemann equation

$$pV^2 = RT\left(1 - \frac{C}{VT^3}\right)\left(V + B_0 + \frac{bB_0}{V}\right) - A_0\left[1 - \frac{a}{V}\right] \tag{14}$$

$$u = u^{\circ}(T) - \left(A_0 + \frac{3Rc}{T^2}\right)\frac{1}{V} - \frac{1}{2V^2}\left(-aA_0 + \frac{3RcB_0}{T^2}\right) + \frac{RcbB_0}{T^2V^3} \tag{15}$$

Values of the constants of the preceding equations for the case of nitrogen as the test gas are shown in Table 1. Although these constants can be adjusted to vary with temperature, as was done by Takeo,[10] in this application, we sought to evaluate the usefulness of the equations while holding the constants at fixed values for the entire range of conditions. A comparison of the results obtained by use of the four previous equations for two values of the reservoir pressure of 2.07 MN/m^2 and 2.76 MN/m^2 are shown in Table 2. The variations in the predicted values of peak pressure and temperature increases as the reservoir pressure increases. The results for P_{max} and T_{max}, obtained by using the perfect gas equations and Abel's equation, correspond closely, although the minimum volumes differ significantly as may be expected. Abel's equation, which is really a special case of Van-der Waals equation for which the cohesion term is neglected due to high pressures, is used extensively in internal ballistic problems, in which correlation of the experimental results with theory has been found to be surprisingly good to temperatures of the order of 3000°K.[11] The largest variations in theoretical predictions in this study were found in the case of the Beattie-Bridgemann equation. The application of this equation has been shown[11] to result in gross error in evaluating the internal energy of CO_2 at $T = 150°C$ and $p = 350\text{-}470$ MN/m^2. Comparison of the perfect gas equation of state with the Van-der Waals equation shows that, at 2.76 MN/m^2 reservoir pressure the percentage difference in predicting peak pressure and temperature are 7 and 2%, respectively. These differences decrease with decrease in the values of the final and temperature.

Table 1 State equation constants for nitrogen[11]

Abel's equation:	
b	$=0.03913$ liters/mole
Van-der Waal's equation:	
a	$=1.390$ liters-atm/(mole)^2
b	$=0.03913$ liters/mole
Beattie-Bridgemann equation:	
A_0	$=1.3445$ (liters)^2 atm/(mole)^2
a	$=0.02617$ liters/mole
B_0	$=0.05046$ liters/mole
b	$=0.00691$ liters/mole
c	$=4.20\times10^4$ liters-K/mole

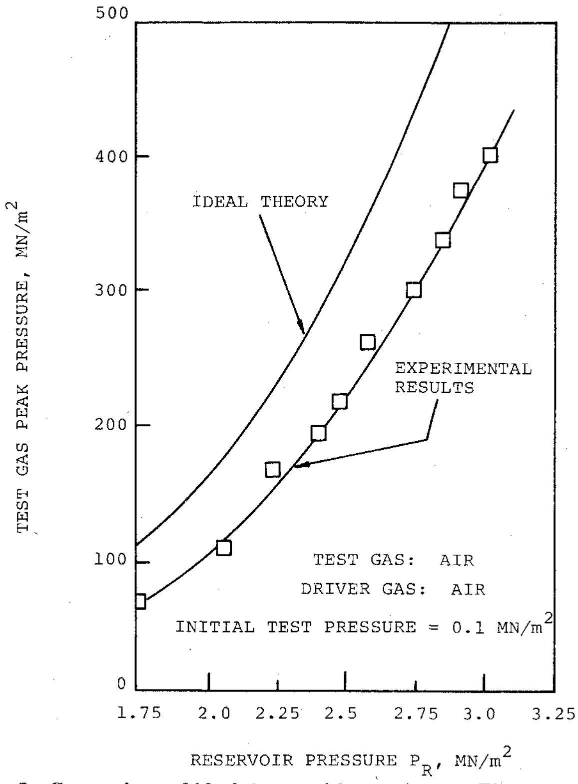

Fig. 5 Comparison of ideal theory with experiments. Theory greatly overestimates the peak pressures obtained in the ballistic compressor.

Table 2 The effect of equation of state of the test gas on the performance predictions of the ballistic compressor[a]

	Initial reservoir pressure $p_R = 2.07$ MN/m^2			Initial reservoir pressure $p_R = 2.76$ MN/m^2		
Equation of state of test gas	p_{max}, MN/m^2	T_{max}, °K	V_{min}, cm^3	p_{max}, MN/m^2	T_{max}, °K	V_{min}, cm^3
Ideal	174.11	2472	7.826	428.35	3187	4.103
Abel's equation	173.60	2469	10.430	427.02	3183	6.702
Van-der Waal's	181.58	2436	9.804	457.61	3132	6.273
Beattie-Bridgeman	165.86	2122	10.559	388.65	2654	6.692

[a]For these tests, $P_{IT} = 1$ atm.

Valve Losses

The gas flow from the reservoir through the transition section and into the compression tube behind the piston experiences a drop in total pressure because of nonideal flow through this so-called valve. This loss is accounted for in the theoretical analysis by the insertion of the discharge coefficient C_D in Eq. (6). In order to obtain C_D experimentally, pressure transducers were mounted in the compression chamber and, during tests, the pressure of the driver gas at $x = 30.5$ cm distance from the transition section was monitored. Solution of Eq. (6), together with the conservation equations to obtain pressures matching those measured, resulted in a value of $C_D = 0.5$.

Figure 6 shows a comparison of the performance curves of the ballistic compressor using $C_D = 1.0$ and $C_D = 0.5$. The curves were obtained using the ideal gas equation of state. At a reservoir pressure of 2.76 MN/m^2, the deviations obtained in peak pressure and peak temperature using $C_D = 1.0$ vs $C_D = 0.5$ are 12.5 and 2.5%, respectively.

Internal Leakage

Following Takeo et al.,[10] the leakage gas flow through the piston gap has been assumed to be incompressible, one-dimensional Couette flow. The velocity of the gas is given by

$$v = V_p\left(1 - \frac{y}{h}\right) - \frac{h^2}{2\mu}\frac{dP}{dx}\frac{y}{h}\left(1 - \frac{y}{h}\right) \tag{16}$$

where h is the radial gap between the piston and the internal radius of the compression tube. At the surface of the piston ($y = 0$), the velocity of the leakage gas is equal to the velocity of the piston V_p, whereas at the bore surface of the compression tube, the gaseous velocity is equal to zero. Equation (16) was derived for a steady-state flow. In the present unsteady calculation, the invoked quasisteady approximation allows the use of Eq. (16) if the pressure differential dP/dx is used as a function of time. The leakage rate dm_L/dt then is given by

$$\frac{dm_L}{dt} = 2\pi r_p \rho \int_0^h (v - V_p)\,dy \tag{17}$$

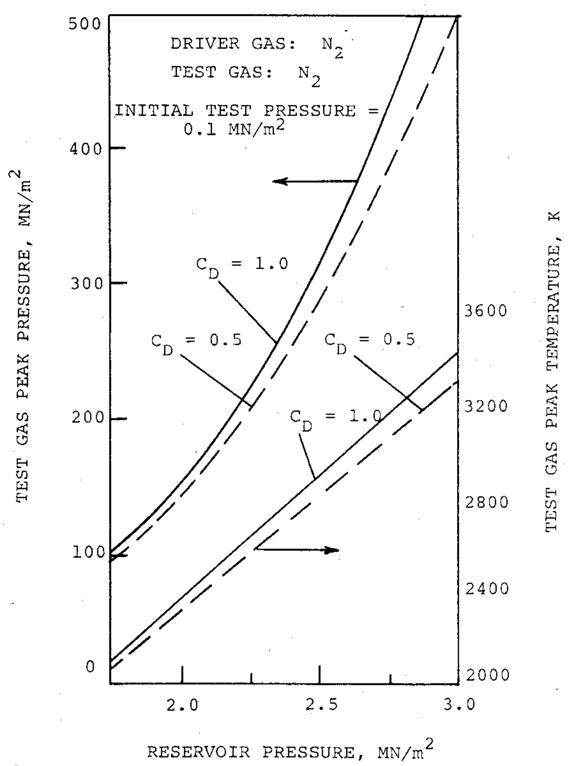

Fig. 6 Effect of valve losses (C_D) on the performance of the ballistic compressor.

since the flow must pass the piston moving at velocity V_p. Combining Eqs. (17) and (16), and integrating, yields

$$\frac{dm_L}{dt} = -\pi r_p \rho h\left[V_p + \frac{h^2}{6\mu}\frac{dP}{dx}\right] \tag{18}$$

Depending on the conditions across the piston and the piston velocity, the leakage rate can be either positive or negative, i.e., during the compression cycle there is a reversal of leakage. Initially, when V_p is positive but small and dP/dx is negative, leakage occurs from the driver section to the test section. The transfer of energy associated with the leakage is [see Eqs. (2) and (3)]

$$q_L^D = -\frac{dm_L^D}{dt}h_D \tag{19}$$

and

$$q_L^T = +\frac{dm_L^D}{dt}h_D$$

Near the end of the compression stroke, when dP/dx is positive, or earlier in the stroke when the following condition is met

$$\left|\frac{dP}{dx}\right| < \left|\frac{6\mu V_p}{h^2}\right|$$

the leakage is negative, i.e., leakage occurs from the test gas to the driver gas. In this case

$$q_L^T = -\frac{dm_L^T}{dt}h_T$$

$$q_L^D = \frac{dm_L^T}{dt}h_T \tag{20}$$

To experimentally determine the leakage, tests were performed using CO_2 as a driver gas and air as a test gas. The test gas was allowed to outflow through the port in the test section, and was collected in a special teflon bag. The mass fraction of CO_2 in the test gas was obtained by means of gas chromatography. Table 3 shows results of the previously described tests for several initial reservoir pressures. The highest resultant leakage was less than 4% at 380 MN/m^2 peak pressure. This method gives the resultant leakage, but one may expect that leakage may have an instantaneous maximum during the compression cycle. The low resultant amount of leakage obtained by this method has been substantiated by estimations of the mass of the test gas by measuring the pressure of the test gas and the final resting place of the piston relative to the test end, i.e., when the piston is in static equilibrium. These estimations gave resultant leakage to be of the order of 4%. For the CO_2-air system, agreement of the numerical results with experiments were obtained by adjusting the value of the effective clearance gap h. Then the succeeding calculations for any driver

Table 3 Experimental determination of "blow-by" leakage of the ballistic compressor using gas chromatography[a]

Reservoir pressure MN/m^2	Maximum test gas pressure, MN/m^2	Mass fraction of CO_2 in test gas
2.07	173.7	0.017
2.07	173.7	0.017
2.31	239.9	0.027
2.76	380.6	0.037

[a]Driver gas: CO_2; test gas: air.

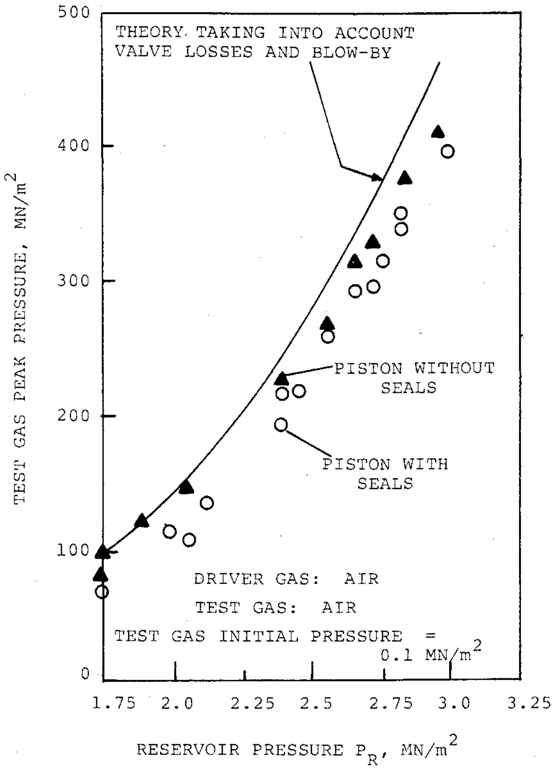

Fig. 7 Comparison of the performance of the ballistic compressor using pistons with and without o-rings. Friction of o-rings reduces the attainable peak pressure.

gas/test gas system have been performed by use of of the derived value of h, since this value should not vary with test conditions when the same compression tube-piston combination were used. The derived radial clearance of the piston h was 21 μm, which is in close agreement with its design value of 25 μm. Although the clearance was assumed to be invariant with test conditions, the blow-by would be strongly dependent on pressures and velocities as given by Eq. (18).

External Leakage

By use of the same CO_2/air system, the external leakage was determined by enclosing the test section with a plastic bag. With no outlet in the test section, tests were performed and the samples from the bag were examined by means of gas chromatography. External leakage was found to be negligible.

Friction of the Piston

Assuming that the friction force acting on the piston is only due to the shear forces of the Couette flow, the friction force is

$$F = -2\pi r_p L_p \mu \frac{dv}{dy}\bigg|_{y=0} \tag{21}$$

Using Eq. (16), Eq. (21) becomes

$$F = \pi r_p L_p \left| \frac{dP}{dx} + \frac{2\mu}{h^2} V_p \right|$$

In all calculations, μ was allowed to vary with temperature according to the Sutherland approximation.[12] Calculations performed using Eq. (22) show that the effect of frictional forces derived from the blow-by of gases is negligible. This is in contradiction to the results obtained by Takeo et al.[10] They stated that friction due to the shearing forces of the blow-by gases is appreciable. Our conclusion can be substantiated by an order-of-magnitude calculation. However, no isolated tests were made to show the effect of friction on the performance of the ballistic compressor. In the previous approach, the friction forces acting on the o-rings of the piston were neglected. Tests performed using pistons with and without o-rings show that o-ring friction is not negligible. Figure 7 shows these results. For a given reservoir pressure, the pressure attained with the ballistic compressor without o-rings is consistently higher than the piston with o-rings. This is especially evident at lower peak pressure. Comparison of the results of the piston without o-rings with the analytical results show that at low peak pressure the agreement is good, but it deteriorates as the peak pressure is increased. This disagreement at high peak pressure may result from the high leakage rates obtained when no o-rings are used on the piston.

Generation of Shock Waves

Shock formation by a piston moving with a finite speed in a closed-end tube have been studied analytically by Evans and Evans,[13] Winter,[14] and Minardi and Schwartz.[15] Minardi and Schwartz found that, below a critical value of the mass of the piston, the pressure generated by the compression process depends on the mass of the piston. Above this critical value the peak pressure is independent of the mass. This has been demonstrated experimentally in the ballistic compressor by Hammond and Lalos.[3] They found that, when the mass of the piston is above 1 kg, in the case of argon as the test gas, the peak pressure generated is independent of the mass of the piston.

Tests performed in the Princeton University ballistic compressor with pistons of 1.34 and 2.6 kg mass show that both pistons generate the same peak pressures when the initial conditions are the same. Thus, for the present results, shock formations due to the motion of the inertial piston are believed to be negligible. Also, no sudden pressure rises were noted in the pressure traces at any time, suggesting that no shock waves were formed.

Heat Losses

To estimate the influence of heat losses on the performance of the ballistic compressor, a comparison of the magnitude of the heat loss relative to the magnitude of the work done by the driver gas on the test gas was made. The rapid rise of the temperature of the gas occurs during the end of the deceleration period of the piston, at which time the velocity of the piston, and correspondingly the induced velocity of the test gas, is small. Consequently, the primary modes of heat transfer at the gas-solid interface will be radiation and conduction.

The influence of radiative losses on the performance of the ballistic compressor is given by

$$L_R = \frac{Q_R}{W_D} \simeq \frac{\sigma A_c (T_T^4 - T_W^4) t}{P_D A_c L_c}$$

where Q_R is the heat loss due to radiation during the effective test cycle and W_D is the total work done by the driver gas. The test gas is assumed to be an ideal emitter of radiation.

A measure of the relative importance of heat loss due to conduction is given by

$$L_C = \frac{Q_C}{W_D} = \frac{4KA_c (T_T - T_W)(t/\pi\alpha)^{1/2}}{P_D A_c L_c}$$

where Q_C is the heat transfer to the end-wall and to the front surface of the piston. The temperature of the walls enclosing the test gas was assumed to be constant. The derivation of the expression for Q_C is given in Appendix A.

In order to obtain an upper bound of the relative magnitude of the heat losses, a simple case was considered. Estimates of heat loss were made based on the assumption that, during the effective test cycle, the temperature and pressure of the test gas were equal to the corresponding maximum conditions ($T_{MAX} = 3190°$K, $P_{MAX} = 428$ MN/m^2). The values of L_R

and L_C for the case considered are 0.001 and 0.011, respectively. That is, assuming the most extreme conditions throughout, the heat losses experienced by the test gas are only about 1% of the total energy available to the test gas. Since it is so small based on this extreme assumption, closer computations based on time-wise conditions seems to be unjustified for the present purposes.

Discussion and Conclusions

The estimation of the final conditions of the test gas undergoing rapid compression in the ballistic compressor apparatus, assuming ideal behavior of the ballistic compressor, can be grossly in error. Typically, the peak pressure of a diatomic gas at 300 MN/m^2 is overestimated by as much as 30%.

The factors that are responsible for the nonideal compression of a test gas in the ballistic compressor have been considered. It has been shown that, in order to predict the true conditions of the compressed gas, the valve losses and the friction of the o-rings must be included in the analysis. By use of pistons without o-rings, at test gas pressure levels of up to 140 MN/m^2, it can be shown that the predictions made by the theory, which take into account valve losses and blow-by, agree with the experimental results. Above this pressure level ($\sim$140 MN/m^2), excessive leakage further reduces the performance of the ballistic compressor.

The variation of predicted pressure and temperature is found to be less critically dependent on the equation of state used. Differences resulting from using the ideal gas equation of state rather than the Van-der Waals equation are of the order of 7% for the prediction of peak pressure and 2% for the prediction of the peak temperature. The heat loss experienced by the test gas undergoing compression was estimated to be less than 1% of the total energy imparted to the test gas.

Appendix A

To estimate the conductive heat loss in the test section, the test gas is assumed to be a one-dimensional slab of thickness $2b$, at a uniform initial temperature T_F. At $t=0$ the boundaries at $x=\pm b$ are assumed to be at temperature T_W. The solution of this problem is given by Carslaw and Jaeger[16]

$$T-T_W=\frac{4(T_F-T_W)}{\pi}\sum_{n=0}^{\infty}\frac{(-1)^n}{(2n+1)}\times\exp\left[\frac{-\alpha(2n+1)^2\pi^2 t}{4b^2}\right]\cos\frac{(2n+1)\pi x}{2b} \tag{A1}$$

The heat flux at the surface $x=+b$ is

$$q(t)=-K\frac{\partial T}{\partial x}\Big|_{x=b}=\frac{k(T_F-T_W)}{(\pi\alpha t)^{1/2}}\left[1+2\sum_{n=1}^{\infty}(-1)^n\exp\left(\frac{-n^2b^2}{\alpha t}\right)\right] \tag{A2}$$

For the conditions considered, $b^2/\alpha t$ is of the order 10^6, therefore, as a first approximation,

$$q(t)=\frac{K(T_F-T_W)}{(\pi\alpha t)^{1/2}} \tag{A3}$$

The total conductive heat loss is given by

$$Q_C=\int 2q(t)\mathrm{d}t=4K\left(T_F-T_W\right)\left(\frac{t}{\pi\alpha}\right)^{1/2} \tag{A4}$$

where the factor 2 accounts for heat losses at the surfaces $x=\pm b$.

References

[1]Price, D. and Lalos, G. T., "Adiabatic Compressor for P-V-T Measurements on Gases to 100,000 Pounds per Square Inch," *Industrial and Engineering Chemistry*, Vol. 49, Dec. 1957, pp. 1987-1992.

[2]Lalos, G. T., "Laboratory Production of Hot Dense Gases," *The Review of Scientific Instruments*, Vol. 33, Feb. 1962, pp. 214-218.

[3]Hammond, G. L. and Lalos, G. T., "The NOL Ballistic Piston Compressor II Operation up to 5,000 Atm," Naval Ordnance Laboratory, White Oak, Md., Rept. NOLTR 71-228, 1971.

[4]Hammond, G. L., "The Broadening of Calcium II H and K Lines by Helium," *Astrophysical Journal*, Vol. 196, 1975, pp. 291-305.

[5]Longwell, P. A., Reamer, H. H., Wilburn, N. P., and Sage, B. H., "Ballistic Piston for Investigating Gas Phase Reactions," *Industrial and Engineering Chemistry*, Vol. 50, April 1958, pp. 603-610.

[6]Ryabinin, Y. N., *Gases at High Densities and Temperatures*, Pergamon Press, New York, 1961.

[7]Minardi, J. E., "Comparison of Transient Compressors for the Production of High Temperature and High Density Gases," Aerospace Research Laboratories Rept. ARL 65-181, 1965.

[8]Lalos, G. T. and Hammond, G. L., "The Ballistic Compression and High Temperature Properties of Dense Gases," *Experimental Thermodynamics of Non-Reacting Fluids*, 1975, Chap. 25, Butterworths & Co. Ltd., London, pp. 1193-1218.

[9]Lalos, G. T. and Hammond, G. L., "Gas Opacity Measurements with a Ballistic Piston Compressor," Naval Ordnance Laboratory, White Oak, Md., Rept. NOLTR 70-15, 1969.

[10]Takeo, M., Holmes, Q. A., and Ch'en, S. A., "Thermodynamic Conditions of the Test Gas in a Ballistic Compressor," *Journal of Applied Physics*, Vol. 38, Aug. 1967, pp. 3544-3550.

[11]Hirschfelder, J. O., Curtiss, C. F., and Bird, R. R., *Molecular Theory of Gases and Liquids*, Wiley, New York, 1954, pp. 250-258.

[12]Schlichting, H., *Boundary Layer Theory*, 4th ed., McGraw Hill, New York, 1960, p. 339.

[13]Evans, C. and Evans, F., "Shock Compression of a Perfect Gas," *Journal of Fluid Mechanics*, Vol. 1, 1956, pp. 339-408.

[14]Winter, D. F. T., "Multiple Shock Compression Using a Piston of Finite Weight," *Journal of Fluid Mechanics*, Vol. 8, 1960.

[15]Minardi, J. and Schwartz, R., "Investigations of Ballistic-Piston and Liquid Driven Piston Compressor for the Production of High Temperature and High Density Gases," Aerospace Research Laboratories, Wright Patterson AFB, Ohio, ARL 63-167.

[16]Carslaw, H. S. and Jaeger, J. C., *Conduction of Heat in Solids*, Clarendon Press, Oxford, 1959, pp. 92-99.