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Design and Construction of the X-2 Two-Stage Free Piston Driven Expansion Tube

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1 Introduction

This report outlines the design and construction of the X-2 two-stage free piston driven expansion tube. The project has completed its construction phase and the facility has been installed in the new impulsive research laboratory where commissioning is about to take place. The X-2 uses a unique, two-stage driver design which allows a more compact and lower overall cost free piston compressor. The new facility has been constructed in order to examine the performance envelope of the two-stage driver and how well it couples to sub-orbital and super-orbital expansion tubes. Data obtained from these experiments will be used for the design of a much larger facility, X-3 utilizing the same free piston driver concept.

1.1 Two-Stage Driver Concept

When designing the high pressure diaphragm station for free piston shock tubes, it is quickly realised that a substantial portion of the facility cost can be removed by adopting a constant area between the driver and the driven tubes. However, if using a conventional single-bore free piston driver, in order to maintain an adequate supply of driver gas after diaphragm burst the overall facility length becomes impractical. By splitting the driver gas compression process into two parts, a reasonable quantity of driver gas can be supplied with a much reduced compression tube length (approximately a factor of four).

Examining the pressure history that would be expected during any free-piston driver operation cycle (figure 1), it can be seen that most of the compression process occurs with the pressure level remaining below 2% of the final pressure achieved. During this low pressure phase, about 85% of the available reservoir work is converted into piston kinetic energy, the remainder being stored in both the reservoir and driver gases. Therefore, by increasing the bore of the compression tube over the low pressure phase, the volumetric displacement required to accelerate the piston to the required speed can be achieved in a reduced length.

A schematic of the two stage driver concept (first suggested by Morgan[1]) is shown in figure 2. The concept requires a compound piston arrangement consisting of a light aluminium outer and heavy stainless steel inner. At the termination of the large bore first stage, the

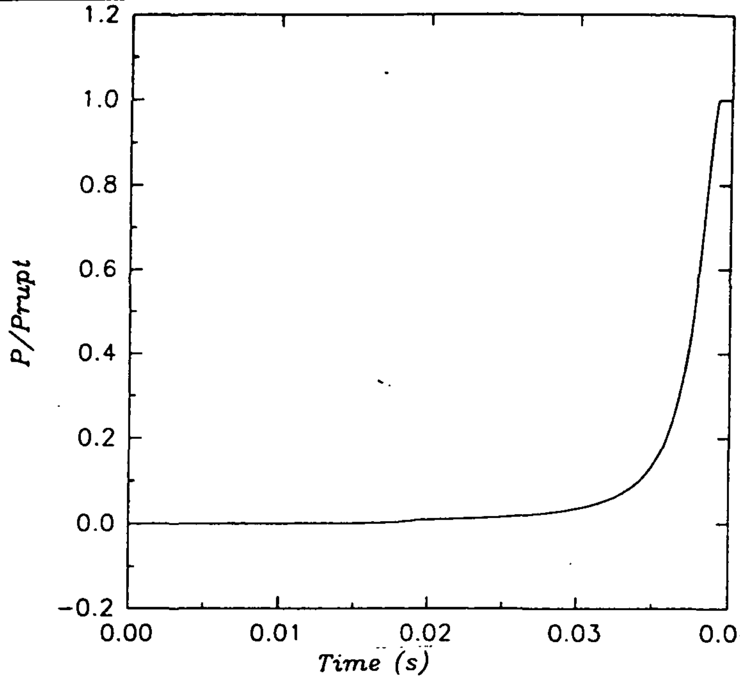

Figure 1: Theoretical pressure history of compression process in a free piston driver

pistons separate. The outer piston is stopped by the use of a buffer and the inner piston (which contains most of the kinetic energy) is allowed to compress the driver gas to high pressure in the small bore second stage.

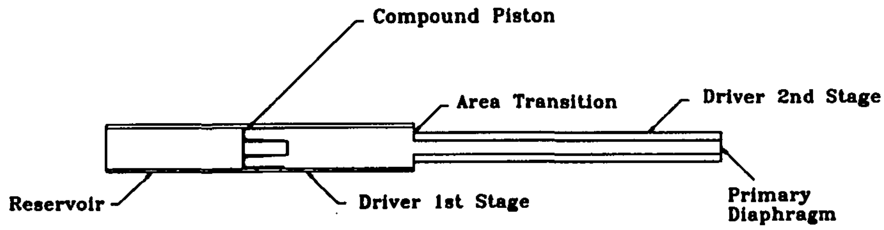

Figure 2: Two Stage Free-Piston Driver Concept

1.2 Basic Theory of Two-Stage Free Piston Drivers

Preliminary operating performance and benefits of two-stage free piston drivers over conventional single stage machines can be seen by considering some elementary gas dynamic theory. Perhaps the two most important criteria for assessing the two-stage driver are size (length) of the final gas slug and its temperature before diaphragm burst. The temperature can be achieved by using a sufficiently high compression ratio, while the final gas slug length depends upon the geomtery of the facility. The designer must be careful in selecting the geometry of the driver as it influences safety as well as operational issues.

The overall compression ratio (λ) for any free piston device can be written as

$$\lambda = \frac{V_t}{V_f} \tag{1}$$

where V_t is the total driver volume and V_f is the volume of the compressed driver gas just prior to rupture. For two-stage drivers this can be rearranged as,

$$\begin{aligned} \lambda &= \frac{V_1 + V_2}{V_f} \\ \lambda &= \left(\frac{V_1 + V_2}{V_2}\right)\left(\frac{V_2}{V_f}\right) \\ \lambda &= \lambda_1 \lambda_2 \end{aligned} \tag{2}$$

where V_1 and V_2 are the volumes of the driver first and second stages respectively. λ_1 can be considered the first stage compression ratio and λ_2 is the second stage compression ratio.

The first stage compression ratio can be manipulated to yield,

$$\begin{aligned} \lambda_1 &= \frac{V_1}{V_2} + 1 \\ \lambda_1 &= A_R L_R + 1. \end{aligned} \tag{3}$$

A_R and L_R are the area and length ratios between the first and second stages respectively. Using equations 1, 2 and 3 gives the ratio of final compressed driver gas lengths for the two and single stage drivers respectively in terms of driver geometry,

$$\frac{x_T}{x_S} = \frac{A_R \lambda_1}{A_R + \lambda_1 - 1} \tag{4}$$

Leaving equation 4 for the moment, we will consider the pressures across the compound piston during the compression in the first stage. If the pressure ratio of the driver gas to the reservoir gas becomes greater than unity, a potentially dangerous situation exists. The more massive inner piston contains more inertia than the aluminium outer therefore if an acceleration reversal occurs, the pistons may separate prematurely, causing damage to the facility. The geometry and operating conditions can be chosen so as to minimise this risk. By assuming an ideal description of the gas dynamics,

$$PV^{\gamma} = const. \tag{5}$$

(where P is the gas pressure, V is the volume and γ is the ratio of specific heats) and using the expressions derived above, the following formulae can be described:

$$\frac{P_{df}}{P_{r*}} = \alpha\lambda\left(\frac{\gamma_d - 1}{\gamma_r - 1}\right)\left(\frac{1-(1+1/\alpha+1/\alpha\lambda_1)^{1-\gamma_r}}{1-\lambda^{1-\gamma_d}}\right) \tag{6}$$

$$\frac{P_d}{P_r} = \frac{P_{df}}{P_{r*}}(\lambda_1/\lambda)^{\gamma_d}(1+1/\alpha-1/\alpha\lambda_1)^{\gamma_r} \tag{7}$$

Here, $\frac{P_{df}}{P_{r*}}$ is the ratio of final driver pressure to initial reservoir pressure, α is the ratio of volumes between the reservoir and the driver and γ_r and γ_d are the ratio of specific heats for the reservoir and and the driver gas respectively. $\frac{P_d}{P_r}$ is the pressure ratio across the piston after the stage one compression process.

Equations 4, 6 and 7 can be used to provide an estimate of facility performance with respect to a particular chosen geometry (i.e. α, A_R and λ_1). Figure 3 shows the increase in slug length available over single stage machines for various area ratios (A_R) and stage one compression ratios (λ_1). Figure 4 shows the variation of the compound piston pressure ratio (P_d/P_r) with λ_1 and compression ratio (λ). These results illustrate that in order to keep P_d/P_r below unity over a large range of compression ratios, λ_1 must be kept below 6. Also, by limiting the area ratio to 9 (to keep the first stage bore reasonably small) 3.5 to 4 times the slug length of compressed gas generated in a single stage machine can be used if λ_1 is kept between 5 and 6. Alternatively, this ratio can be thought of as the decrease in length a two stage driver offers over conventional drivers for the same amount of driver gas, at the same condtions before diaphragm rupture. For these calculations, α was assumed equal to 2.9.

Although this analysis is simple, it allows a convenient way of sizing the components for X-2 and for showing the benefits of the two-stage free piston driver for a constant area diaphragm station facility.

2 X-2 Facility Construction

This section will outline the major features of the X-2 two-stage expansion tube. The primary aim of the X-2 project is to determine the capability of a two-stage driver with respect to

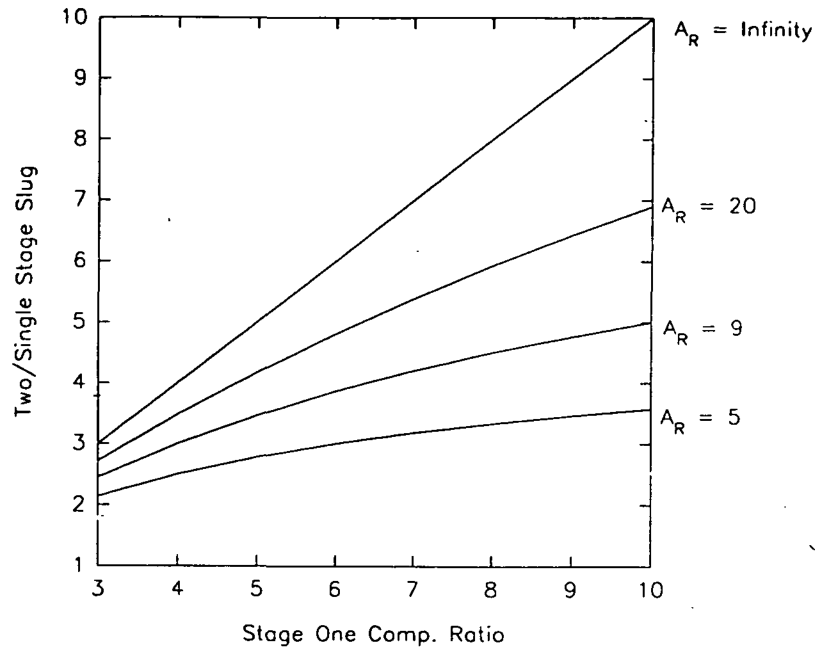

Figure 3: Increase in slug length of two/single stage free piston drivers

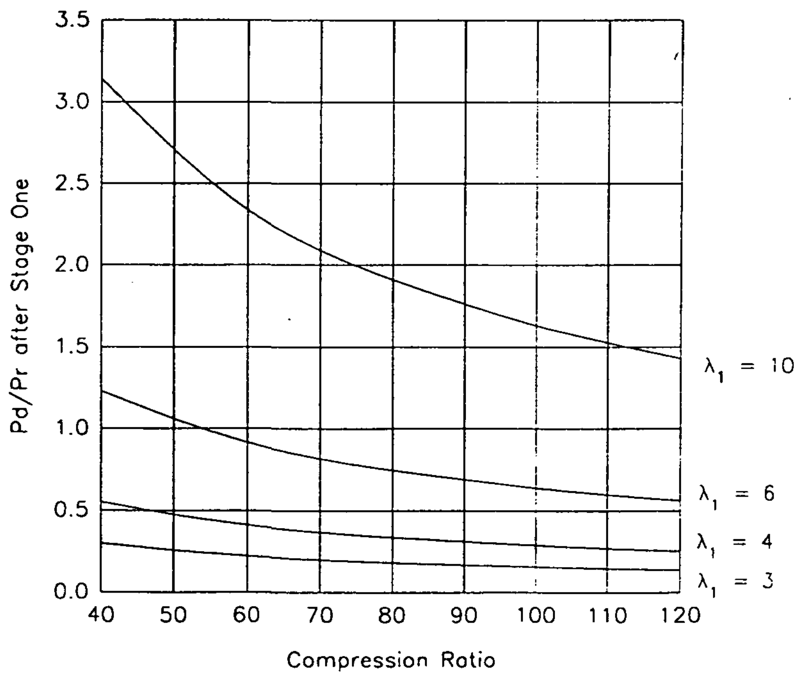

Figure 4: Pressure ratio accross compound piston after stage one

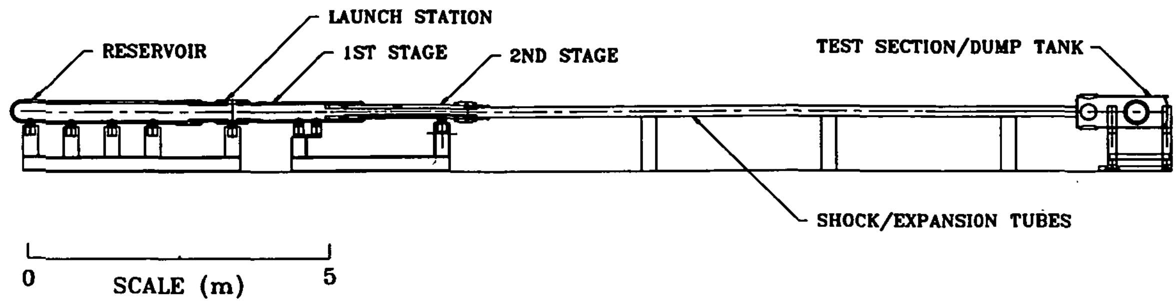

Figure 5: X-2 Facility Layout

the operation of expansion tubes. In order to achieve reasonable test times, the driver must leave an adequate volume of driver behind the diaphragm proir to burst. In a constant area driver, this equates to the distance between the front piston face and the diaphragm. If this distance is too small, unsteady expansion waves reflect off the piston too early and cause premature termination of the test flow.

Figure 5 shows the layout of the X-2 facility. The driver uses a large reservoir to accelerate the piston. The reservoir volume is 0.23 m^3 and is approximately three times the volume of the driver itself. The reservoir is shown in figure 6.

The compound piston is required to be accelerated to a high speed (50-100 m/s) within the first stage (1.1 metres of compression allowed) and because the first stage has a large bore (273 millimetres), a piston launch mechanism is required which provides little flow resistance to the reservoir gas. For this reason a double diaphragm arrangement is used to launch the piston. The launch station is shown in figure 7. The operating principle is relatively simple and well known: the space created between the two diaphragms is filled to half the diaphragm burst pressure. The reservoir is now filled to the full burst presssure. Piston launch is initiated by the venting of the space between the diaphragm to atmosphere, resulting in diaphragm burst.

The first stage compresson tube provides the bulk of the compression process, which is done at low pressure (up to 2 MPa). The outer piston is stopped at the end of the first stage by the use of a large rubber buffer (fig. 8), bonded to a mounting plate attached to the end wall of the second stage. The size of the buffer depends on the amount of energy required to be shed from the outer piston at impact. The current buffer has been designed to withstand a 25 KJ impact which equates to a 6 kg piston travelling at 90 m/s. The size of the buffer also affects the first stage compression ratio (λ_1). As mentioned previously, the overall performance is sensitive to this ratio and must be kept large enough for an adequate volume of compressed driver gas. The current buffer allows $\lambda_1 = 5.24$. The first stage compression tube is shown in figure 9.

After the pistons separate, the inner piston travels down the second stage compression tube, where it compresses the driver gas to its final conditions and brings itself to rest before diaphragm burst. The second stage is of a multi-wall design, with an 18 mm thick sleeve interference fitted via liquid nitrogen cooling along the bore. The inner bore is 91 mm and the outer diameter is 225 mm. In order to compensate for any misallignment of the first and second stages, a tapered teflon conical entrance is provided for the second stage, allowing the inner piston some protection as it traverses the area transition. Figure 10 shows the second stage compression tube.

A sectioned view of the primary diaphragm station in shown in figure 11. The station can accomodate up to a 5 mm stainless steel diaphragm and is designed to withstand a 100 MPa impulsive load. The diaphragm is held in place with the use of a free-piston clamp. This clamp transmits an initial pre-load to the diaphragm from the tourque developed by tightening the capstan nut. As the pressure increases in front of the piston during operation, a small space allows this pressure to work on a large area of the free-piston clamp, making the axial load on the diaphragm directly proportional to the burst pressure. This ensures a large clamping load on the diaphragm and minimises the risk of slippage during operation.

The two-stage driver will be used to generate flow in 10 metres of shock tubes with an eightyfive millimetre nominal bore. These tubes are machined into seven lengths varying in length from 2.5 metres to 500 millimetres. This array of shock tubes allows study into the interactions of waves, boundary layers and multiple interfaces and their cumulative effect on expansion tube test-time. These tubes are constructed from the barrel of an ex-World War Two 17 pound anti-tank gun. Figure 12 displays the shock tubes installed in the laboratory.

The test section/dump tank has been constructed from the reservoir of an ex-submarine torpedo launcher (figure 13). It has dimensions of 500 mm bore and is 1500 mm long.

In order to achieve the very low pressures required for super-orbital testing (fill pressures approximately 1 Pa), a 13 inch Edwards oil diffusion pump has been fitted to a large flange welded to the dump tunk wall. Four access ports 90 degrees apart are provided at the test section end for instrumentation and optical access.

3 Compound Piston

The compound piston consists of an aluminium outer piston ($\sim$ 5kg) and an AISI 316 stainless steel inner piston ($\sim$ 14 kg). A sketch of the compound piston design is shown in figure 14. The light outer piston uses carbon/graphite filled teflon wear rings as bearing surfaces. A tangible link connection exists at the rear. This is used as a safety device to prevent premature separation of the pistons in the first stage. The tangible link is basically a disposable shaft of steel rated to withstand the maximum reversing force which can be calculated for a given fill pressure using the theory discussed previously. This unsafe condition may occur if the piston is accidentally pre-launched during reservoir fill, or the conditions are improperly set. The reversing force is an order of magnitude below that experienced during normal separation at the area transition so the tangible link has negligible effect on the piston dynamics.

The heavy inner piston also uses filled teflon wear rings. It uses a high pressure chevron seal similar in design to the T4 and T5 pistons. Also shown are provision for piston brakes. It is important that the inner piston is held in position after final conditions are achieved. If brakes were not fitted, and the diaphragm failed to open (or running in blanked off mode), the piston would return to the outer piston with the velocity it left (50-100 m/s). This would cause regular destruction of the outer piston (or worse). In the unlikely eventuality that the brakes did fail, the outer and inner pistons have been designed with matching tapered surfaces at their rear. The energy of impact could be then disspated through a combination of sliding friction and plastic deformation at this tapered interface.

4 Preliminary Commissioning Results and Future Work

The commissioning of the X-2 facility entails a study of the performance envelope of the free piston driver and how well it couples to an expansion tube. Driver studies have commenced with experiments involving a special 17.5 kg steel outer piston only. These tests have qualified the energy absorbing capacity of the rubber buffer located at the end of the first stage. A quasi-one-dimensional Lagrangian code[6] has been used to simulate the operating cycle of X-2. The transient performance of the rubber buffer has been modelled using a simple stress wave/lumped piston mass model. This model assumes the stress in the rubber is proportional to the velocity of the piston, which is in turn decelerated by that stress acting over a small time step.

Figure 15 compares the experimental pressure trace at the 'blanked off' diaphragm station to the quasi-one-dimensional numerical simulation. Agreement is good with both traces showing similar pressure rises and peak pressures matching to within 6%.

There was some uncertainty regarding the impact performance of the rubber buffer prior to the commencement of tests. The rubber has shown to be an excellent impact medium showing no signs of damage. The buffer has been impacted with a maximum energy of 11.5 kJ and it is planned to take it to the full 25 kJ required for 100 MPa diaphragm burst.

Also, different driver gases will be used to study the flow effects of heavier gases (such as argon) through the area transition and their influence on piston dynamics and final conditions.

Tests involving dual pistons driving a shock tube will begin soon, followed by expansion tube tests. An estimate of expected conditions to be achieved in X-2 as an expansion tube are presented in table 1. These results were calculated using an analytical wave tracking code including equilibrium chemistry for air[2] and Mirels[3] boundary layer theory to correct the interface trajectories. Case one and two use the conventional expansion tube configuration of shock and acceleration tube. Case three uses another shock tube immeadiately before the shock tube known as the secondary driver tube, as described by Morgan and Stalker[4]. This arrangement allows the generation of suber-orbital flow, as explored experimentally by Neely and Morgan[5].

Table 1: Calculated X-2 Conditions

Case	Test Gas	Primary Shock Speed (m/s)	Secondary Shock Speed (m/s)	Total Enthalpy (MJ/kg)	P_7/P_4^* (10^{-6})	Test Time (μs)
1	He	4111	6914	27	250	90
2	Air	2800	5500	13.5	500	100
3	Air	12000	16000	180	850	35

$^*P_7$ is the test section static pressure and P_4 is the driver pressure prior to burst.

It is hoped that the knowledge gained from these experiments will allow the design and construction of a much larger expansion tube using the two-stage driver concept.

5 Acknowledgments

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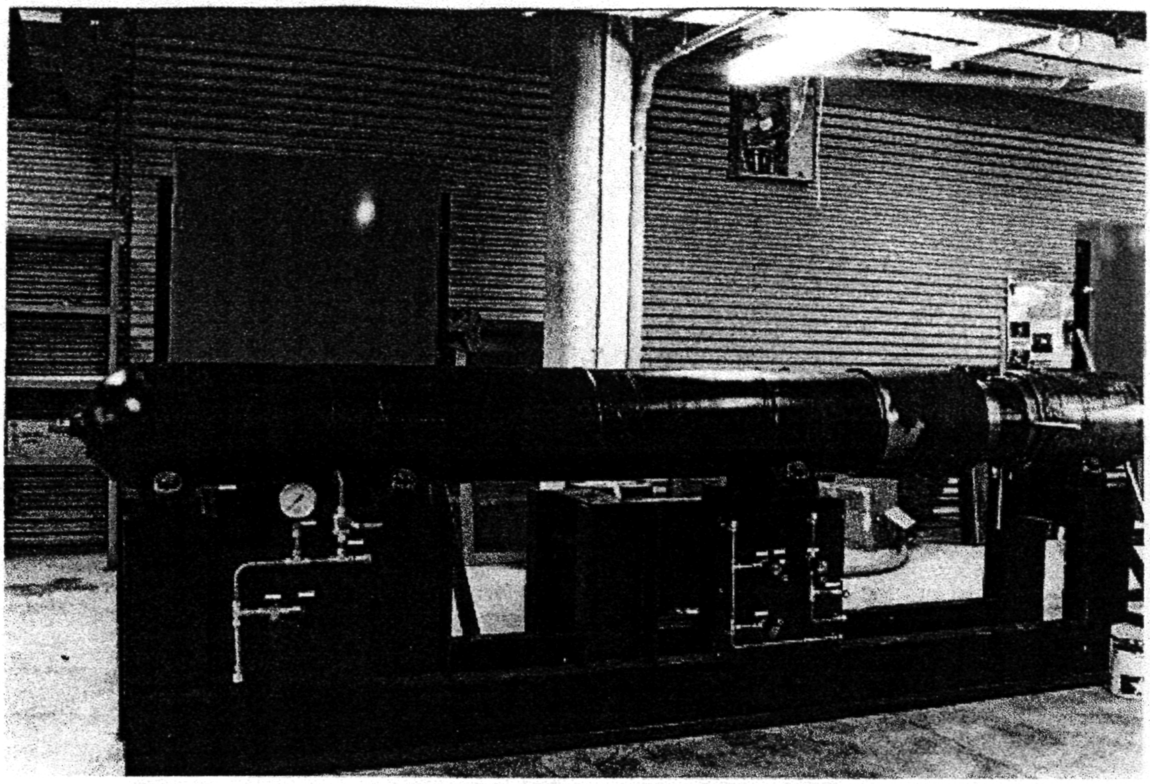

Figure 6: Photograph showing X-2 reservoir installed in Laboratory

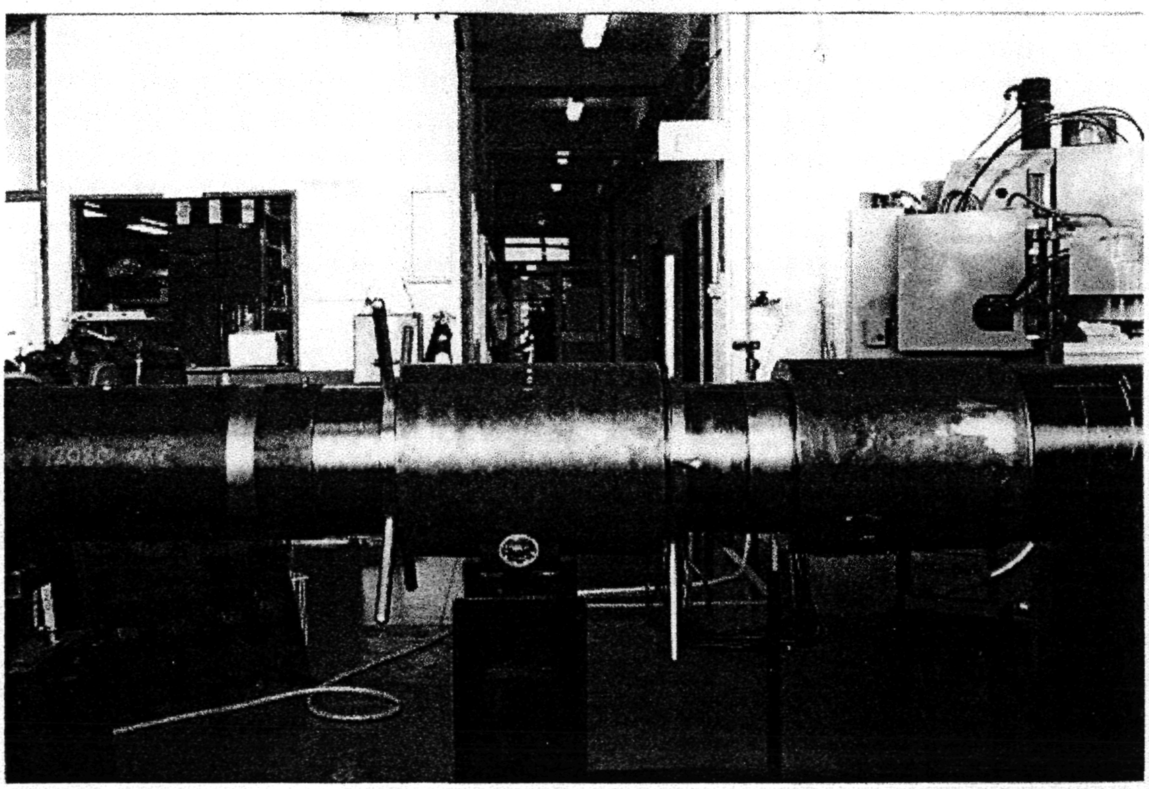

Figure 7: Photograph of double diaphragm launch station

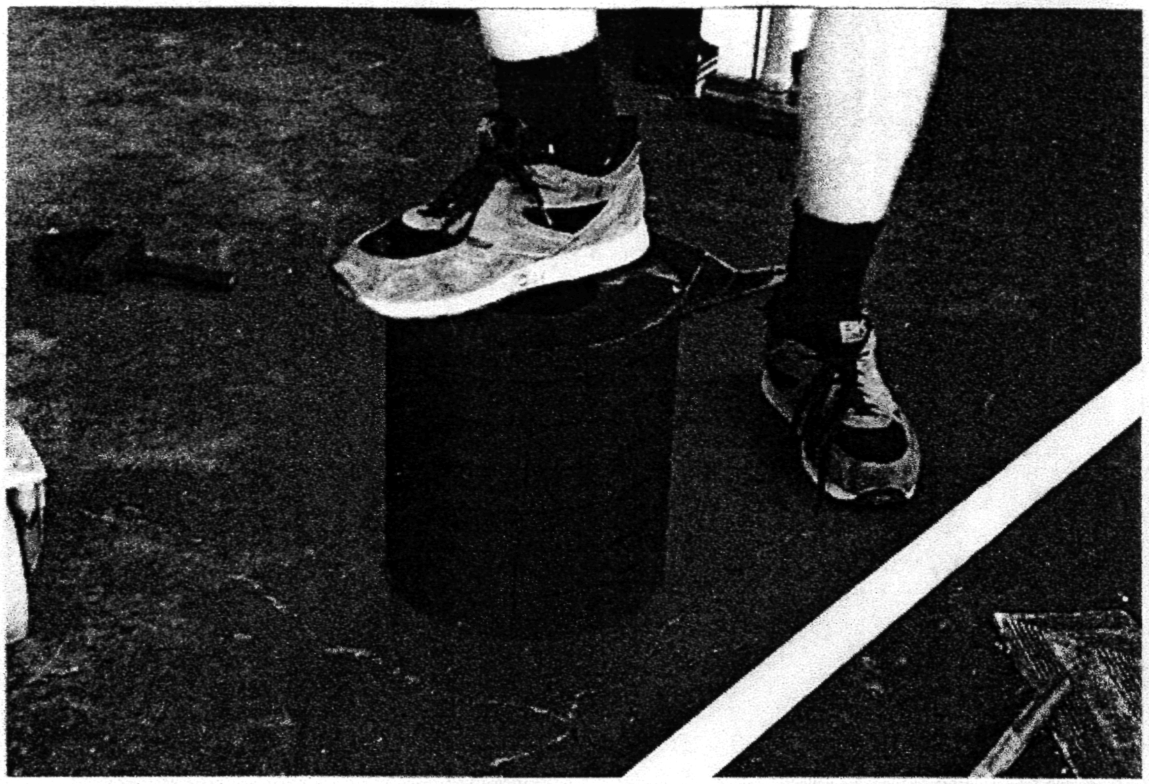

Figure 8: Photgraph of the buffer used to stop the outer piston

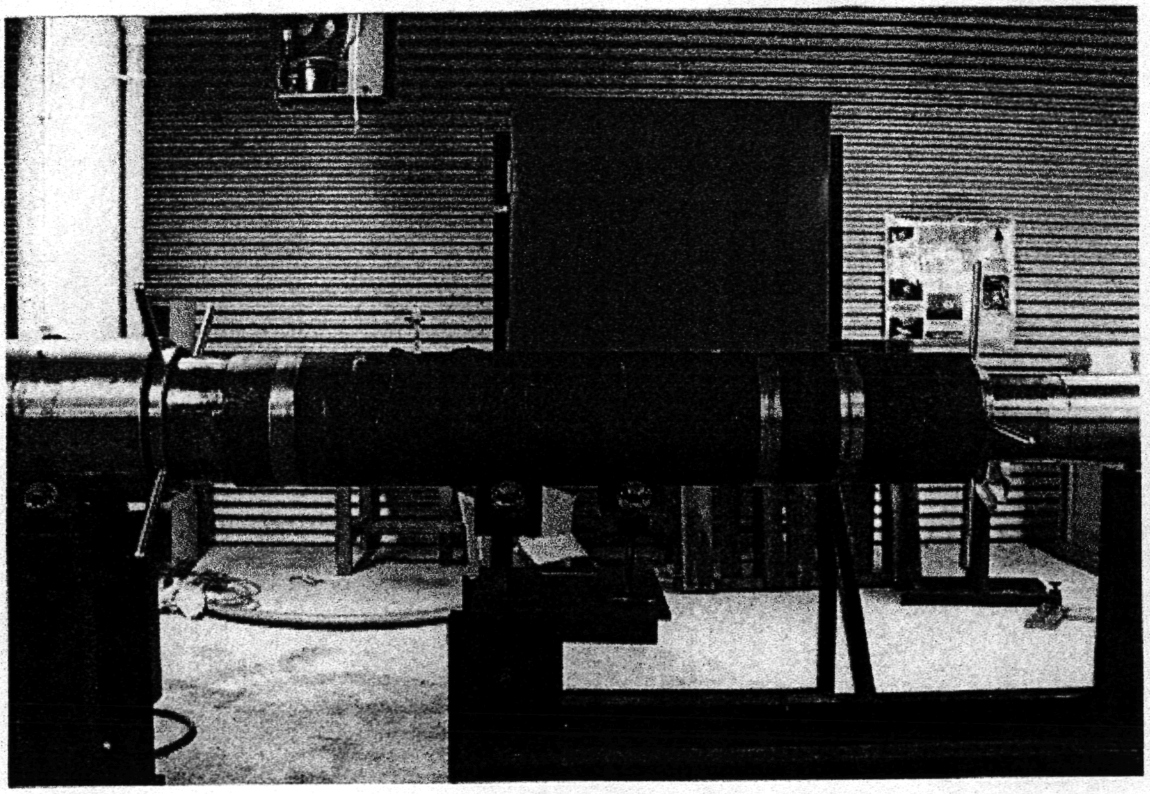

Figure 9: Stage one compression tube

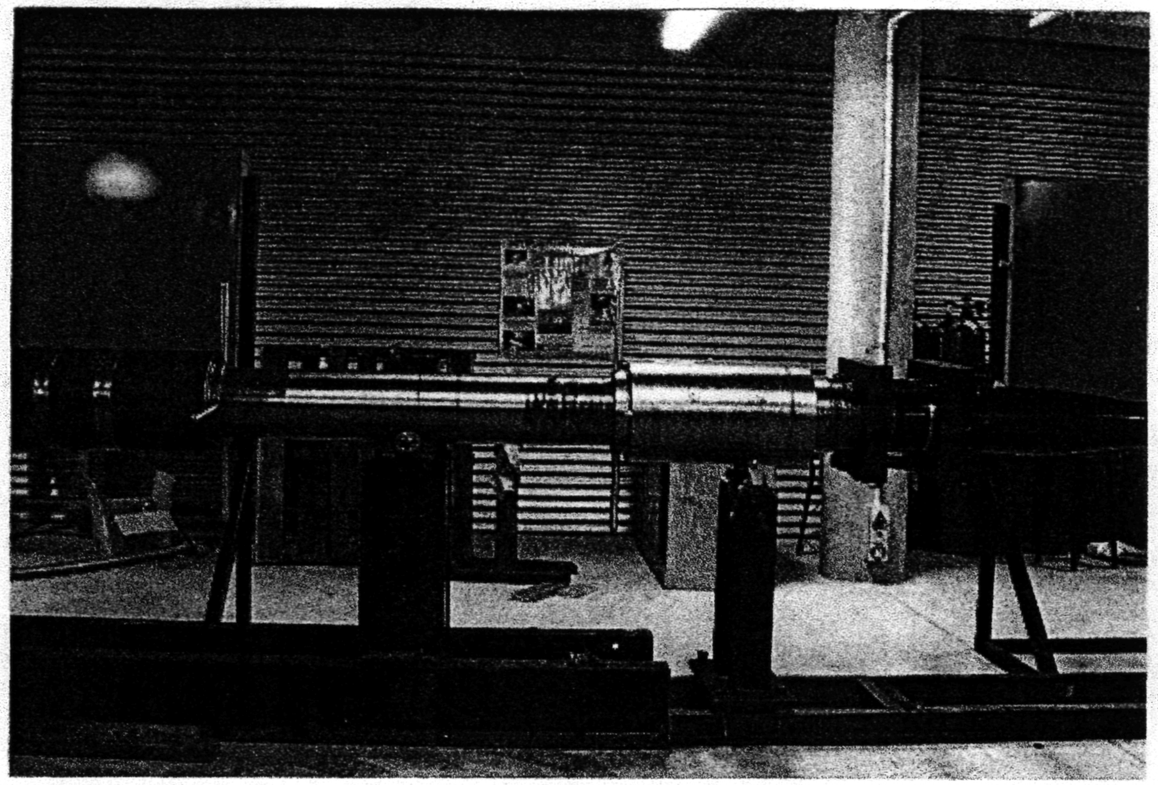

Figure 10: Stage two compression tube

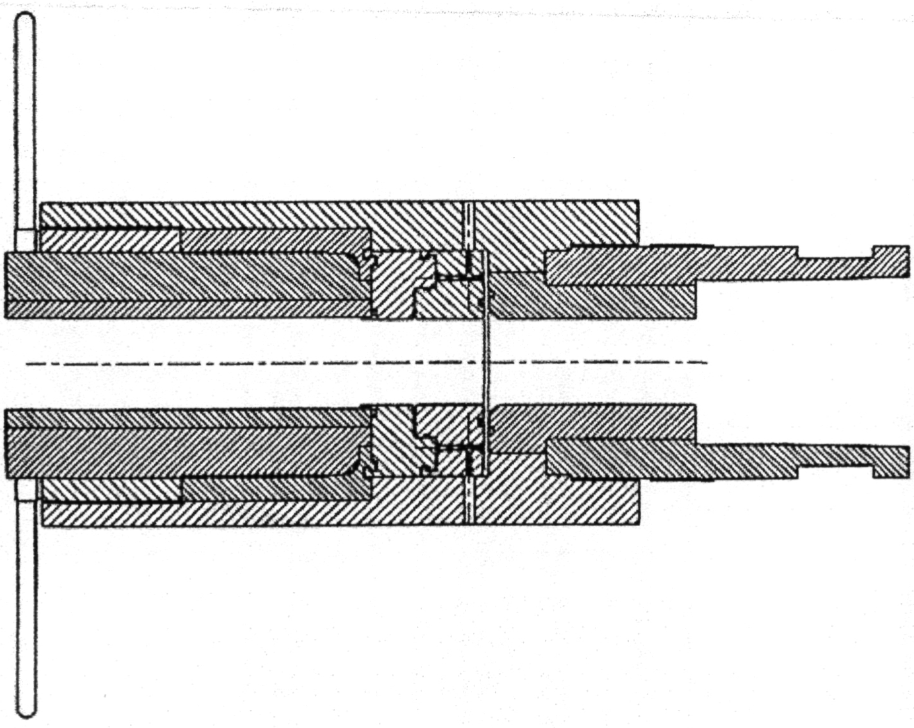

Figure 11: Sectioned view of primary diaphragm station

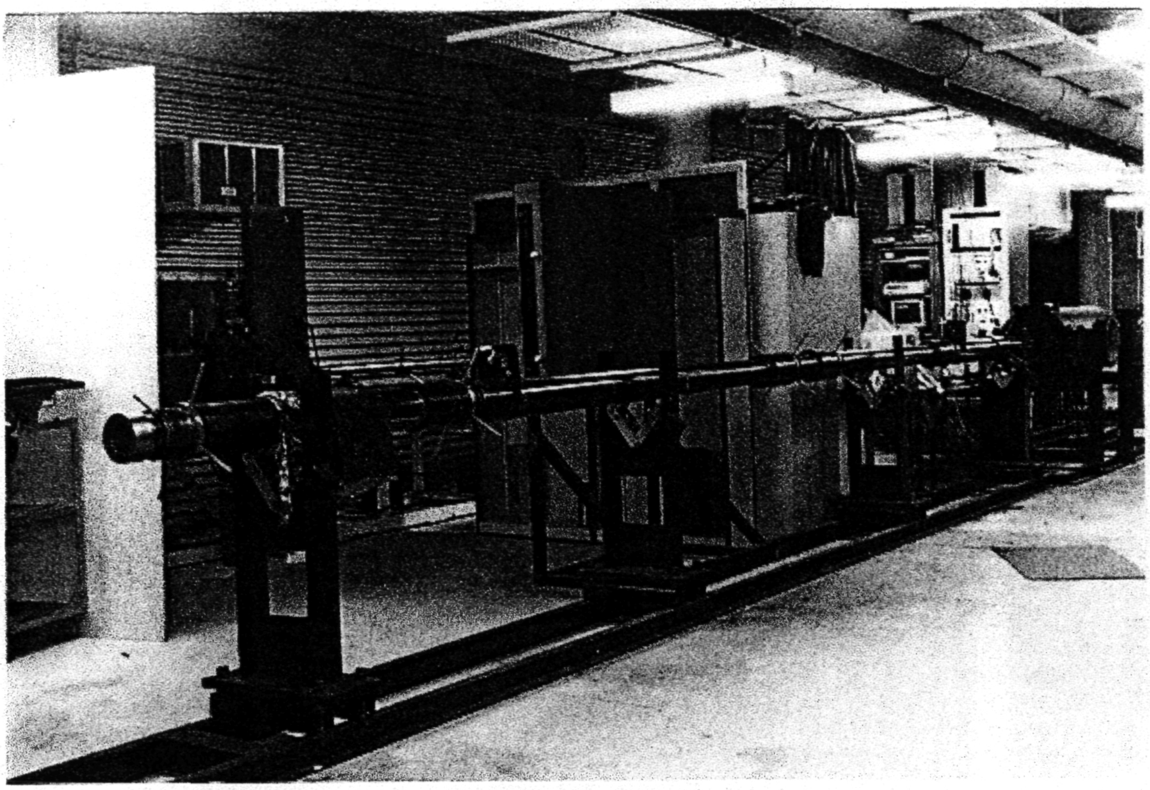

Figure 12: X-2 shock tubes installed in laboratory

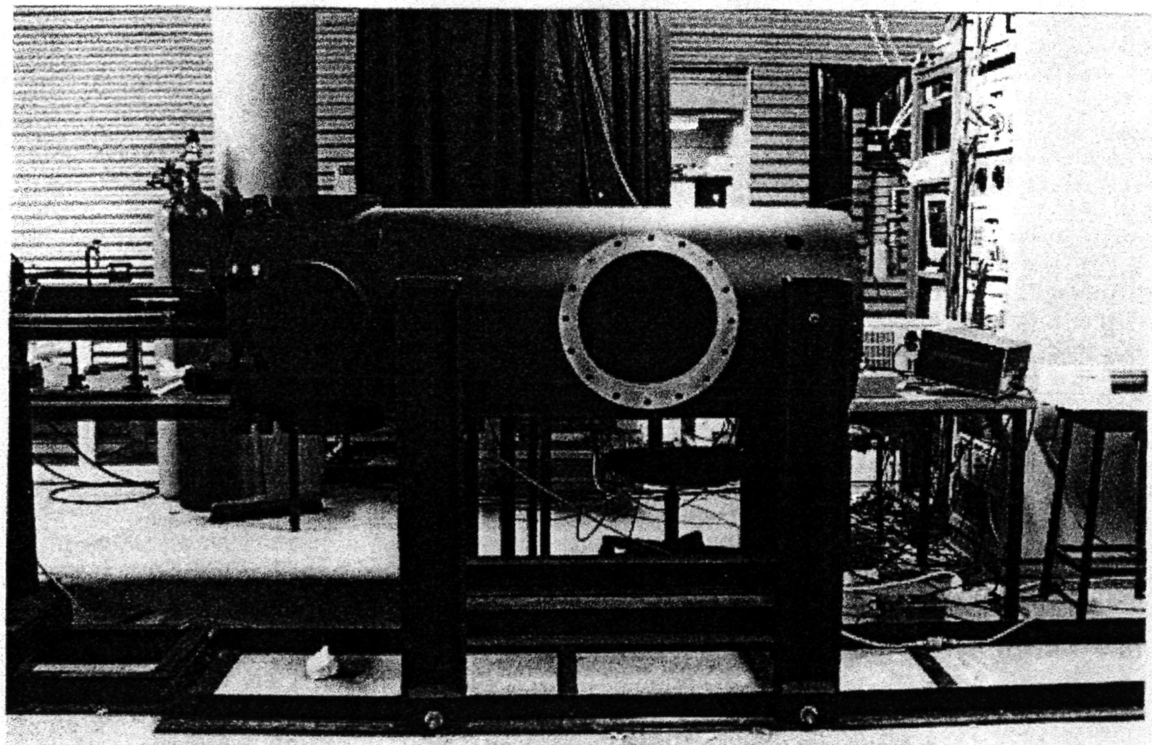

Figure 13: X-2 test section/dump tank installed in laboratory

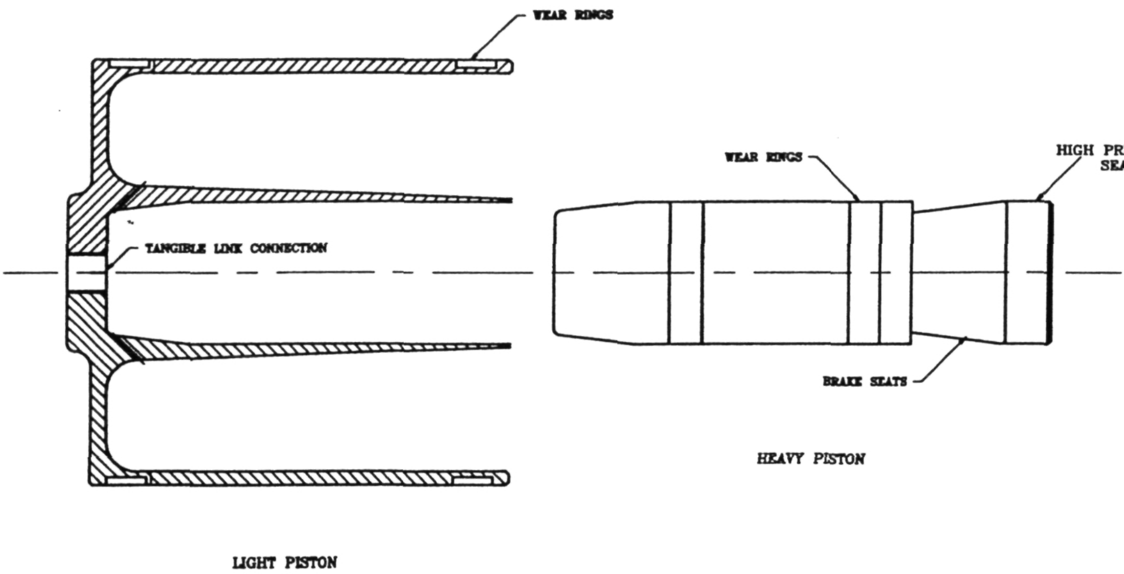

Figure 14: Sketch of X-2 Compound Piston

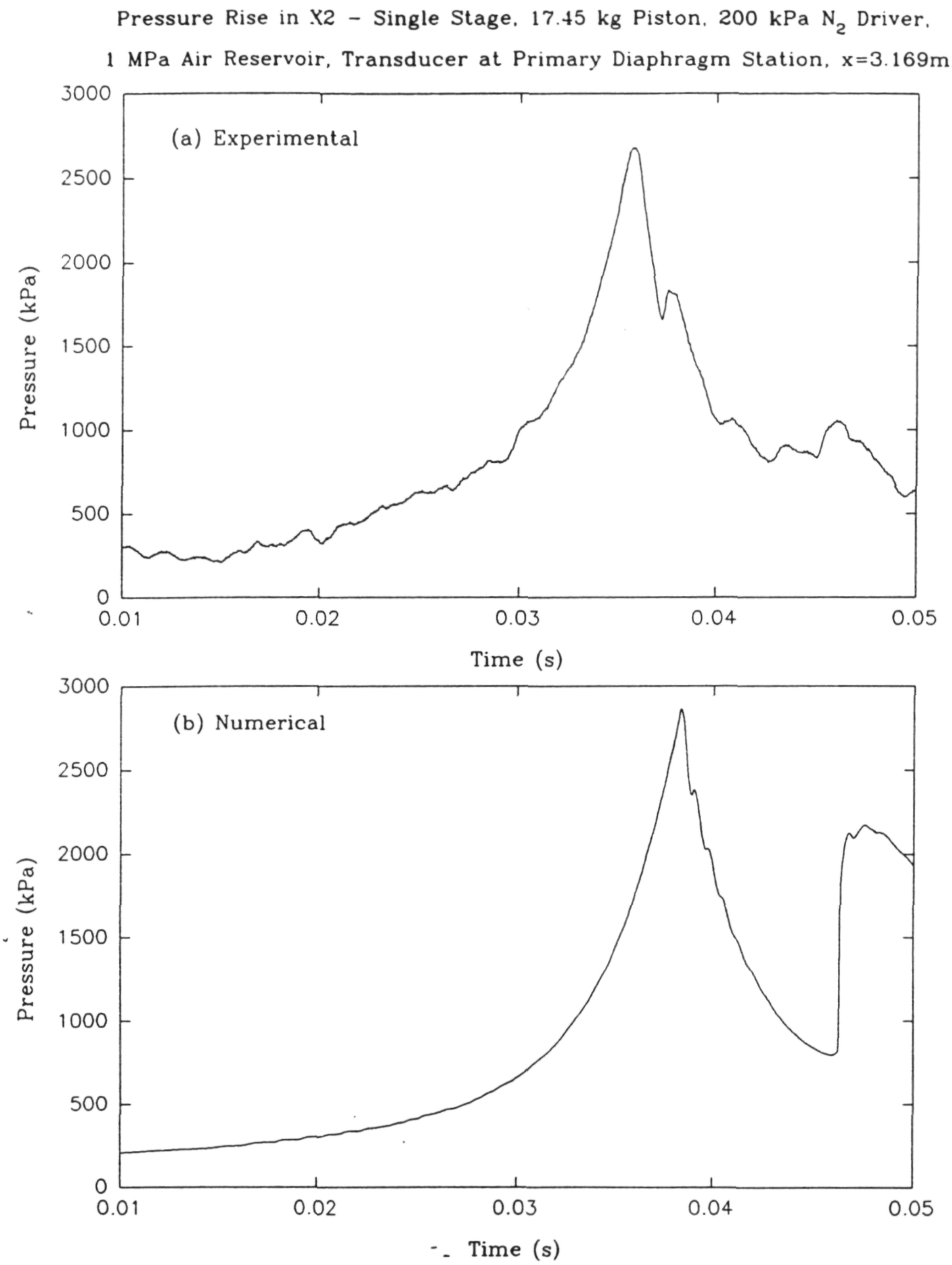

Figure 15: Comparison between experiment and numerical simulation