11th Australasian Fluid Mechanics Conference
University of Tasmania, Hobart, Australia
14-18 December 1992

A COMPOUND ISENTROPIC FREE PISTON DRIVER FOR EXPANSION TUNNELS

R.G. MORGAN

Department of Mechanical Engineering
University of Queensland
QLD 4072, AUSTRALIA

Abstract

This paper presents a proposal for using a two stage, in line, isentropic free piston compressor for use as a driver for shock and expansion tunnels. Noting that the early stages of the compression process transfer energy from the reservoir to the piston, and are characterised by large volumetric displacements and low pressures, the first stage of the compression tube is large bore and of a light construction. The pistons of the two stages are concentric, with the light first stage piston acting as a carrier for the second piston, which contains most of the mass of the two piston combination. At the end of the first stage most of the reservoir stored energy has been transferred to kinetic energy in the secondary piston, which separates from the primary carrier piston and enters the secondary compression tube. The second stage is where energy is transferred from the piston to the driver gas, and is characterised by low volumetric displacements and high pressures. The secondary driver therefore, is of small bore and constructed to withstand significant pressures, ~200MPa. The advantage of the concept lies in reduced total axial length per unit length of compressed driver gas.

Introduction.

Free piston compressors have been used successfully on reflected shock tunnels for many years, and have recently also been shown to have promise as drivers for expansion tubes, Neely et al (1991), and Morgan and Stalker (1991). However, as the compression process is nearly isentropic, a large volumetric compression ratio is required to acheive a given driver gas enthalpy, when starting from room temperature gas. Large shock tunnels therefore require large driver tubes, which being vessels typically rated to pressures of the order of 200Mpa, form a significant fraction of the overall facility cost. It is the goal of this project to produce a superorbital expansion tube with flow simulation capability in the region of 20km/sec and with shock tube diameters of the order of 200mm. The approximate driver requirement will be a temperature of 6500K, sound speed 5km/sec in helium and a pressure of 200MPa, which calls for an isentropic compression ratio of 100. In order to do this within typical University budjet restraints, ie total project costs of the order of 100,000's rather than millions of dollars, an new form of driver concept will be required, as well as a supply of cheap high quality steel. Any process which causes an increase in driver gas entropy may be used to reduce the pre-compression size of the driver gas slug.

Several non-isentropic processes have been proposed to produce the required enthalpy ratio in the driver gas , but with a lower overall volume ratio. Knoos (1969), built a driver which utilised a 'bypass piston' , Fig 1a, to shock heat the driver gas before the start of the isentropic compression process. The driver concept was successfuly proven, but has not been followed up since. For the application proposed here, pre compression shock speeds of the order of 2 km/sec would be required in order to produce sufficient volume reduction. To acheive this shock speed using room temperature helium for both the primary driver and reservoir gas would require excessive quantities of helium, or the complication of helium recovery, and has not been considered in its original form. However , the concept of combined shock and isentropic heating does have the potential to form a compact driver.

Bogdanoff (1990), has proposed a technique for the pump tubes of light gas guns which could potentially be used for shock tube drivers. The driver gas is allowed to expand irrreversably on diaphragm rupture into an exacuated section of the tube, Fig 1b, conserving internal energy, and thus static temperature. This technique is attractive because of its simplicity, and has the capacity for 'cascading' several non isentropic processes together to produce a very compact overall driver configuration. An unknown with this technique is the uniformity of driver gas conditions which can be produced. The irreversability comes about through a series of shocks, which will not produce the same entropy change through all of the gas. The effect of this driver gas non uniformity on shock propagation and interface stability has not been investigated sufficiently to prove the device for the purposes of driving high enthalpy shock tunnels. It is possible that if the non isentropic process is followed by further isentropic piston compression, then mixing will be sufficient to create an adaquately uniform driver mixture. If this is shown to be the case, then it will form a very attractive shock tube driver.

Anfimov (1992) has combined a throttling process with free piston compression to increase the volume of gas in the stagnation region of a supersonic wind tunnel, Fig 1c. This resulted in very long test times, and moderate enthalpies corresponding to total temperatures in air ~3500K, Roffe et al (1991). The throttling process is acheived by releasing the isentropically compressed gas through a relief valve into a nozzle supply chamber at a reduced pressure. The concept can in principle be used to provide the driver gas for a shock tunnel. However, a number of potential problems are evident at the energy levels required for the super orbital expansion tube. If the compressed gas is throttled to its final pressure with no further compression, then the pre- throttling pressures will be prohibitively high, and the pre throttled gas would be able to drive a higher shock speed, although there would be less of it. If post-throttling compression is to be used then mass flow rates through the throttling device must be high , and may present an erosion problem. The same theoretical advantages apply to the Bogdanoff approach, above, which is simpler and more controlable, and probably therefore the preferred of the non-isentropic concepts for this application.

Before adopting one of the entropy raising schemes discussed above, it is instructive to investigate the isentropic process a little further, to see if it can be improved upon, and that is the purpose of this paper.

The isentropic compression.

In a free piston compressor, Stalker (1972), the piston acts as an energy storage device in the transfer of energy from the reservoir to the driver gases. The compression process may simplisticly be considered to take place in two parts, namely transfer of energy from reservoir to piston, and transfer from piston to driver gas. The two processes evidently overlap, because at all times when the piston is moving work is being done on both gas volumes. However, due to the ramp like rise in driver gas pressure towards the end of the piston stroke, the region where the pressures on each side of the piston are of the same order is small, and it is descriptively convenient to separate the processes.

During the early stages of piston displacement, driver gas pressures are low, and swept volume is high. In a similar manner to two stage reciprocating compressors, it is proposed have a two stage compression, with a change in compression tube area at approximately the point where the pressures on either side of the piston are matched, Fig 2. The primary compression tube will be of large bore, and be rated to moderate pressures ~2MPa. When the piston reaches the change in section, the reservoir will already have done most of its work, typically 80 to 95%, Fig 3. However, only 5 to 20% of this work has so far been transferred to the driver gas, so the piston must be able to transfer the stored energy to the gas in the secondary compression tube. It is proposed to do this by means of a double piston as shown in Fig 4. The outer piston acts as a carrier for the inner piston, which contains most of the mass of the combination. On reaching the change in tube section, the inner piston is released and enters the secondary compression tube where it uses its residual kinetic energy to compress the driver gas to the final conditions. The outer piston is stopped at the tube interface by a gas buffer. Due to its low mass it contains little kinetic energy.

To further reduce facility cost and complication, it is proposed to match the secondary compression tube diameter closely to that of the attached shock tube, rather than have the area ratio which free piston compressors have used in the past. The beneficial effect of this is the reduction in diameter of the highly stressed driver components, but there are two adverse influences which must be addressed.

Firstly, the area change from driver to shock tube allows for a region of steady flow before the start of the unsteady expansion, which gives increased shock speed for a given driver condition. This is not possible for the constant area configuration, and will lead to a drop in performance as discussed theoretically and experimentaly in Morgan and Stalker(1991). It is estimated that this will lead to an increase by a factor of approximately 2 in the rupture pressures required to drive the shock tube in the present application. This will require the smaller diameter driver to be designed to withstand higher pressures, which partially offsets its advantage.

Secondly, piston velocity helps maintain driver pressure after rupture, which if there is sufficient area ratio may offset the reflected expansion waves from the piston face and produce a 'holding time' characteristic of a much larger fixed volume driver. To do this in constant area configuration would require piston velocities of the order of the driver gas/driven gas interface velocity, which is impossible for the primary shock speeds envisaged here, ie 5 to 10km/sec. The slug of compressed driver gas must therefore be long enough to adaquately delay the arrival the reflected expansion waves. This consideration, in conjunction with the pressure penalty above, is the prime factor in determining the most economical driver tube to driven tube area ratio for a given aplication, and applies equally to isentropic and non-isentropic compressors.

In the analysis that follows, reservoir and driver gas work has been calculated assuming steady isentropic expansion and compression processes. This approach is offered as a means of estimating the interaction of the various tunnel parameters, and not for calculating detailed experimental conditions for commissioning purposes.

The overall length of the driver will be a function of the area and length ratios of the two driver sections. The length ratio is more conveniently expressed as compression ratio at the end of the first stage of compression, in which form it is dependent on area ratio. The independant nondimensional variables chosen are overall volumetric compression ratio, λ, the compression ratio at the end of the first stage, $\lambda 1$, and the area ratio between the two driver tubes, Ar. The interaction of these parameters is shown in Fig 5 for a typical condition with a λ of 90. The normalisation of $\lambda 1$ with the square root of λ is chosen because the compression ratios multiply, and a value of one indicates that the compression is balanced between the two stages. It is seen that increasing area ratio is subject to a law of diminishing returns, and for subsequent analysis a value of 9 has been chosen. In Fig 6 driver tube non dimensional length, normalised by the length which would be required for a single stage, is plotted for a range of compression ratios with an area ratio of 9. It is seen that length reductions by a factor of between 3 and 6 are to be expected.

At the design stage the parameters λ,$\lambda 1$, and Ar may indeed be considered independent, but once a geometry has been selected $\lambda 1$ is fixed, and λ is the only variable left. Also shown on Fig 6 in dashed lines are the fixed geometry loci giving the dependence of driver length on compression ratio, which indicate the range of slug lengths a given design will be able to provide.

Other considerations however, provide restrictions to the operating range shown in Fig 6. The pressures in the primary driver must be low, in order to be able to build the large bore tube cheaply and show an advantage over the single stage compressor. In Fig 7 the pressure at changeover is plotted, and setting a maximum limit to P1 of 2% rupture pressure restricts λ1 to values less than the square root of λ.

If the conditions are incorrectly set, there is a possibility that the heavy secondary piston may separate before reaching the secondary tube. In this instance the piston would be unsupported and extensive damage would result. To protect against this,a fail safe locking mechanism will be required to prevent early release. To do this reliably it is preferred that under normal operation the mechanism does not have to open under load, ie when there is a net separating force between the pistons, which occurs when the pressure in front of the piston combination is higher than the pressure behind it. The pressure ratio P1/Pr1 is plotted in Fig 8, and the permitted regions of operation are identified. Combining the two plots Figs 7 and 8 indicates a solution with a primary compression ratio of approximately 6, and a minimum total compression ratio of 60. Operation at lower compression ratios would lead to excessive pressure at the changeover point, and could only be acheived by reducing the final rupture pressure. However, considering the use proposed for the driver, it is unlikely that such conditions would be of interest.

The length reduction over a single stage compression which would be acheived by this design can be seen from Fig 6 to be approximately 4. This is about half the value to be expected from the entropy increasing schemes discussed above. The isentropic option therefore can only be justified if it leads to better flow quality and shock propagation. An experimental investigation is currently under way to address this question.

Conclusions.

A two stage isentropic free piston compressor for driving super orbital expansion tubes is proposed, which can in principle be more compact than a single stage isentropic driver. The design is constrained by the requirements to limit the magnitude of the first stage pressure, and to prevent piston separation before the second stage. Axial length reductions of the order of four over a single stage isentropic compression seem to be possible, which is not as good as has been claimed for compressors which include non isentropic processes. The concept has the theoretical advantage of being able to produce driver gas with more uniform properties than the alternative proposals.

Acknowledgements.
This work was supported by the Australian Research Council.

References

ANFIMOV, N (1992) TSNIIMASH Capabilities for Aerogasdynamical and Thermal Testing of Hypersonic Vehicles. AIAA paper no 92-3962, AIAA 17th Aerospace ground testing conference, July 6-8,1992 Nashville,TN

BOGDANOFF, D W (1990) Improvement of pump tubes for gas guns and shock tube drivers. AIAA Journal Vol 28, No 3, March 1990.

KNOOS, S (1969) Study of a bypass piston shock tube. 7th International shock tube symposium, Toronto, 1969.

MORGAN, R G, STALKER, R J (1991) Double diaphragm driven free piston expansion tube. 18th International symposium on shock waves, Sendai, Japan 1991.

NEELY, A J, STALKER, R J, PAULL, A, (1991)High enthalpy, hypervelocity flows of air and argon in an expansion tube. The Aeronautical Journal, June/July, 1991, 175-186

ROFFE, G, ERDOS, J, TAMAGNO, J. Description of a unique blowdown wind tunnel and evaluation of its capabilities for hypersonic airbreathing propulsion research. GASL TM 245A, General Applied Science Laboratories, Inc., New York

STALKER, R J, Development of a hypervelocity wind tunnel. Aero Journal of the Royal Aero Society V76, pp374-384, 1972

Notation

Ar area ratio between first and second stage compression tubes
L total length of both driver stages
P1 pressure in driver gas at end of first stage compression
Pr1 pressure on rear piston face at end stage 1 compression
Pu primary diaphragm rupture pressure
x final length of compressed slug of driver gas
α ratio of reservoir to driver total volumes
λ overall driver volumetric compression ratio at rupture
λ1 stage 1 volumetric compression ratio

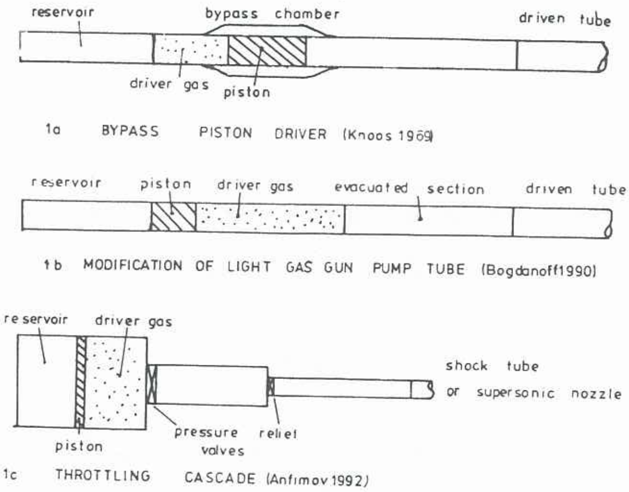

FIG 1 TECHNIQUES FOR COMBINING ISENTROPIC AND NON ISENTROPIC COMPRESSION

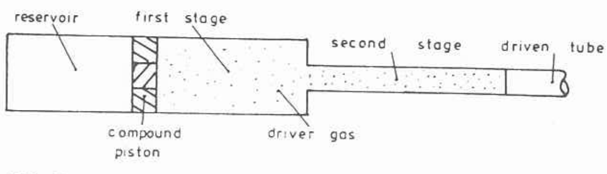

FIG 2 TWO STAGE ISENTROPIC COMPRESSION

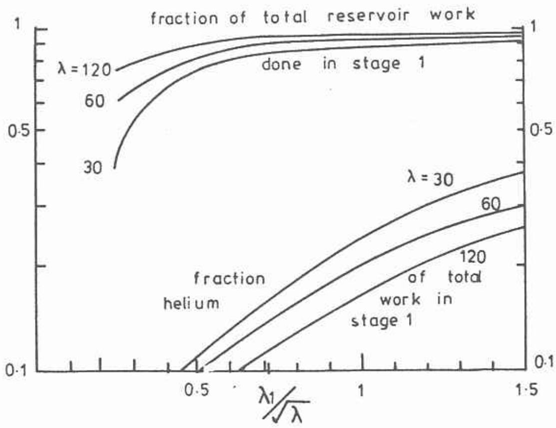

FIG 3 WORK DISTRIBUTION BETWEEN STAGES, $\alpha = 1$

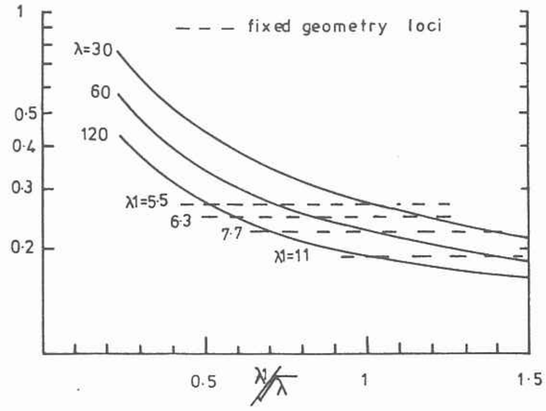

FIG 6 TWO STAGE/SINGLE STAGE DRIVER LENGTH RATIOS, Ar = 9

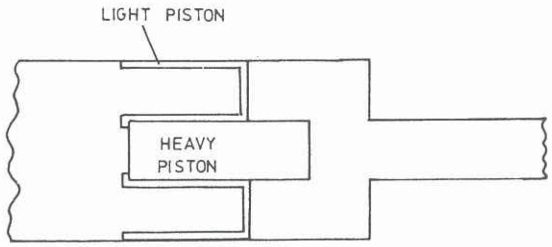

FIG 4 TWIN PISTON SCHEMATIC

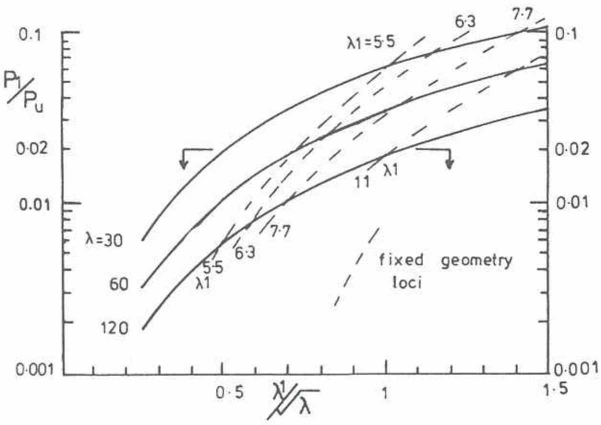

FIG 7 STAGE 1 PEAK PRESSURE

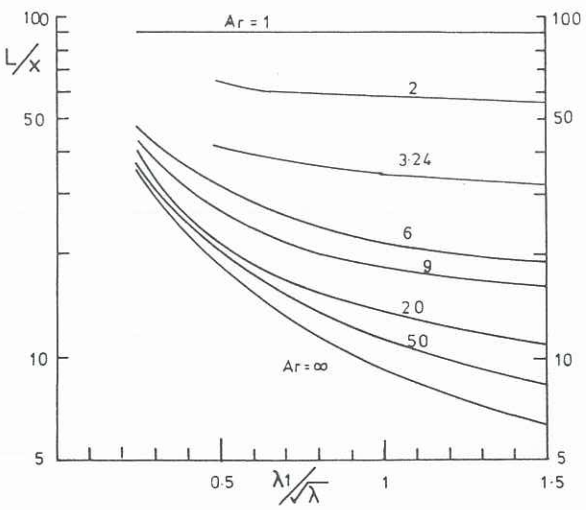

FIG 5 DRIVER TO SLUG LENGTH RATIO $\lambda = 90$

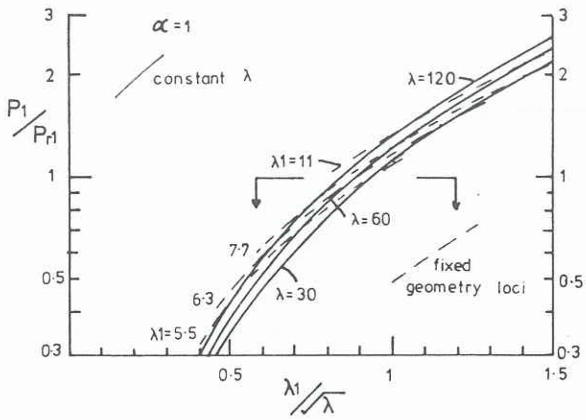

FIG 8 PISTON FORCE BALANCE AT SECTION CHANGE