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A STUDY OF PRESSURE PULSES IN A PIPE
WITH REFERENCE TO
THE EXHAUST SCAVENGING OF TWO-STROKE ENGINES

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ABSTRACT

The intake, transfer, and exhaust processes of the conventional two-stroke engine cycle, are described. The utilization of the exhaust pulse wave reflection process, to increase the two-stroke scavenging efficiency, is discussed. Several related theoretical and experimental investigations are mentioned, and in particular the work of the Institute of Sound and Vibration Research (I.S.V.R.) at Southampton University, is described.

The design and construction of test facilities for studying exhaust pulse shaping as related to the two-stroke engine, is detailed. The calibration and testing of these facilities is discussed along with the initial results of tests on a few promising exhaust components. Where possible results are compared with those of the I.S.V.R. experiments.

Results show a general agreement with the I.S.V.R. work, however, some important exceptions are noted.

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NOMENCLATURE

(1,2,3 etc.)	refers to reference list in appendix
BDC	bottom dead center
TDC	top dead center
EPO	exhaust port opens
EPC	exhaust port closes
TPO	transfer port opens
TPC	transfer port closes
η_s	scavenging efficiency
A_2/A_1	expansion or contraction, cross-sectional area ratio
θ	angle of cone divergence
Pcb.	picocoulombs
V	velocity (ft./sec.)
t	time (sec.)
X	distance (ft.)
T	temperature ($^{\circ}$ R)
γ	specific heat ratio
R	gas constant (ft. lb./ Slug, $^{\circ}$ R)
a	acoustic velocity (ft./sec.)
M	Mach number
ΔP	shock wave amplitude
L,d	cone dimensions as defined below

(v)

Subscripts
1,2,3 etc.

unless otherwise noted, denote a quasi-steady
uniform state of the basic shock tube flow
shown below.

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ERRATA

Page (vi), Line 5	add "Defence" after "Division."
Page 2, Paragraph 2, Line 1	"Closed" should read "close"
Page 3, Paragraph 4, Line 6	"vave" should read "wave"
Page 4, Line 5	delete "of" in "area of expansion"
Page 4, Paragraph 2, Line 4	"it" should read "is"
Page 4, Paragraph 2, Line 5	"as" should read "has"
Page 6, Paragraph 1, Line 5	"empty" should read "apply"
Page 29, Paragraph 4, Line 6	" " should read "l"
Reference Page, Item #16	"Kiepmann" should read "Liepmann"

INTRODUCTION

Until recent years the small, naturally aspirated two-stroke engine received little attention from manufacturers. Only outboard motor and a few motorcycle companies were willing to tolerate their low specific power and high specific fuel consumption as a penalty for simplicity and light weight. With the exception of the large supercharged Diesel versions (1) the development of the "conventional" two-stroke engine has lagged behind that of the four-stroke engine. The recent demand for light, portable engines to power an increasing array of consumer goods has renewed interest in the two-stroke engine and has helped to close the technological gap.

1. THE CONVENTIONAL TWO-STROKE ENGINE CYCLE

Figure 1 shows the conventional two-stroke engine being considered for this investigation. This is the design most widely used for the small displacement two-strokes used in motorcycles, industrial and household machinery, and some small autos. There has been a trend in the U.S.A., especially among the outboard motor manufacturers, to port the carburetor directly into the crankcase through a non-return "reed" valve. This modification has certain advantages and disadvantages but does not greatly alter the exhaust cycle being considered.

Each revolution of the two-stroke engine involves the intake, transfer and exhaust processes, thus making the discussion of any one process on an individual basis very difficult. For convenience one fresh fuel-air charge will be considered as it journeys from intake to exhaust.

If the piston is considered on the upstroke from BDC, a partial

vacuum will be created in the crankcase. This vacuum will draw in the fresh fuel-air charge when the piston skirt uncovers the intake port at typically 65° before TDC. When the intake port closes at 65° after TDC the fresh charge is compressed in the base by the piston downstroke. When the piston crown uncovers the transfer ports at, typically, 60° before BDC, the fresh charge is forced into the combustion chamber. The remaining burned gases of the previous fresh charge are driven out the exhaust port (scavenged). Under most conditions a certain amount of mixing of the burned and unburned gases will occur and some of the fresh charge may be lost out the exhaust port.

A measure of the effectiveness of this scavenging process is the scavenging efficiency η_s defined by H. List (2) as the ratio of the fresh charge to the fresh charge plus residual gases.

Continuing with the cycle, the transfer ports closed at 60° after BDC and the exhaust port typically 20° later. The piston then has the remaining 70% of the upstroke to compress the cylinder charge prior to ignition. After ignition this 70% of stroke is available for power until the exhaust port opens at 30° before BDC. At this point the cylinder pressure will be approximately 70 psi., and will reduce rapidly to atmospheric during the following 40° before the transfer ports opens. The incoming fresh charge from the transfer ports will then scavenge most of the remaining burned gases thus completing the cycle of the preceding fresh charge.

In general the scavenging efficiency of the two-stroke is inferior to that of the four stroke. Figure 2 shows a comparison of the timing of typical two and four-stroke engines. From these diagrams

can be seen the considerably reduced time intervals during which the two-stroke cycle processes must occur. In the two-stroke the cylinder volume increases from 70% to full and back to 70% during the exhaust cycle. Thus the transfer charge receives little assistance from the piston in scavenging the exhaust gases. Exhaust gas contamination of the fresh charge causes a low specific power output and fresh charge loss increases the specific fuel consumption.

Supercharging and fuel injection will considerably improve the two-stroke η_s , but the associated increase in mechanical complexity makes their use on engines below 150 cu. in. displacement, of questionable value.

For two-stroke engines in the above category, utilization of wave reflections in the exhaust pipe due to the primary exhaust pulse, offers the easiest way to improve η_s .

Fortunately the exhaust pulsations present in the two-stroke exhaust pipe are somewhat stronger than those of a similar four-stroke engine. This is due to the two-stroke exhaust port opening at 70% of the power stroke as compared to 90% for the four-stroke exhaust valve.

2. EXHAUST PIPE WAVE REFLECTION PROCESS

Figure 3a is a simplified distance-time diagram of the wave reflection process in a plane, parallel, two-stroke engine exhaust pipe. When the exhaust port opens a compression wave will propagate rapidly into the exhaust pipe while a rarefaction wave will propagate back into the cylinder. A property of compression waves is that they tend to steepen into a shock wave, while rarefaction waves tend to flatten out. Once formed, the shock wave will propagate at a supersonic velocity down

the pipe to the open end where it will be reflected as a rarefaction wave. The rarefaction wave propagated back into the cylinder will be immediately reflected by the cylinder wall, still as a rarefaction wave and will follow close behind the incident shock wave. It should be noted that an area of expansion or an open end will reflect waves of opposite type while a closed end or area contraction will reflect waves of similar type. The rarefaction following the shock wave will always be reflected as a wave of the opposite type and this will tend to reduce any change in local pipe pressure caused by the preceding wave. For simplicity the details of the reflections of this "trailing" wave have been omitted from Figure 3a, but its effect on the exhaust port pressure-time history is evident.

The rarefaction wave due to the reflection of the primary shock wave at the open end will arrive back at the exhaust port where it will be reflected from the cylinder wall as a rarefaction wave. If the rarefaction arrives at the exhaust port between TPO and TPC the incoming transfer charge will receive considerable assistance in scavenging the cylinder, (exhaust pulse scavenging).

The rarefaction wave returning downstream will be reflected by the open end as a compression wave which may or may not steepen to a shock before it returns to the exhaust port again. If this compression wave is timed to reach the exhaust port between TPC and EPC the portion of the fresh charge which has been drawn into the exhaust pipe during the scavenging process will be forced back into the cylinder under pressure, (exhaust pulse ram-charging).

In general it is not possible to obtain the timing required by the above, from a plain open end. Experience has shown that an exhaust

system of the form shown in Figure 3b can produce the desired pressure-time history at the exhaust port.

The first area expansion reflects the required "scavenging wave" while the following area contraction reflects the required "ram-charging wave". The parallel pipe after the area contraction is required to insure that the rarefaction reflected from the open end does not arrive at the exhaust port before EPC, thus reducing the "ram-charging" effect.

It has been found that gradual rather than abrupt area changes are desirable, since the wave reflection process takes place over their entire length. The result is that the reflection time interval can be "shaped" to fit within a desirable crank angle interval.

Good exhaust system design can result in power increases of as much as 30% over a poor design (constant area pipe) (3). However it must be appreciated that a rigid exhaust system designed to optimize η_s will be effective over a very narrow RPM range. If the engine is operated outside this range the exhaust system may have a decidedly detrimental effect on η_s .

If the exhaust system of the hypothetical engine of Figure 3b produces the indicated characteristic at an engine speed of 5000 RPM, an increase of 500 RPM will result in the "scavenging wave" arriving after TPC thus removing fresh charge. The "ram-charging wave" will arrive after EPC when it will be ineffective. A decrease of 500 RPM will cause the "scavenging" wave to arrive during the cylinder blowdown period where its effectiveness is reduced and the "ram-charging wave" will arrive before TPC impeding the outflow of the exhaust gases.

Thus to optimize η_s a constant RPM application is required.

To date most manufacturers side step the entire exhaust problem by designing neutral systems which damp or destroy all pressure waves. Notable exceptions have been several German and Japanese motorcycle firms which have made efforts to employ exhaust pulse scavenging and ramming on their two-stroke engines. Results have been impressive; 2.4 HP/cu. in. for a 15 cu. in. single cylinder road model and 3.0 HP/cu. in. for a 4.9 cu. in. twin cylinder racer. Unfortunately, for the reasons of the previous paragraph, these high outputs are obtained at the expense of multiple speed gearboxes (9 for the racer) to maintain the engine in its effective RPM range.

In the case of multicylinder engines, the best system is a separate exhaust pipe for each cylinder. However if no attempt at "ram-charging" is made, the pipes can be grouped effectively to provide good scavenging, a practice in use on some four-stroke automotive engines.

3. Theoretical Background.

In the thirties a French engineer, Michel Kadenacy, proposed a theory of exhaust scavenging based on mass transfer rather than wave phenomena. He proposed that an engine having a rapidly opening exhaust port of large cross-section and preferably no exhaust pipe to restrict gas flow, would benefit from a large depression in the cylinder caused by the momentum of the outflowing gases. His theories caused considerable controversy at the time, (7) and later investigations have denied the existence of the effect in anything like the magnitudes he proposed. (5,8)

Although considerable theoretical and experimental work has been done on exhaust system performance since the thirties, (2,4,5,6), the results involved limitations from the designers point of view and in most cases only provided a starting point for cut and try methods.

Engineers at the Institute of Sound and Vibration Research (I.S.V.R.) at Southampton University are involved in an investigation of motor vehicle noise. Their preliminary research amplified the need for a better method of predicting the behavior of pressure pulses in exhaust systems. The application of steady flow theory modified by experimental coefficients proved inflexible. The digital or graphical, finite difference application of small perturbation theory (Method of Characteristics), produced no basis for predicting the performance of new systems. The small perturbation theory becomes more tractable when modified with the assumptions that changes of state are isentropic and all disturbances are propagated with acoustic velocity (6,9) In this form the theory will hold for expansion waves but not necessarily for compression waves, especially when pipe area changes

are substantial and shock waves are present.

The approach followed by the engineers at I.S.V.R. is based on one-dimensional unsteady flow theory as applied to the shock tube. This theory is a particular application of a more general theory described fully by G. Rudinger (10), and others.

With this method it was possible to calculate the amplitude of the transmitted and reflected waves produced by an incident compression wave travelling into still air in a pipe of given area change. The calculations for any one set of conditions involved a lengthy iterative procedure described fully in reference (11). With the aid of a computer, charts were compiled for contraction and expansion area ratios of 1 to 16 and incident compression wave pressure ratios of 1.2 to 2.0 in steps of 0.2. Records indicated that incident compression wave amplitudes of half an atmosphere are typical for most engine exhaust pipes. The I.S.V.R. charts are reproduced in Figs. 4 & 5. In these charts all area changes are assumed abrupt. Gradual area changes are assumed to affect only the time required for the reflection process to occur.

The I.S.V.R. Engineers justify their assumption that each primary pressure pulse can be treated as an independent disturbance by the fact that the primary pulses in the exhaust pipe of a four cylinder, four-stroke, engine operating at 3000 RPM, may be separated by as much as 18 feet. In the case of a two-stroke engine producing twice as many primary pulses per revolution at typically higher engine speeds, the above assumption may not be valid. An attempt to investigate the dividing line is reported later.

Further support for the shock tube approach is obtained from the fact that pressure rise rates greater than half an atmosphere per millisecond have been observed near the exhaust valves of internal

combustion engines. This will result in flow behavior which, within a few diameters, will approach that in a shock tube (ie: compression waves will steepen into shock fronts while rarefaction waves will flatten out).

Experimental investigations were carried out at Southampton University on various exhaust system components, including area increases and decreases, orifice plates, perforated tubes and branch points, using a simple shock tube and air at room temperature (12). Good correlation was obtained with the theoretical predictions in the wave amplitude ranges of interest. Their experiments indicated that the only factor affecting the amplitude of the compression waves were changes in pipe cross-sectional area.

It remained to be shown that a bursting diaphragm in a shock tube produces the same disturbances as an engine exhaust pulse. Experiments are continuing at Southampton University with this in mind.

INVESTIGATION OBJECTIVES

The primary purpose of the I.S.V.R. experiments was the suppression of motor vehicle exhaust noise by the reduction in amplitude of the compression waves propagated out of the exhaust pipe.

It was felt that a further investigation of exhaust pipe phenomena, with respect to performance improvement rather than noise reduction, was required. In addition, it was felt that the investigation should be specifically aimed at the two-stroke engine since this cycle can benefit most from the exhaust pipe wave reflection process.

The unavailability of a suitably instrumented two-cycle engine resulted in the choice of the shock tube technique as the approach most likely to produce useful results in the time available.

The primary mechanical and experiment objectives of the test program were as follows:

1. MECHANICAL DESIGN

- (i) To design, and instrument a simple diaphragm-type shock tube which could be constructed by the University shop.
- (ii) To design a suitable number of test sections which could be constructed in the relatively short time available.
- (iii) To design a simple exhaust pulse generator which would duplicate with air at room temperature, the action of the two-stroke exhaust cycle.

2. EXPERIMENTAL STUDIES

- (i) To verify the satisfactory operation of the shock tube and its instrumentation.
- (ii) To record and study the amplitudes of the transmitted and reflected waves from various pipe area changes both abrupt and gradual.

- (iii) To correlate where possible with the Southampton University experiments.
- (iv) To test the effectiveness of the exhaust pulse generator in simulating the two-stroke exhaust cycle.
- (v) To study the extent to which wave reflections from previous exhaust pulses modify succeeding ones.
- (vi) To compare, for the same pipe geometry, the transmission and reflection of a compression wave produced by a bursting diaphragm with one produced by the exhaust pulse generator.

THE APPARATUS

1. THE SHOCK TUBE DRIVER SECTION

Figures 6 & 7 illustrate the chief interior and exterior details of the shock tube driver section. The tube is 1.50in. I.D. x .25in. wall, seamless steel tubing. This bore size was chosen since it is typical of the two-stroke engine exhaust pipes being considered. Seamless steel tubing of the correct I.D. and O.D. was also used for all flanges to save shop time. All joints were brazed with Handy and Harman, "Easy-Flo #45", silver brazing alloy, to minimize tube distortion, and were sealed where necessary with neoprene "O" rings.

Because of the difficulty in obtaining naturally bursting diaphragm materials for the pressure ratios required (1.34 to 5.0) a spring loaded lance was employed to burst the diaphragm at the desired pressure ratio. It is normal practice on larger shock tubes to mount this lance diagonally through the tube wall, but due to the small tube bore, the lance was mounted concentrically through the closed end. The lance diameter of .25in. was chosen so as to require a minimum of spider supports without providing excessive "piston" action for the driver spring to overcome at the maximum test pressures.

The lance travel is $\frac{1}{2}$ in. with release controlled by a manually operated pin. The lance position with respect to the diaphragm can be adjusted and locked. When necessary the driver section can be extended from 3ft. to 5ft. by the addition of 2ft. sections to the tube and lance.

The diaphragm chuck design represents the best balance between ease of operation and ease of construction and was quite reliable. Circular alignment rings were used at all tube flanges instead of dowel pins or spigots. This allowed a greater number of combinations of the

test sections since all tubes were interchangeable end for end.

The design of the tube support legs is simple, rigid and allows height adjustment as well as location anywhere on the tube.

The driver section was pressurized with compressed air from the building supply, through a Quick-Lock pressure fitting.

2. DIAPHRAGM MATERIALS

Information on diaphragm materials for low pressure shock tubes is relatively scarce and what was found (13) did not include information on the array of new plastic films presently available. Previously, the most successful material for this application was a type of cellophane, "Red Zip", manufactured by the American Tobacco Company. It was reported to give good shock formation but was not available locally. It also had the disadvantage of shredding into fine pieces which necessitated frequent cleaning of the tube.

With the help of the Defence Research Board's Physical Testing Laboratory a number of locally available materials were obtained and tested. The results of these tests are presented in Table 1. The material finally selected was a .002in. thick, clear cellophane manufactured by Canadian Industries Limited. One layer gave good manual bursts from 15 to 30 psi. and two layers from 30 to 60 psi. Figure 8 shows a typical burst with almost no shredding. Most of the missing portions in the photo were due to handling during photography. When necessary, bursts below 15 psi. were made with household Saran which shredded but was reliable. Attempts at encouraging the tougher materials such as mylar and aluminum foil to burst at lower pressures by scribing, failed. Complete mechanical control of scriber pressure and speed would be required for burst pressure repeatability.

TABLE I

LOW PRESSURE DIAPHRAGM MATERIALS TESTED

Material	Manufacturer	Thickness (in.)	Natural Burst Pres. (PSI.)	Min. Pres. For Lance Burst (PSI.)	Remarks
Mylar	Dupont	.001, .002	-	-	lance caused slow leak up to 60 psi., scribing unreliable
Household Saran	Dow Chemical	.0005 two layers	7 21	4 10	shreds
Industrial Saran	Dow Chemical	.002	40	20	shreds
Polystyrene	Dow Chemical	.002	20	10	shreds
Cellophane	CIL	.002 two layers	40	15 30	negligible shredding
Aluminum Foil	Reynolds Alum. Ltd.	.002	-	-	lance caused slow leak up to 60 psi., scribing unreliable

It can be seen from Figures 6 & 7 that the pointed end of the diaphragm lance has four sharp, radial fins. It was found that these helped produce sharper diaphragm bursts at the lower test pressures.

3. TEST SECTIONS

Figures 9 & 10 show details of the abrupt and gradual area changes along with the various parallel tubes necessary to study the transmitted as well as the reflected waves. Even with the use of heavy wall mechanical steel tubing, to minimize metal removal, the shop time required for the construction of the taper sections eliminated the possibility of obtaining more than two. However by careful design fifteen different combinations were available for testing. These combinations are listed in Table II.

The area ratios of 3.362 and 7.111 were chosen to fall between the ratios tested and reported in (12). A study of the cone angles used in the tuned exhaust pipes of several two stroke motorcycle engines (14), indicated a preference of 8° for the divergent portion and 15° for the convergent portion. These angles were chosen for the two test sections with the object of finding some reason for their preference.

Only one abrupt area change was constructed since gradual area increases were of more interest for this investigation.

4. SHOCK TUBE INSTRUMENTATION

Figure (11) shows the general layout of the shock tube and instrumentation with the large 15° test section in position for testing. The distance between the diaphragm and the test section has been shortened for the photo. In general at least 20 diameters were allowed to insure the formation of a plane shock wave.

TABLE II

TEST SECTION COMBINATIONS

#	Test Section	A_2/A_1	θ°	Wave Measurements Possible	
				Trans.	Ref.
1.	Open End		180		x
2.	Open End with Cone	3.362	8		x
3.	Open End with Cone	7.111	8		x
4.	Open End with Cone	3.362	15		x
5.	Open End with Cone	7.111	15		x
6.	Gradual Expansion	3.362	8	x	x
7.	Gradual Expansion	7.111	8	x	x
8.	Gradual Expansion	3.362	15	x	x
9.	Gradual Expansion	7.111	15	x	x
10.	Abrupt Expansion	7.111	180	x	x
11.	Gradual Contraction	3.362	8	x	x
12.	Gradual Contraction	7.111	8	x	x
13.	Gradual Contraction	3.362	15	x	x
14.	Gradual Contraction	7.111	15	x	x
15.	Abrupt Contraction	7.111	180	x	x

The pressure transducers used were Kistler quartz type, model 701A, having a dynamic pressure range of .01 to 30 psi., a resolution of .01 psi. and a limited use to 3000 psi. Transducer sensitivity was nominally 5 picocoulombs/psi. and rise time 6 microseconds. The transducer output was fed through a Kistler multi-range charge amplifier, model 502 to the vertical input of a Tektronix type 564, 4 trace, storage oscilloscope with Polaroid camera attachment. Figure 12 shows a pressure transducer, a typical pipe mount and a charge amplifier.

The storage feature of the scope was particularly useful in this application since only one horizontal sweep was triggered per reading, at typical speeds of 1 ms./cm. This speed was too fast for eye observation and would have required a camera record of every reading taken on a non-storage type scope. The camera feature on the 564 scope was used to record interesting trace features for later verification or representative readings for report purposes. The multiple trace feature enabled the reflected and transmitted wave traces to be recorded for the same incident compression wave.

On a typical test run a pressure transducer a short distance downstream of the diaphragm was used to trigger the horizontal time base of two of the four beams. A second transducer mounted six inches in front of the test section and connected to the vertical input of the first beam, recorded the amplitudes of the incident and reflected waves. Unless the test involved an open end, a third transducer was mounted six inches downstream of the test section and fed the transmitted wave amplitude to the vertical input of the second beam.

For dynamic calibration of the pressure transducers it was necessary to obtain the shock wave velocity and amplitude over a known length of a parallel pipe. For this purpose the output of one transducer was used to start both the scope horizontal time base and a digital electronic counter. A second transducer, one foot downstream of the first was used to stop the counter and provide a wave amplitude reading on the scope vertical input. The Hewlett Packard Model 522B, 10^{-5} sec, counter shown in figure 12 was later replaced with a Transistor Specialties Inc. (TSI), model 365AR, 10^{-7} sec. counter for greater accuracy.

5. EXHAUST PULSE GENERATOR

Figures 13 & 14 show details of the pulse generator used to duplicate the two-stroke exhaust process. Originally it had been planned to design and construct the complete generator with provision for variable displacement and port timing. However time considerations ruled this out, with the result that a Villiers, model 6E, two-stroke motorcycle engine of 12 cu. in. displacement was used as a base.

The engine induction and transfer system was left unmodified except for the replacement of the carburetor with a bellmouth air intake. A new outlet pipe was welded to the exhaust port to allow mating with the 1.5" bore test sections used with the shock tube. A $\frac{1}{2}$ HP electric motor was used to drive the pulse generator through a "V" belt and multiple pulley arrangement providing an RPM of 970 to 2000, approximately. A flywheel was fitted and painted with a stobe indicating stripe. Useful timing marks (TDC, EPO, BDC, EPC) were added to the flywheel edge to indicate on a fixed pointer.

The standard cylinder head was replaced with one containing a rotary valve. This rotary valve was driven from the engine crankshaft with a roller chain and was fed compressed air at approximately 80 psi.

The valve was timed to open at engine TDC and close a few degrees before EPO. In this manner it was hoped to have a realistic cylinder pressure of approximately 70 psi. when the exhaust port opened. Provision was made in the new cylinder head for mounting a pressure transducer.

6. EXHAUST PULSE GENERATOR INSTRUMENTATION

Wave amplitude measurements in the exhaust pipe were made in the same manner as for the shock tube experiments. The scope horizontal time base trigger was set on single sweep and connected to the output trigger of an electronic stobe light. With this arrangement it was possible to trigger the horizontal time base at any desired point in the engine cycle, by pushing the timebase trigger reset switch. For example if a pressure-time history was required adjacent to the exhaust port with the trace starting exactly at EPO. With the appropriate pressure transducer connected to the scope vertical input and the pulse generator in operation, the strobe was started and focused on the flywheel pointer. The strobe frequency was adjusted until the EPO indicator on the flywheel appeared stationary next to the pointer. The timebase trigger reset switch was then pressed causing the time base to trigger on the next strobe flash. RPM readings could be read directly from the strobe or determined from the pressure trace by adjusting the time base to include at least one revolution.

EXPERIMENTAL PROCEDURE

1. PRESSURE TRANSDUCER CALIBRATION

Experience gained by the engineers conducting the I.S.V.R. shock tube experiments with similar pressure transducers, indicated that a dynamic rather than a static calibration curve must be used when dynamic measurements are made. For the sake of interest both calibrations were performed.

The static calibrations were performed with the transducers mounted in the shock tube. The shock tube open end was capped to form a pressure vessel. Transducer outputs were read from the scope and later checked with a precision DC. voltmeter.

The transducer output is related to the applied pressure by the following:

$$\frac{\text{AMPLIFIER RANGE (PCB/VOLT)}}{\text{TRANSDUCER SENSITIVITY (PCB/PSI)}} \times \text{OUTPUT (VOLTS)} = \text{PRES. (PSI)}$$

— (I)

As a check on the static calibrations, the transducer sensitivity was determined from (I) above using the measured output at 10 psi. The result was compared with the static calibration certificate supplied by Kistler.

The measurements necessary for the dynamic calibration were taken in the manner described previously in the section on shock tube instrumentation. From the measured transit time of the shock wave over

the known distance, the shock velocity was calculated from (II).

$$V = \frac{x}{t} \quad \text{---(II)}$$

From the measured temperature of the gas in the test section (air at room temperature) the local acoustic velocity was calculated from (III) assuming for air: $\gamma = 1.4$, $R = 1715 \text{ ft. lb. / slug, } ^\circ\text{R}$

$$a = \sqrt{\gamma R T} = 49.1 \sqrt{T} \quad \text{---(III)}$$

From the results of (II) and (III) the shock Mach number was calculated from (IV).

$$M = \frac{V}{a} \quad \text{---(IV)}$$

Knowing the shock Mach number, the shock pressure ratio P_2/P_1 was read from table 48 of (15) or with less interpolation from Table Ia of (10). Assuming $P_1 = 14.7 \text{ psi.}$ the shock wave amplitude was calculated from (v)

$$\Delta P = (P_2/P_1 - 1) 14.7 \quad \text{---(V)}$$

and plotted against the corresponding output voltage to obtain the dynamic calibration curve.

To check the accuracy of the shock pressure ratios determined for the dynamic calibration, the shock tube characteristic curve of P_2/P_1 vs. P_4/P_1 was plotted from the dynamic calibration results.

This curve was then compared with the theoretical characteristic curve obtained from the following relation, given in (16).

$$P_1/P_2 = P_2/P_1 \left[1 - \frac{(\gamma_1 - 1)(a_1/a_2)(P_2/P_1 - 1)}{\sqrt{2\gamma_1} \sqrt{2\gamma_1 + (\gamma_1 + 1)(P_2/P_1 - 1)}} \right]^{\frac{-2\gamma_1}{(\gamma_1 - 1)}} \quad \text{--- (VIa)}$$

For air at room temperature in both the driver and test sections (VIa)^a reduces to:

$$P_1/P_2 = P_2/P_1 \left[1 - \frac{.2389 (P_2/P_1 - 1)}{\sqrt{2.8 + (2.4)(P_2/P_1 - 1)}} \right]^{-7} \quad \text{--- (VIb)}$$

Information obtain from (17) indicated that good experimental correlation with (VIb) should be possible with the low shock Mach numbers and pressure ratios being used.

2. GRADUAL & ABRUPT AREA CHANGES

The amplitudes of the reflected and, where possible, the transmitted waves produced by the various test section combinations of Table II, were recorded and plotted for various incident shock pressure ratios. The resulting curves were compared with the theoretical predictions obtained from the I.S.V.R. charts, Figures 4, 5, and where applicable, with the experimental results reported in (12).

During the area expansion tests, a range of incident shock amplitudes of up to 15 psi. were used. Typical primary amplitudes of 7.5 psi. have been measured in engine exhausts. For the area contraction tests it was necessary to pass the incident shock through an area expansion first. As a result the maximum incident shock amplitude obtainable in the parallel section leading to the contraction was reduced to

approximately 8 psi. This amplitude reduction would occur in the actual exhaust system, thus the incident amplitude range was still realistic.

During the area contraction experiments it was necessary to extend the shock tube driver section to its full five feet, to delay the arrival of the rarefaction wave reflected from the closed end. It was also necessary to extend the parallel section downstream of the area expansions by two feet to delay the arrival of the rarefaction wave reflected from the open end.

3. EXHAUST PULSE GENERATOR

The generator was first operated with the cylinder head removed to study the level of mechanical vibration indicated by the pressure transducers in the exhaust pipe.

With the rotary valve in operation, pressure-time records were taken of the cylinder pressure at 970 and 1290 RPM to verify the correct duplication of the cylinder blowdown pressure.

Simultaneous pressure-time records were made at 970 RPM with one transducer adjacent to the exhaust port and another near the open end of the exhaust pipe. A similar record was made with the generator set at BDC and accelerated as rapidly as possible to 970 RPM. By this method a record of a primary exhaust pulse was obtained, unmodified by the reflections of previous waves. This result was compared with the previous record for continuous operation. The above procedure was repeated with the 8°, 7.111 area ratio expansion cone added to the end of the exhaust pipe.

DISCUSSION OF RESULTS

1. PRESSURE TRANSDUCER CALIBRATION

Typical static and dynamic calibration curves are shown in Figure 15. All readings had excellent repeatability, resulting in negligible point scatter. The transducer output was typically 7.5% higher under dynamic conditions than under static conditions. This phenomena was observed by engineers at the high velocity range of Computing Devices of Canada using Kistler transducers, and by the I.S.V.R. engineers using similar SLM transducers. In the latter case the dynamic outputs were approximately 40% greater than the static.

It was established by methods discussed in the next section that the velocity measurements from which the dynamic calibration pressures were calculated, were correct. Thus it was concluded that the voltage output of the transducer was higher in the dynamic case. No satisfactory explanation could be obtained for this phenomena with respect to the operation of the transducer or amplifier. The transducer resonant frequency of 60,000 cps. was well above the frequencies being measured.

It is possible that the method of mounting the transducers allows some of the dynamic pressure of the flow behind the normal shock to be sensed by the transducer. This would explain the nonlinearity of the dynamic curve. The transducer face is flat, 0.37" in diameter, and mounted tangentially to the tube bore. This results in a small recess on either side of center, offering a slight scoop effect to the gas flow. At a static calibration pressure of 10 psi., the 7.5% increase in

transducer output for the dynamic measurement, represents approximately 16% of the dynamic pressure of the flow behind the shock wave.

No details of the transducer mounts used in the I.S.V.R. experiments were available. However if the crystal faces were recessed to any extent from the tube bore a greater portion of the dynamic pressure might be observed, explaining the greater percent increase over static in their dynamic measurements.

No measurable difference was found between the calibration curves of the three transducers used. One of the three amplifiers was found to be slightly off calibration and thus was used for scope triggering purposes only.

The transducer sensitivity as determined from Eqn. (I) and the static calibration results, agreed with the static calibration certificate supplied by Kistler to within 0.5%.

Figure 16 shows the theoretical and experimental shock tube characteristic curves. The first attempt at verification resulted in shock pressure ratios (calculated from measured velocity and temperature) being approximately 3% less than the theoretical prediction. As a check on the shock velocity measurements various geometrical arrangements of the transducers measuring the time interval were tried. It was thought possible that the transducer starting the counter might distort the top portion of the shock wave causing it to lag slightly. This would cause a slightly longer time interval to be recorded when the lagging portion of the wave stopped the counter at the downstream transducer. However propagation velocities were unchanged when the downstream transducer was rotated 180° with respect to the upstream one.

As a further check on the velocity results, the distance between the diaphragm and the test section was increased to 32 diameters from 20. The measuring distance was also increased from 12" to 30", with the shock amplitude measurement taken from a transducer half way between. In both cases the shock velocities obtained were identical with the previous results. The tube distance between the measuring transducers was known to an accuracy of $\pm .001$ ". As a result of these tests, the shock velocity measurements and the dynamic calibration curve based on them, will have an error of the order of 0.5%.

It should be noted that the verification of the dynamic calibration curve was sufficient for the experiments conducted. In all tests the incident shock amplitude was obtained from the output of a pressure transducer a few diameters in front of the component being investigated. The diaphragm pressure was used only as a convenient reference for spacing and repeating readings. However for the sake of interest, the investigation of the characteristic curve deviation was continued.

Since the bourdon tube gauge from which the diaphragm pressures were recorded had been calibrated on a dead weight testers, the diaphragm pressure ratios of Figure 16 could be assumed as correct. It was known that the compressed air used to pressurize the shock tube driver section contained a certain amount of moisture and oil. If the γ and R of this mixture was significantly different from that of the air in the test section, equation (VIa) would not reduce to (VIb) and the theoretical curve of Figure 16 would not be correct.

To test this theory, new theoretical and experimental characteristic curves were plotted using pure, dry, nitrogen to pressurize the shock

tube driver section. Again the experimental results were approximately 3% less than the theoretical predictions.

It was finally suggested by Dr. R. E. Meyer, of the National Aeronautical Establishment, that the presence of the .25" diam. diaphragm lance in the 1.5" bore tube caused a decrease in driver cross-sectional area of 2.8%. Thus the theoretical characteristic equation (VIa) which is based on an area ratio of 1.0 across the diaphragm, was not valid in this application. Since the characteristic equation for a shock tube with an area increase at the diaphragm was not readily available, a new experimental curve was plotted with the diaphragm lance removed. Compressed air was again used in the driver and the diaphragms were scribed to obtain a range of natural burst pressures. It can be seen from Figure 16 that agreement was obtained with the original theoretical curve to within 0.85%.

Thus it was concluded that the experimental characteristic curve obtained with the diaphragm lance in place was sufficiently accurate.

2. TEST SECTIONS

Table III below contains dimensional details of the various test sections used in the experiments. For convenience these will be referred to by number in the following discussion.

TABLE III TEST SECTION DIMENSIONAL DETAILS

#	θ°	A_2/A_1	L (in.)	d (in.)	L/d
1.	8	3.362	8.95	10.73	.835
2.	8	7.111	17.90	10.73	1.67
3.	15	3.362	4.75	5.7	.835
4.	15	7.111	9.50	5.7	1.67
5.	180	7.111	-	-	-

(I) AREA EXPANSIONS

Figures 17 and 18 show a comparison of the experiment results with the one-dimensional theoretical predictions of figures 4 and 5. No significant differences were observed between the experimental results based on the two cone angles of 8° and 15° , (ie: between #1 and #3 and between #2 and #4).

The amplitude of the shock waves transmitted through the abrupt expansion #5 agreed closely with the theoretical prediction while the amplitudes for the gradual expansions were higher. At an incident wave amplitude of 7.5 psi. the transmitted wave amplitude produced by #1 and #3 was high by 5.6% while #2 and #4 produced results high by 38%, in comparison with theory.

For the reflected rarefaction wave amplitude, the results of the gradual expansions agreed closely with theory for incident wave amplitudes up to approximately 7 psi. However the results of the abrupt expansion #5 at this point were low by approximately 50%.

The sharp leveling off of the theoretical and experimental curves for the reflected waves can be explained by the occurrence of sonic flow at the entrance to the area change. At this point the amplitude of the reflected rarefaction wave is just sufficient to induce sonic flow behind the incident shock wave. Sonic flow occurs sooner experimentally than theoretically, due to the formation of a ring vortex at the entrance to the area change.

The formation of this vortex progressively reduces the tube flow area thus causing sonic conditions to be reached sooner than predicted by theory. The formation of this vortex ring, at abrupt and gradual area increases, is supported by Schlieren photos of two-dimensional models in (12).

It seems possible that the sharp transition of the abrupt area increases, encourages an earlier formation of the vortex ring than the smoother transition of the gradual area increases. This would explain the greater deviation from theory in the former case.

In general it can be seen that for transmitted waves, the results with the abrupt area increase agree more closely with the theoretical predictions than those with the gradual expansions. For reflected waves the opposite case is evident. The I.S.V.R. engineers reported in (12) that the pressure records obtained for abrupt and gradual expansion were "practically the same." Since the point scatter on their graphs was considerable it is possible that the differences observed in these experiments were not evident in theirs.

Table III contains a column of L/d ratios for the various cone combinations. This dimensionless ratio is suggested in (9) and appears to be a better correlation parameter than a simple area ratio, since both cone angle and length are involved. It is interesting to note that the 8° and 15° cones which produced similar results have the same L/d ratios. The use of " " in place of "d" would avoid confusion with other similar ratios (i.e. Lift/Drag ratio in aerodynamics).

Figures 19a and b show typical scope traces obtained for gradual and abrupt expansions respectively. Some trace "noise" is

evident in the transmitted and reflected pressure signals of figure 19b. This is due to mechanical vibrations in the shock tube being picked up by the pressure transducers. In general these vibrations did not present a problem, since only in the abrupt expansion experiment did they reach significant proportions before quickly decaying.

In the I.S.V.R. experiments considerable "ringing", due to mechanical vibrations, was experienced with the transducers rigidly mounted. To overcome this the I.S.V.R. engineers isolated the pressure transducers from the shock tube with rubber "O" rings. Even with this arrangement the mechanical vibration level was higher than in these experiments where all transducers were rigidly mounted. Since the mechanical details of the I.S.V.R. shock tube, other than its 1.375" bore, were not known, no reasons can be given for the observed reduction in "ringing".

A comparison of the reflected wave amplitudes of figures 19a and 19b will emphasize the advantages of using a gradual rather than an abrupt area expansion to produce a strong "scavenging wave". The time interval over which wave reflection occurred was considerably longer in the case of the gradual expansion and was, in general, proportional to the length of the cone.

The strong compression wave evident at 4.6 ms. in the upper trace of figure 19a, is due to double reflections within the expansion cone. This effect was more pronounced in the 15° cones than in the 8° cones.

In all the scope traces presented, any unexplained rarefactions observed after the initial transmitted and reflected waves, are due to reflections from the open or closed ends of the shock tube.

(II) AREA CONTRACTIONS

Figures 20 and 21 show a comparison of the experimental results with the theoretical predictions. In all but one instance, no significant differences were observed between the experimental results based on the cone angles of 8° and 15° (ie: cones of the same L/d ratio). The exception was in the reflected compression wave amplitudes obtained from cone #4, figure 21, which were less than those of either the more gradual contraction #2 or the abrupt contraction #5. It would be reasonable to assume that the reflected wave amplitude curve of cone #4 should fall between those of #2 and #5, which in fact were almost identical. Further investigation of this phenomena would be worthwhile.

For test sections #1, 2, 3, and 5 the reflected wave amplitudes obtained were less than the theoretical predictions by approximately 15%. It is possible that this deviation is due to frictional attenuation.

For the small area ratio test sections, #1 and #3, the transmitted wave amplitudes obtained agree closely with the theoretical predictions. However for the large area ratio test sections, #2 and #4, the transmitted wave amplitudes were approximately 7% higher.

In contrast to the above, the transmitted wave amplitudes of the abrupt area contraction #5, were approximately 17% lower than the theoretical predictions. This can be explained by the formation of a ring vortex at the entrance to the contraction outlet pipe. The small area expansion downstream of this vortex would cause an additional reflection and attenuation of the transmitted compression wave. The amplitude of the wave reflected by the abrupt contraction was the same as that reflected by the 8° gradual contraction #2. However, as for gradual expansions, the time duration of the wave reflection process of the gradual contraction is approximately proportional to its length. The reflected compression wave will tend to steepen as it travels up the exhaust pipe and since it is desirable to have the "ram-charging wave" spread over at least 20° of crank angle, the gradual contraction would be more suitable than the abrupt.

Figures 22a and b show typical scope traces obtained for gradual and abrupt contractions respectively. The increase in the wave reflection time interval caused by the gradual contraction is evident.

In general it would appear that the chief reason for the preference of 3° expansions and 15° contractions, is cone length rather than cone angle. For a given area ratio, the 8° cone is longer and thus has a longer reflection time interval than the 15° cone. This fact is associated with the requirement that the desirable time interval of the "scavenging wave" reflected by the expansion cone, be longer than the time interval of the "ram-charging wave" reflected from the contraction cone.

(III) OPEN END

Figure 23 shows a comparison of amplitudes obtained from a plain open end, with the theoretical prediction. For interest, the reflected wave amplitudes resulting from the addition of the four expansion cones to the plain open end, were plotted on the same graph.

For the plain open end the experimental correlation with theory is good up to incident shock amplitudes of approximately 10 psi. As with the abrupt area expansion the formation of a ring vortex causes sonic flow to occur at the pipe outlet earlier than predicted by the theory.

It is interesting to note that the addition of a conical diffuser on the open end in no way shifts the point where sonic conditions occur. This fact was also observed for the gradual and abrupt expansions.

It appears from figure 23 that the total reflected wave amplitude increases with increasing cone length. However since this trend was not evident in the gradual expansion experiments it may in some way be related to the superposition of the reflections from the gradual expansion and the cone open end.

Figure 24a shows a comparison of the scope traces obtained for a conical pipe end and a plain pipe end. The gentle slope of the upper trace from 1.6 to 4.6 ms. represents the rarefaction reflected from the gradual expansion. The steep portion following this is the additional rarefaction from the open end. The steep compression wave at 5.9 ms. is due to a double reflection within the cone, as observed on the gradual expansion traces.

3. EXHAUST PULSE GENERATOR

Very little time was available for the adjustment and testing of the exhaust pulse generator. However, what information was obtained will be useful for future experiments on the apparatus.

Unfortunately, for the few experiments performed, a compromise compressed air supply was used. As a result the air flow rate was insufficient, even at 970 RPM, to maintain the 70 psi. cylinder pressure until EPO. Cylinder pressures of approximately 10 psi were obtained at EPO resulting in incident compression wave amplitudes of approximately 1.3 psi.

Due to the low RPM and low cylinder blowdown pressure, the incident compression waves did not steepen into shock waves within the 4 ft. of exhaust pipe used. Because of this, a close correlation with the shock tube experiments was not possible. Another difficulty in direct correlation was the fact that the rarefaction wave propagated back into the cylinder and reflected off the wall followed immediately behind the incident compression wave. In the shock tube it was desirable to separate these two waves by providing a longer driver section.

Figure 24b shows a scope trace obtained from two pressure transducers mounted in a plain ended parallel exhaust pipe, with the generator operated continuously at 970 RPM. It can be seen that there is little wave attenuation between primary exhaust pulses. Figure 25a shows scope traces made under the same conditions with the exception that the generator was accelerated from rest through one cycle, before EPO. The right hand portions of the traces indicate the negligible degree of mechanical vibration picked up by the transducers. Figure

25b was made under the same conditions as 25a with the addition of a conical expansion on the exhaust pipe end.

Figure 26 shows a superposition of the lower traces of 24b and 25a for the first few reflections. The extent to which the reflections from previous exhaust pulses modify succeeding ones is evident. With hot exhaust gases the wave speed would be considerably increased causing faster attenuation but this would be offset somewhat by the higher engine RPM.

In Figure 25b the greater amplitude of the reflected rarefaction wave available from the conical diffuser, is evident. However the relatively slow horizontal time base used obscures the increased duration of the primary reflection available from the cone.

In general, except for the amplitude and time scales, the pressure traces of Figure 25a and b compare reasonably well with published light spring indicator diagrams taken in actual two-stroke exhaust systems (3,4).

CONCLUSIONS

1. SHOCK TUBE

- (I) The shock tube and its associated instrumentation performed satisfactorily as a generator of single pulses for detailed study of the effect of wave shaping.

2. TEST SECTIONS

- (I) Gradual rather than abrupt expansions and contractions are desirable for exhaust pulse "scavenging" and "ram-charging".
- (II) The reflected wave amplitude predictions of the I.S.V.R. one dimensional exhaust pulse theory agree more closely with the results of the gradual rather than the abrupt expansions.
- (III) The reflected wave amplitude predictions agree reasonably well with the results of both the abrupt and gradual contractions.
- (IV) There appeared to be no difference, with respect to reflected wave amplitudes, between the 8° and 15° expansion or contraction cones, of the same area ratio.
- (V) In general the cone reflection time interval was proportional to its length.

3. EXHAUST PULSE GENERATOR

- (I) The exhaust pulse generator gave a good simulation, on a reduced amplitude scale, of the two-stroke exhaust cycle.
- (II) It would appear that each primary exhaust pulse can not be considered independent by ignoring the reflections of previous pulses.

SUGGESTED FUTURE EXPERIMENTAL PROGRAM

1. To study wave shaping by a greater variety of expansions and contractions with the aim of obtaining a more complete picture of reflected wave amplitudes on the basis of cone angle, area ratio and L/d ratio.
2. To study the "ram-charging" compression wave, reflected from the downstream contraction, as it travels back through the upstream expansion. This will require modification of the shock tube.
3. To obtain and instrument a two-stroke engine similar to the exhaust pulse generator. The engine should be able to operate normally on fuel with provision for power measurement, or be driven externally as an exhaust pulse generator.
4. To investigate, using the above, the feasibility of a variable geometry exhaust system to extend the engine RPM range over which the pipe tuning would be effective.

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FIG.1 THE CONVENTIONAL TWO-STROKE
ENGINE

FIG.2 TYPICAL TWO & FOUR-STROKE ENGINE TIMING

FIG. 3a WAVE REFLECTION PROCESS FOR A
PLAIN, PARALLEL, TWO-STROKE
EXHAUST PIPE

FIG. 3b WAVE REFLECTION PROCESS FOR A
DIVERGENT-CONVERGENT, TWO-STROKE
EXHAUST PIPE

FIG.4a Compression wave transmitted through an expansion by an incident compression wave.

(REPRODUCED FROM REFERENCE 12)

FIG.4b Compression wave transmitted through a contraction by an incident compression wave.

(REPRODUCED FROM REFERENCE 12)

FIG. 5 Pressure waves reflected from expansions and contractions by incident pressure waves.

(REPRODUCED FROM REFERENCE 12)

FIG. 7 SHOCK TUBE DRIVER SECTION

1. SHOCK TUBE DRIVER
2. DIAPHRAGM CHUCK
3. ADJUSTABLE TUBE SUPPORT
4. QUICK-LOCK AIR SUPPLY CONNECTOR
5. NEEDLE VALVE (PRESSURIZE)
6. NEEDLE VALVE (DEPRESSURIZE)
7. PRESSURE GAUGE
8. DIAPHRAGM LANCE
9. LANCE RELEASE COMPONENTS

FIG. 8 TYPICAL BURST DIAPHRAGM

SCALE:	FULL SIZE
MANUAL BURST PRESSURE:	25 PSI.
MATERIAL:	.002" THICK, C.I.L. CELLOPHANE
MAX. NATURAL BURST STRENGTH:	40 PSI. APPROX.

TEST SECTION NO.	A	B
1	17.90"	8"
2	9.50"	15"

FIG. 9
EXPANSION & CONTRACTION
TEST SECTIONS

FIG.10 SHOCK TUBE TEST SECTIONS

1. PARALLEL SECTION, 12" x 4.00 I.D.
2. 8° TAPER SECTION
3. 15° TAPER SECTION
4. PARALLEL SECTION, 12" x 2.75 I.D.
5. PARALLEL SECTION, 36" x 1.50 I.D.
6. PARALLEL SECTION, 24" x 1.50 I.D.

FIG. 11 SHOCK TUBE INSTRUMENTATION

1. ELECTRONIC COUNTER, HEWLETT PACKARD MODEL 522B, (10^{-5} SEC.)
LATER REPLACED WITH, TRANSISTOR SPECIALITIES INC.
MODEL 365 AR, (10^{-7} SEC.)
2. TYPICAL TRANSDUCER MOUNT
3. KISTLER, MULTI-RANGE, CHARGE AMPLIFIERS, MODEL 502
4. TEKTRONIX TYPE 564, 4 TRACE, STORAGE OSCILLOSCOPE
5. POLAROID CAMERA ATTACHMENT, TYPE T-12
6. PRESSURE GAUGE

FIG. 12 PRESSURE TRANSDUCER & AMPLIFIER

1. KISTLER, QUARTZ, PRESSURE TRANSDUCER MODEL 701A
2. TYPICAL TRANSDUCER MOUNT
3. TRANSDUCER CHARGE AMPLIFIER

FIG. 13 EXHAUST PULSE GENERATOR

1. ELECTRIC MOTOR, 1/2 HP.
2. STEP PULLEY AND "V" BELT DRIVE
3. MODIFIED VILLIERS, 12 CU. IN., MODEL 6E, TWO-STROKE ENGINE
4. AIR INTAKE
5. EXHAUST PIPE (SHOCK TUBE SECTIONS, 1.50" BORE)
6. QUICK-LOCK AIR SUPPLY CONNECTOR AND VALVE
7. CHAIN DRIVEN ROTARY VALVE
8. FLYWHEEL
9. ENGINE CYCLE EVENT INDICATOR

FIG. 14 EXHAUST PULSE GENERATOR

**FIG. 15 PRESSURE TRANSDUCER
CALIBRATION CURVES**

CHARGE AMPLIFIER S/N 585
RANGE 100 PC8/VOLT

TRANSDUCER OUTPUT (VOLTS)

DYNAMIC

STATIC

PRESSURE (PSI.)

FIG 16 SHOCK TUBE CHARACTERISTIC CURVE

TRANS. COMPRESSION WAVE AMP. (PSI.)

REF. RAREFACTION WAVE AMP. (PSI.)

FIG. 18 GRADUAL & ABRUPT EXPANSIONS,
REF. & TRANS. WAVE AMPLITUDES
($A_2/A_1 = 7.111$)

FIG. 19 SCOPE TRACES, GRADUAL & ABRUPT EXPANSIONS

(a) GRADUAL EXPANSION #4

$A_2/A_1 = 7.111$	$\theta = 15^\circ$	$P_2/P_1 = 1.442$
HORIZ. = 1 MS./CM.	VERT. = .1 VOLT/CM.	
UPPER TRACE:	INC. & REF. WAVES	
LOWER TRACE:	TRANS. WAVE

(b) ABRUPT EXPANSION #5

$A_2/A_1 = 7.111$	$P_2/P_1 = 1.442$
HORIZ. = 1 MS./CM.	VERT. = .1 VOLT/CM.
UPPER TRACE:	INC. & REF. WAVES
LOWER TRACE:	TRANS WAVE

FIG. 20 GRADUAL CONTRACTION, REF. & TRANS.
WAVE AMPLITUDES
($A_2/A_1 = 3.362$)

FIG. 21 GRADUAL & ABRUPT CONTRACTIONS,
REF. & TRANS. WAVE AMPLITUDES
($A_2/A_1 = 7.11$)

FIG. 22 SCOPE TRACES, GRADUAL & ABRUPT CONTRACTIONS

(a) GRADUAL CONTRACTION #3

$A_2/A_1 = 3.361$ $\theta = 15^\circ$ $P_2/P_1 = 1.265$
HORIZ. = .5 MS./CM. VERT. = .1 VOLT/CM.
UPPER TRACE: INC. & REF. WAVES
LOWER TRACE: TRANS. WAVE

(b) ABRUPT CONTRACTION #5

$A_2/A_1 = 7.111$ $P_2/P_1 = 1.170$
HORIZ. = .5 MS./CM. VERT. = .1 VOLT/CM.
UPPER TRACE: INC. & REF. WAVES
LOWER TRACE: TRANS. WAVE

**FIG 23 REF. WAVE AMPLITUDE FOR OPEN END,
PLAIN & WITH CONE**

θ°	A_2/A_1	L°
0	3.362	8.95
5	7.11	12.5
15	7.11	9.5
15	3.362	4.75

--- THEORY FOR PLAIN END

REF. RAREFACTION WAVE AMP. (PSI.)

INCIDENT SHOCK WAVE AMP. (PSI.)

FIG.24a SCOPE TRACES, OPEN END

$P_2/P_1 = 1.233$ HORIZ. = 1 MS./CM. VERT. = .1 VOLT/CM.
 UPPER TRACE: INC. & REF. WAVE FROM OPEN END WITH GRADUAL EXPANSION #2
 ($A_2/A_1 = 7.111$, $\theta = 8^\circ$)
 LOWER TRACE: INC. & REF. WAVE FROM PLAIN OPEN END

**FIG.24b EXHAUST PULSE GENERATOR
 (CONTINUOUS OPERATION)**

RPM = 970 HORIZ. = 10 MS./CM. VERT. = .05 VOLT/CM.
 UPPER TRACE: 18" FROM PLAIN OPEN END
 LOWER TRACE: 5" FROM EXH. PORT

FIG. 25 EXH. PULSE GENERATOR (STARTED FROM REST)

(a) EXHAUST PIPE WITH PLAIN OPEN END

RPM ~ 970 HORIZ. = 10 MS./CM. VERT. = .05 VOLT/CM.
 UPPER TRACE: 13" FROM PLAIN OPEN END
 LOWER TRACE: 5" FROM EXH. PORT

(b) EXHAUST PIPE WITH GRADUAL EXPANSION #2 ($A_2/A_1 = 7.111$, $\theta = 8^\circ$)

RPM ~ 970 HORIZ. = 10 MS./CM. VERT. = .05 VOLT/CM.
 UPPER TRACE: 13" FROM CONJ. ENTRANCE
 LOWER TRACE: 5" FROM EXH. PORT

FIG.26 PRESSURE-CRANK ANGLE HISTORY AT
PULSE GENERATOR EXHAUST PORT

(4 FT., PLAIN END, EXHAUST PIPE ; 970 RPM)

—— STARTING FROM REST (LOWER TRACE FIG.25a)

----- CONTINUOUS OPERATION (LOWER TRACE FIG.24b)