

AN INVESTIGATION OF THE IGNITION
OF HYDROCARBON-AIR MIXTURES IN
AN ADIABATIC COMPRESSION MACHINE .

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ABSTRACT.

In connection with the problem of accidental explosions in pneumatic pipe lines which are contaminated with oil, an investigation has been made of the mechanism and phenomenon of the ignition of oil films in contact with air heated by sudden compression. For this purpose a single stroke rapid compression machine, which can create reproducible temperature and pressure histories in the compressed material was designed and constructed.

The compression machine, driven by compressed nitrogen, has a minimum compression time of 6 msec.; and its compression ratio is variable up to 30:1. The instrumentation of the apparatus is such that a record can be made of the pressure and piston displacement histories during and after the compression stroke. An optical device of novel design which employs a photomultiplier tube is used to measure the piston displacement. When compressing dry air, the adiabatic efficiency of the machine is in the region of 95%.

New measurements were made of the ignition delay periods of compressed n-heptane vapour-air mixtures under various conditions. The results for stoichiometric mixtures are in good agreement with those published by other investigators using different compression machines.

When films of n-decane with air were compressed the characteristics of the combustion process were found to conform to one of three different types of behaviour depending on the initial temperature of the film and the compression ratio. The different types of result arise from the fact

that combustion of the liquid film is initiated by ignition of the vapour above the liquid. The criterion of whether ignition will occur under certain conditions is chiefly determined by the combination of the temperature and pressure history of the compressed vapour above the liquid, the limit of inflammability of the vapour, and the rates of heat and mass transfer between the liquid and the vapour. The investigation of these factors will form a basis for future studies.

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CHAPTER 1.1. INTRODUCTION.

During the past decade widespread use has been made of high pressure pneumatic and hydraulic control systems in which it is possible for lubricating oil, from the air compressors, or hydraulic oil to collect in the pneumatic pipe line. Subsequent investigations of several disastrous explosions which have occurred on board ship have revealed that the explosions could have been caused by the sudden release of high pressure air into a low pressure pneumatic pipe line which is contaminated with oil. In such circumstances the heat generated by the shock wave resulting from the rapid expansion of the air may have been sufficient to ignite the oil and initiate the explosion^(17,18). Although this hazard exists in many industrial applications, it is particularly important on board ship where the high ambient temperatures, which occur locally, increase the likelihood of ignition taking place.

To investigate the practical aspect of this problem, King and Coil⁽¹⁷⁾, Feath and White⁽⁹⁾ and the Electric Boat division of General Dynamics Corp.⁽¹¹⁾ have constructed experimental pipe rigs with rapid opening valves to simulate the condition under which the accidents occurred. The experimental rigs consisted essentially of a reservoir, containing air at pressures up to 5000 psi, connected via a rapid opening valve to a low pressure pipe section, which was closed at the opposite end. Oil was introduced into the low pressure pipe either by impregnating a fibre glass wick or by wetting the walls. Tests carried out by releasing high pressure air into the low pressure pipe end confirmed that

ignition of oils was likely under such conditions.

A surprising result of these experiments was that it was possible to ignite phosphate esters and water based fluid as well as petroleum oils. Apart from the nature of the fluid, other factors influencing its ignitibility were the release pressure, the initial temperature of the fluid, the rate of pressure rise in the low pressure section, the pipe dimensions and the manner in which the oil was introduced into the pipe. For example, work carried out at the Electric Boat Division of G.D.C.⁽¹¹⁾ showed that for a given valve opening time, ignition of a hydrocarbon oil occurred in a 2 in. diameter pipe 18 ft. long when air pressure was released into it from 150 psi to atmospheric; whereas White⁽⁴³⁾ found that a $\frac{3}{8}$ in. diameter pipe, 5 ft. long, required a release pressure 10 times higher for ignition to occur with the same oil. In another example, White observed flame propagation velocities in the order of 60 ft./sec. in a $\frac{3}{8}$ in. diameter and 18 in. long tube with its walls wetted with a lubricating oil, whereas Loison⁽²⁹⁾ observed stable flame propagation velocities of 1200 m/sec. in a 250 mm. diameter tube 100 m. long and wetted with a lubricating oil film 0.3 mm. thick.

Experiments such as these have provided valuable qualitative data on the behaviour of various oils in particular test rigs, but in practice it is very difficult to predict what will happen if, for example, the characteristics of the quick opening valve or the geometry of the rig is changed. Thus, although experiments of this type may provide useful

information, on which the design of pneumatic systems may be based, the physical conditions of the experiments hamper an investigation of the mechanism of ignition. These conditions include the aerodynamic effect of the gas flow in the pipe system; the unsteady temperature and pressure conditions arising from the reflection and attenuation of the pressure waves; and uncertainties in the maximum values and durations of the pressure and temperature pulses at any point.

The work described in this thesis was directed towards providing basic ignition data which can be used to predict whether or not ignition will occur in any system in which the pressure and temperature ~~histories~~ of the pulse are known at all points throughout the system. To this end, it was decided to build a single stroke rapid compression machine in which the fuel in the form of a gas, liquid or a foam may be subjected to an accurately known temperature and pressure history.

CHAPTER 2.SURVEY OF METHODS USED FOR RAPID COMPRESSION.

The first compression apparatus known to have been constructed for ignition studies of gas mixtures was described by Falk⁽⁸⁾ in 1906. The gas mixture was contained in a cylinder fitted with a piston and rapidly compressed by dropping a weight on to the piston. Means were provided to mark the maximum compression position of the piston; so that when ignition occurred, the temperature and pressure conditions could be calculated assuming that the compression was adiabatic. In 1914, Dixon, Bradshaw and Campbell⁽⁵⁾ carried out similar experiments in a stout glass tube instead of a steel cylinder and recorded the ignition process by high speed photography. They discovered that with some hydrocarbon-air mixtures, there is an ignition delay period which occurs after the temperature and pressure to cause ignition have been attained. It was then realised that in order to obtain accurate values of the minimum temperature needed to cause ignition at any particular pressure, it was necessary to arrest the piston motion when the pressure was attained. The ignition temperature obtained by Falk tended to be higher than the correct values because of the delay effect. At the time when ignition occurred, the piston had travelled further than the position corresponding to the ignition temperature.

It was for this reason that later investigators constructed machines in which the motion of the piston was arrested in the position of maximum compression. Their various techniques which enabled this to be done are

briefly described in the following sections.

2.1. Driving Methods.

2.1.1. By falling weights.

The method, first employed by Falk, in which a falling weight was used to drive the compression piston was subsequently adopted by Jost and Teichmann⁽⁴⁰⁾ and Kröger and Gondermann⁽¹⁹⁾. In these investigations, weights of up to 120 kg. falling through distances of a few metres were used to generate an 18:1 compression ratio in a cylinder of 53 mm. bore; the compression time being in the order of 0.05 sec. Springs were used to damp out shocks arising from the impact of the weight hitting the piston. The weights required to achieve compression ratios greater than 20:1 are so heavy that the method is impracticable if a rapid compression time less than about 20 msec. is required.

2.1.2. By the inertia of a flywheel.

Tizard and Pye⁽⁴¹⁾ compressed gases in a cylinder of $4\frac{1}{2}$ inch diameter with a piston travelling over a stroke of 8 inches. The piston was supported by two links connected in series. The lower end of the links was fixed to a pivot so that when the two links were in line in a vertical position, the piston was at the end of the stroke. The joint between the two links was connected to a rotating crank via a compound link which incorporated a ratchet mechanism. This was so arranged that the compounded link would transmit a pushing force only; when pulled, the ratchet enabled the link to extend in a telescopic manner. The driving force for the system was derived from a heavy rotating flywheel

driven by a motor. In order to operate the machine, a friction clutch was engaged suddenly to rotate the crank and drives the linkage system until the maximum compression position was attained. On the return stroke of the crank, the ratchet mechanism enabled the piston and supporting links to remain at rest at the end of the compression stroke.

The machine was normally operated in such a manner as to compress the gas in 0.14 sec. When the compression times of less than 0.05 sec. were attempted, difficulties were experienced and the moving parts tended to fracture because of the larger inertia effect at greater speeds.

2.1.3. By expansion of Compressed Gas.

In recent years, many investigators including Scheuermeyer and Steigerwald⁽³⁵⁾, Jost and Rögner^(15,33), Martinengo and Wagner^(31,32), Taylor et al⁽³⁹⁾, Beckers⁽²⁾, Havemann⁽¹²⁾ and Rogowski⁽³⁴⁾ have used pneumatic drive systems employing compressed air or nitrogen. In most cases, the pressure of the compressed gas is allowed to act on a drive piston which is connected by a rod to the compression piston.

The advantage of this system is that the total mass of moving parts is small, consequently high piston speeds and short compression times can be achieved. Taylor et al^(39, 16) achieved compression times of 6 msec. for a piston stroke of three inches. Limitations of the maximum gas pressure and the size of drive piston imposed by practical considerations are not severe, and it is possible therefore to generate large compression forces to achieve high compression ratios by this method.

2.2. Braking Methods.

As greater piston speeds are achieved, problems arise in decelerating the piston in as short a time as possible without causing any damage to the apparatus. In the following sections the methods used by various investigators are briefly described.

2.2.1. By spring force.

Jost and Teichmann⁽⁴⁰⁾ made use of stiff springs to absorb the kinetic energy of the moving piston until the piston came to a standstill. Mechanical claws were then released to prevent the piston from bouncing.

2.2.2. By deformation of Metal.

In the machine developed by Scheuermeyer and Steigerwald, the passage of an enlarged part of the piston rod was obstructed by plates of aluminium located at a position near the end of the stroke. Since the kinetic energy of the moving piston was absorbed in plastically deforming the metal, it was necessary to extract and replace the damaged plates with new ones after every run.

2.2.3. By transfer of momentum.

Rögener^(15, 33) and Martinengo and Wagner^(31, 32) arrested the piston motion by allowing the moving parts to impact on a stationary mass M which was either suspended on springs or balanced on an air cushion. If the mass of the moving part is also M , and if the coefficient of restitution for the impacting materials is unity then, after impact, the piston parts remain stationary. In practice the coefficient of restitution was less than unity, because the impact faces had to be padded with a soft material

in order to prevent fracture of the metal parts. It was found that the period during which the piston can be held stationary after impact is dependent on the mass of the moving parts and the magnitude of the compressed gas pressure.

2.2.4. By viscous drag.

Rogowski⁽³⁴⁾ connected a sharp edged snubbing piston to the rod which connected the drive and the compression pistons in his compression ignition machine. The snubbing piston was immersed in hydraulic oil contained in a cylinder, the bore profile of which was shaped so that near the end of the stroke the clearance between the bore and the snubbing piston was very small. Very high snubbing forces were generated by the viscous drag of the oil which was forced through the small clearance. It was found that by suitably shaping the bore profile, the correct snubbing force could be generated so as to retard the piston rapidly at a predetermined position.

However, as a separate chamber is necessary for the hydraulic snubber, this arrangement suffers from the disadvantage of being bulky. Since the moving parts have a greater mass the compression is not so rapid unless provision is made to store more energy in the compressed driving gas. Although Smith's⁽³⁷⁾ arrangement for using an hydraulic snubbing device only during the final part of the stroke avoids the disadvantage, his machine did not appear to have been very successful⁽²²⁾.

2.2.5. Isolation by a valve.

Havemann⁽¹²⁾ did not attempt to arrest the piston motion, but instead arranged for a valve to isolate the combustion chamber from the

compression piston when maximum pressure was attained. The system requires an intricate valve operating mechanism. Furthermore, not all the compressed gas can be forced into the combustion chamber.

2.2.6. By gas pressure.

Taylor and co-workers⁽³⁹⁾ designed their machine so that gas trapped below the drive piston was compressed during the stroke. Thus the gas pressure below the piston increased and retarded the piston until its velocity was reduced to zero at the end of the stroke. The compressed gas was then released from the machine through radial ports in the cylinder wall, so that the expanded driver gas pressure could be used to hold the piston in position. The gas in the combustion cylinder was compressed by a piston connected to the drive piston by a rod. This method has the advantage of allowing the machine to be compact and the mass of the moving parts small. By choosing a high ratio for the compression of the gas below the drive piston, the damping force retarding the piston can be generated rapidly at the end of the stroke. This enables a high piston speed to be achieved and a rapid deceleration at the end of the stroke.

2.3. Conclusion from survey:

A comparison of the driving and braking methods described in this section shows that the main differences between them are in the compression times over which the machines are capable of operating successfully, and the velocity displacement characteristics of the pistons. Of special importance is the compression speed just before the completion of the stroke.

It has been pointed out by Jost⁽¹⁵⁾ that a high piston speed is desirable near the end of the stroke so that the final adiabatic temperature arising from compression of the gas is attained in as short a time as possible. This feature is essential when studies of ignition delay are being made with compression apparatus.

Table I : Survey of compression ignition machines.

Investigators	Driving method.	Braking method.	Shortest compression time msec.	Velocity characteristic near end of compression.
Teichmann	by falling weight	by spring force.	50	gradual slow down.
Tizard and Pye.	flywheel inertia.	by linkage.	50	slow down as in reciprocating engine.
Martinengo	gas expansion.	impact & transfer of momentum.	15	almost instant stoppage.
Scheuermeyer and Steigerwald.	gas expansion.	deformation of metal.	15	fast retardation over last 9% approx. of stroke.
M.I.T. Taylor et al.	gas expansion.	gas compression.	6	fast retardation over last 20% approx. of stroke.

From Table I, in which the characteristics of the various machines have been summarised, it may be seen that the use of compressed gas as driving force results in the shortest compression times.

The machines constructed by Martinengo and Taylor et al at the M.I.T. are superior to the others because with Martinengo's machine, high piston speeds are maintained until the end of the stroke, while the M.I.T. machine is capable of very short compression times. Although in the latter machine, the piston is retarded over a larger portion of the stroke, the final rate of gas temperature rise may be comparable to that in the other machine because of the short compression time. After compression is completed, there is a positive downward force in the M.I.T. machine to retain the piston in place. This is an advantage because, with Martinengo's machine, the explosion pressure of the ignited gas may cause the piston to be displaced from its end position.

The method of braking the piston by compression of damping gas enables a lighter form of piston construction to be used which is essential when high piston speeds are required. Furthermore, the inevitable high acceleration and deceleration of the moving piston result in smaller forces which can be adsorbed more easily when the construction is lighter.

As a result of the survey, it was decided to utilise the method of driving by gas expansion and braking by compression of damping gas for the compression machine to be designed in this project.

CHAPTER 3.

DESIGN OF COMPRESSION MACHINE.

3.1. Basic design requirements.

It was envisaged that the machine would be required to compress fuel-air mixtures, either in the form of vapour or combinations of liquid and vapour, and to raise the temperature sufficiently to ignite the mixture. Consequently, the mixture should remain in a compressed state for a convenient length of time so that experimental observations can be made.

From these considerations, the following requirements were specified.

3.1.1. Compression Ratio.

The piston action is to be of a single stroke type and the piston should remain in the final compressed position at the end of the stroke without bounce. The mixtures to be ignited include n-heptane-air mixtures which has a spontaneous ignition temperature in the region of 260°C .⁽³⁰⁾ or benzene-air mixtures which ignite in the region of 600°C .⁽⁴⁾ In addition the machine must be capable of igniting fuel-air combinations with the fuel existing in liquid form and having a latent heat of vaporization of not less than 100 Btu/lb and an ignition temperature

in the region of 600°C . It is shown in appendix A that by assuming $\gamma=1.3$, the compression ratio of the machine should cover a range from 7:1 to 30:1 in order to fulfill these requirements.

3.1.2. Compression time.

To reduce the heat lost through the cylinder walls during compression, it would be desirable to achieve a pressure rise in the form of a step function such as that produced by a shock wave.

In a piston compression machine, mechanical difficulties prevent the attainment of this ideal, because the acceleration forces would fracture the moving parts. The survey in section 2 revealed that of the results obtained with existing machines the best compression time achieved was 6 milliseconds⁽³⁹⁾.

One of the aims in designing the present machine was to achieve a comparable compression time and to arrange for the period during which the piston is decelerated to be as short as possible.

3.1.3. Compression piston diameter and piston stroke.

Since the compression space in the machine should be accessible by a pressure transducer or other measuring probes, a 1.5 in diameter was thought to be the smallest

dimension to provide sufficient space.

The length of the compressed volume directly affects the length of piston stroke which in turn influences the total height of the machine structure. For a given length of clearance space, the ~~higher~~ the required compression ratio, the longer is the piston stroke and the height of machine structure.

A final compressed volume in which the length is equal to the diameter would be ideal because such a shape has a minimum surface to volume ratio and therefore minimum heat loss to the walls. Unfortunately such an arrangement requires a piston stroke of 43.5 in. for a piston diameter of 1.5 in. and a compression ratio of 30:1. The overall height of a machine to accommodate such a long stroke would be 9 ft or more.

Therefore in order to avoid excessively large machine structures, it was necessary to compromise and specify an average length for the clearance space of 0.5 in. which is a third of the piston diameter.

3.1.4. Maximum compression pressure.

When calculating the maximum pressure developed as a result of combustion, there are three cases to be considered. Firstly, normal ignition of a known vapour, for example

n-heptane-air or benzene-air mixtures. For a typical experiment in which the adiabatic compression pressure is 300 psia, and temperature is 600° K, the calculated ratio of the final to the initial pressure for a stoichiometric mixture of n-heptane and air, neglecting dissociation of combustion products, is 5.5 to 1. (see appendix B & C) Thus in this case, the maximum pressure developed in the combustion cylinder would be 1670 psia.

Secondly, if a liquid and vapour mixture initially at atmospheric pressure is subjected to a compression ratio of 30:1, the final compressed pressure will be approximately 100 atmospheres. If now an ignition occurs, the pressure increase of between 3 and 4 fold will result in a maximum pressure of the order of 5000 psia.

Thirdly, if when compressing an ignitable mixture to a pressure of 100 atm a detonation is to occur, then the transient pressure resulting from detonation may probably be as high as 1000 atm. for a very short duration.

The combustion cylinder should therefore be designed to withstand elastically a maximum working pressure of 5000 psia. Furthermore its bursting pressure, and dynamic loading conditions, should be such that a detonation will not cause the cylinder to fracture.

3.1.5. Filling arrangement and temperature control.

Arrangement should be such that the following fuels can be admitted into the combustion cylinder.

- a). Single phase multicomponent gas mixtures.
- b). Two phase mixture with the fuel in the form of thin films.

Provision should be made to pass the gas mixtures, prepared in a stainless steel vessel under pressure, into the evacuated compression cylinder by a pipe connection. Therefore the seals of the compression piston should be designed to withstand a vacuum as well as positive pressure.

For the vapour/ liquid phases, it should be possible to introduce the film of liquid into the cylinder by detaching the cylinder head.

Temperature control should be such that the temperature of the compression cylinder and piston can be regulated to within $\pm 0.25^{\circ} \text{C}$.

3.1.6. Experimental observation needed during course of experiment.

- a). Pressure measurement.

It is necessary to measure the pressure in the compression cylinder before and during compression, as well as during that period after compression when combustion may take place.

- b). Piston displacement measurement.

Observation of the piston displacement-time history are required to evaluate the apparent adiabatic exponent during compression, and also to ascertain whether or not the piston remains stationary after compression.

The machine was designed so that future modifications of the compression section could be made with the minimum amount of work. The possible modifications which were considered are:

- (i), Provision for mounting the pressure transducer in the piston as an alternative to the position in the cylinder head.
- (ii), Installation of a quartz window in the cylinder head.
- (iii), Enlargement of the compression cylinder bore.
- (iv), Installation of a temperature probe into the compression cylinder.
- (v), The interchange of different types of cylinder heads.

3.2. General outline of the compression machine.

The machine shown in fig. 3.1 is of single stroke type.

The drive gas container (6) holds nitrogen at an initial pressure ranging between 150 psi and 500 psi depending on the working condition required. The outlet of the container is closed by a poppet valve (8) which is held in the closed position by a brass shear disc (2') clamped on the valve stem.

The shear disc is supported upwards by a hardened steel die ring, and a corresponding steel punch is mounted directly

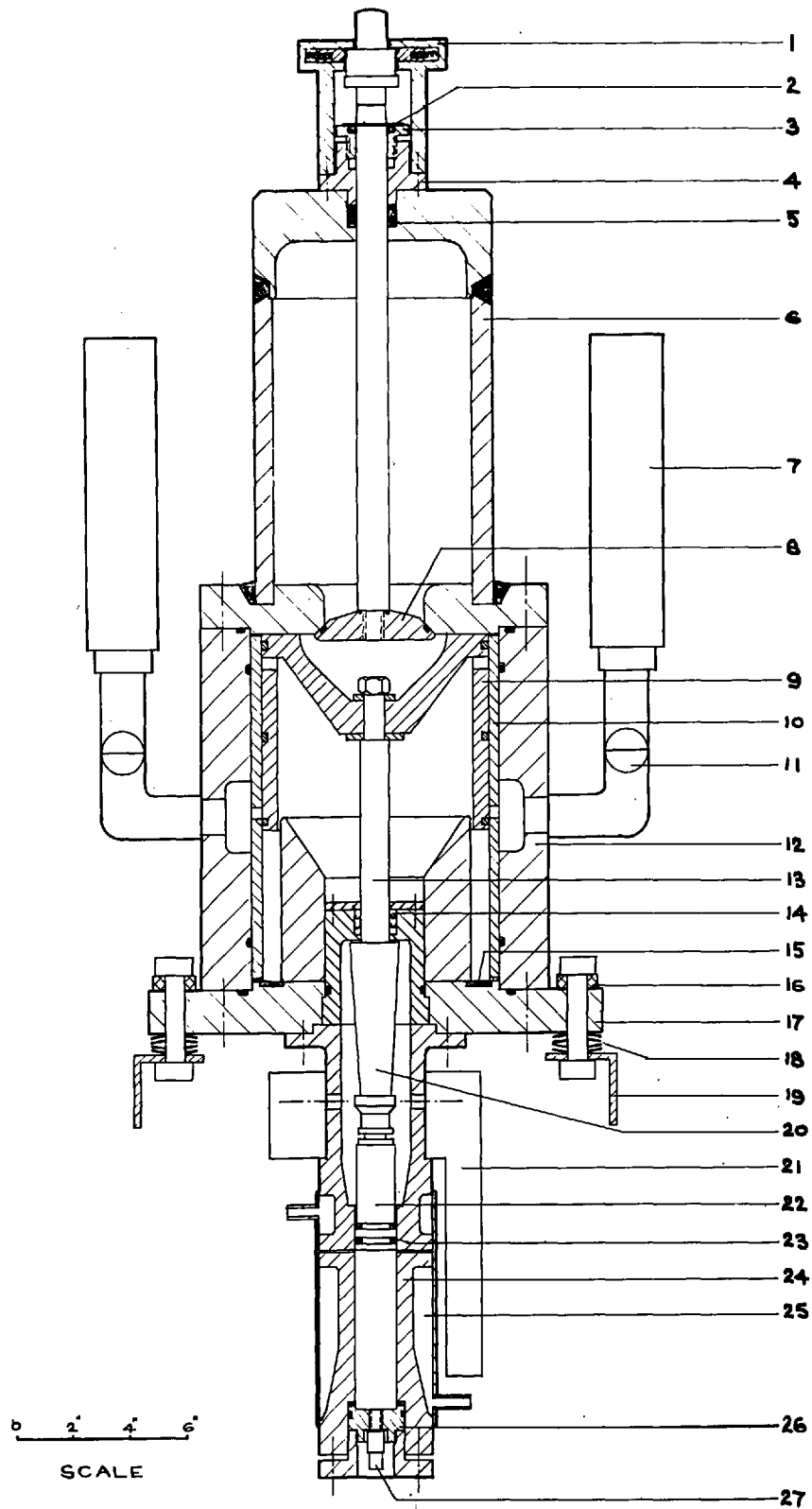


FIG. 3.1. COMPRESSION MACHINE.

Fig.3.1. - Compression machine.

1. Non-return catch.
2. Shear disc with hardened steel punch above and die below.
3. Valve stem tightening nut.
4. Top gland retainer.
5. Top gland - Chevron packing.
6. Drive gas container.
7. Silencer.
8. Valve.
9. Drive piston with three piston rings.
10. Cylinder liner.
11. Fullway on/off valve.
12. Cylinder casing.
13. Connecting Rod.
14. Midgland - Chevron packing.
15. Lead groove.
16. Rubber (TICO pad).
17. Midflange.
18. Disc springs.
19. Supporting frame.
20. Optical shutter.
21. Optical unit with light source and photomultiplier.
22. Compression piston.
23. Two PTFE piston seals.
24. Compression cylinder.
25. Water jacket.
26. Cylinder head.
27. Piezoelectric pressure transducer.

above the disc. The valve is opened by allowing a 7 lb. weight to fall and hit the top of the valve stem, when the impact load causes the disc to shear. A nonreturn catch (1) locks the valve in the open position and prevents it from being closed again by the internal gas pressure. On release of the gas, the aluminium alloy drive piston (9) is forced downwards. Consequently, the compression piston (22) driven via the connecting rod (13) is forced into the compression cylinder (24).

The braking system is operated as follows:

Before commencing a run, nitrogen is filled into the space below the drive piston (9) to an absolute pressure which is $1/6.8$ of the initial drive gas pressure. When the drive piston is forced downwards, the damping gas below it is compressed. The upward force caused by the compression increases to a maximum value when the piston is within 0.1 in. from the end of the stroke, and the piston velocity is retarded until it is almost zero at the end of the stroke. When in that position, radial ports in the drive piston coincide with ports in the cylinder liner (10) and allow the compressed damping gas to escape from the machine through eight silencers (7). The downward force of the expanded drive gas serves to hold the piston in position at the end of the stroke. To return the piston to the upper position, the drive gas is vented to atmosphere, valves (11) are closed and the pistons are then raised by admitting gas pressure to the enclosure beneath the drive piston (9).

The main dimensions of the machine are given in Table I :-

Table I - Principal Dimensions of the Compression Machine.

Drive Piston diameter	8 in.
Compression piston diameter	1.5 in.
Piston stroke length	5.25 in.
Compression ratio	adjustable up to 30:1
Drive gas volume	$450 \text{ in}^3 / 714 \text{ in}^3$.
Damping gas volume	$273 \text{ in}^3 / 13 \text{ in}^3$.
Weight of moving part	19.83 lb.
Overall height of machine	50.5 in.

Further details of the machine are given in section 3.5 and appendix E.

3.3. Theoretical analysis of the machine characteristics.

Nomenclature (see Fig.3.2).

x = piston displacement from its upper position .

x_e = " " at point where exhaust ports begin to open.

L = Length of piston stroke.

p_K = driving gas pressure.

p_D = damping " "

p_C = gas pressure inside compression cylinder.

A_K = driving piston area

A_D = damping " " (under side of drive piston).

A_C = compression piston area.

$P_K = A_K p_K$ = driving force.

$P_D = A_D p_D$ = damping force.

$P_C = A_C p_C$ = resisting force by the compressed gas.

m = mass of moving parts.

v = velocity of pistons.

γ = ratio of specific heats of nitrogen.

l_{K1} = length of initial driving gas vol. of cross-section area A_K .

l_{D1} = " " " damping " " " " " A_D .

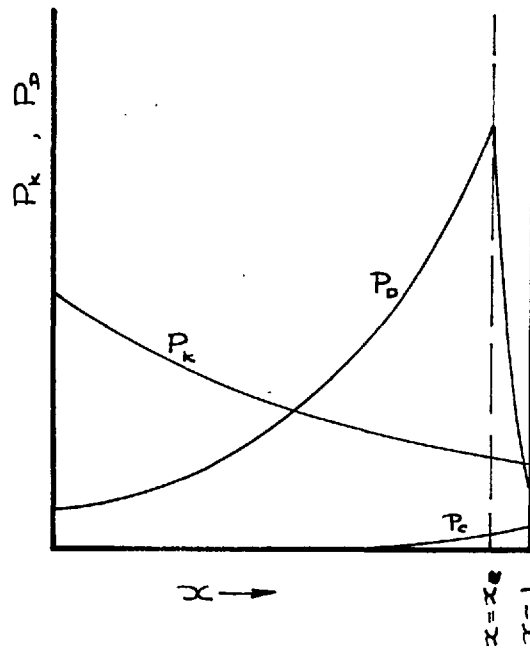
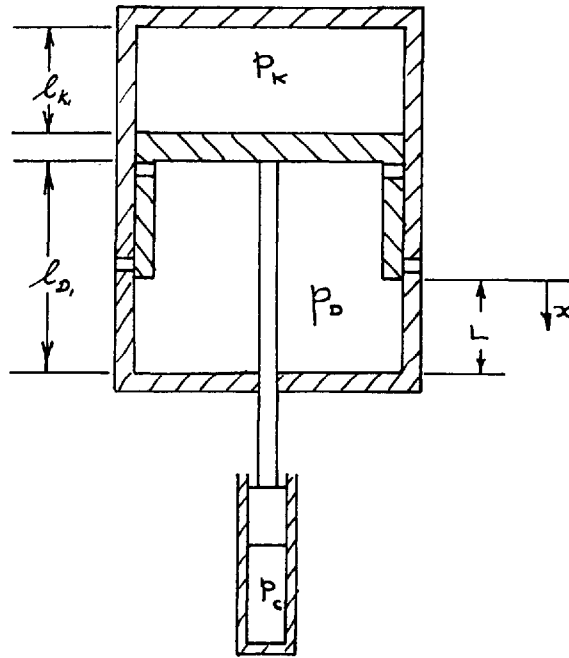
K = Area coefficient for damping gas discharge curve.

t = time.

If the piston of mass m is subjected to a driving force P_K , the resulting velocity v at a distance x from the starting position may be calculated from the energy balance

$$\frac{mv^2}{2} = \int (P_K - P_D - P_F - P_C) dx \quad \dots (1)$$

where P_D , P_F and P_C are the instantaneous forces arising from the damping gas, friction and the compressed gas in the compression cylinder respectively. If the compression and expansion of the gases are assumed to be adiabatic and, if the initial gas pressures are known, then a force-displacement diagram similar to that shown in Fig.3.2. may be constructed for the machine. It may be seen from Fig.3.2. that the force arising from the driving gas, P_K decreases from $x = 0$ to $x = L$ as the gas expands, whilst that arising from the damping gas P_D increases from $x = 0$ to $x = x_e$ as the damping gas is compressed. At $x = x_e$, the exhaust ports begin to open, and at $x = L$ they are completely open. The venting of the damping gas



FORCE DISPLACEMENT DIAGRAM.

Fig. 3.2.

from the exhaust ports causes P_D to decrease rapidly from its peak value, approximately at $x = x_e$, to a low value at $x = L$.

In practice, the force arising from friction effects P_F , and that due to the compressed fuel-air mixture P_C are small compared with P_K and P_D . The effect of these terms as expressed in equation (1) may therefore be neglected.

The shape of the P_D curve between $x = x_e$ and $x = L$ is a function of the initial damping gas pressure p_D , the port opening area, and the discharge condition of the ports. However, for the purpose of the analysis it is assumed that this part of the curve can be taken as a straight line. (This assumption is discussed in detail in section 3.3.3.).

3.3.1. Initial pressure setting.

If the piston is to reach the end of its stroke with minimum shock, then its velocity should be zero at the end of the stroke. Under these conditions, equation (1) becomes

$$0 = \int_{x=0}^{x=L} (P_K - P_D - P_F - P_C) dx$$

if P_F and P_C are neglected, in comparison with P_K and P_D , then

$$\begin{aligned} & \int_{x=0}^{x=L} (P_K - P_D) dx \stackrel{?}{=} 0 \\ \text{i.e.} \quad & \int_{x=0}^{x=L} P_K dx \stackrel{?}{=} \int_{x=0}^{x=L} P_D dx \quad \dots (2) \end{aligned}$$

Now at displacement x

$$P_K = p_{K1} A_K \left(\frac{l_{K1}}{l_{K1} + x} \right)^{\gamma}$$

where suffix 1 denotes condition at $x = 0$.

While between $x = 0$ and $x = x_e$

$$P_D = P_{D1} A_D \left(\frac{l_{D1}}{l_{D1} - x} \right)^\gamma$$

and between $x = x_e$ and $x = L$

$$\int_{x=x_e}^{x=L} P_D dx = K p_{D1} A_D \left(\frac{l_{D1}}{l_{D1} - x_e} \right)^\gamma (L - x_e)$$

where K is the area coefficient for the damping gas discharge curve which depends on the shape of the P_D curve between $x = x_e$ and $x = L$.

If it is assumed that between these limits P_D falls linearly to a zero value, then $K = 0.5$. Equation (2) may be written as

$$\int_{x=0}^{x=L} p_{K1} A_K \left(\frac{l_{K1}}{l_{K1} + x} \right)^\gamma dx = \int_{x=0}^{x=x_e} p_{D1} A_D \left(\frac{l_{D1}}{l_{D1} - x} \right)^\gamma dx + K p_{D1} A_D \left(\frac{l_{D1}}{l_{D1} - x_e} \right)^\gamma (L - x_e)$$

By integrating and rearranging, the following equation is obtained :

$$\frac{p_{K1}}{p_{D1}} = \frac{A_D l_{D1}^\gamma \left[\frac{l_{D1}^{1-\gamma}}{l_{D1} - (l_{D1} - x_e)} + K \frac{(1-\gamma)(L - x_e)}{(l_{D1} - x_e)} \right]}{A_K l_{K1}^\gamma \left[(l_{K1} + L)^{1-\gamma} - l_{K1}^{1-\gamma} \right]} \quad \dots (3)$$

where $\frac{p_{K1}}{p_{D1}}$ is the necessary ratio of the initial driving gas pressure to damping gas pressure for a smooth piston landing.

If the $\frac{p_K}{p_D}$ ratio is improperly set, the piston will either rebound before reaching the end of the stroke, as in the case where the ratio $\frac{p_K}{p_D}$ is too small, or rebound as a result of an impact in which case the

ratio $\frac{P_K}{P_D}$ is too large. These conditions are illustrated in Fig.3.3.

The range of values for the ratio $\frac{P_K}{P_D}$ needed to cause a smooth landing is limited by the condition that $\frac{P_D}{P_K}$ at $x = L$, $P_K > P_D > 0$.

3.3.2. Compression time and piston acceleration.

On differentiating equation (1) :

$$mV \frac{dv}{dx} = P_K - P_D - P_F - P_C$$

Thus the acceleration is given by

$$\frac{d^2x}{dt^2} = v \frac{dv}{dx} = \frac{1}{m} (P_K - P_D - P_F - P_C) \quad \dots (4).$$

where t is time.

In order to obtain t as a function of x from equation (4), P_K and P_D have to be expressed in terms of A , p , l and x . If this is done, two differential equations can be derived, one covering the range between $x = 0$ and $x = x_e$, and the other the range $x = x_e$ to $x = L$. In order to avoid the difficulty of solving the equations which are not of standard form, values for t were derived graphically according to the following procedure:

- a) P_K and P_D curves were plotted against x for initial values of P_{K1} and P_{D1} , which satisfied equation (3).
- b) The integral $\int_{x=0}^{x=x} (P_K - P_D) dx$ was graphically integrated at convenient intervals of x . If P_F and P_C are neglected because of their small magnitude, then equation (1) may be rearranged as

$$\frac{1}{V} = \sqrt{\frac{m}{2 \int (P_K - P_D) dx}}$$

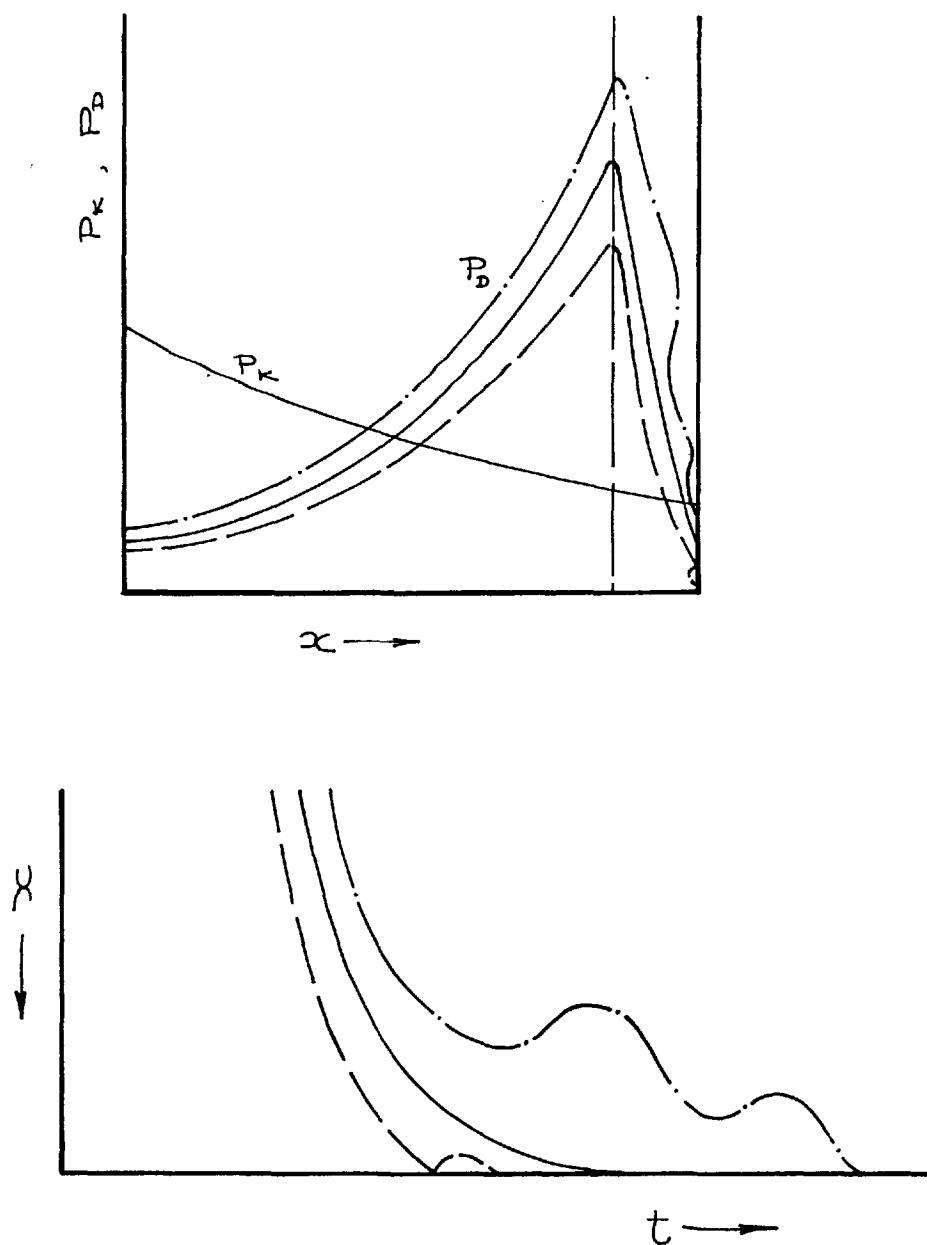


Fig. 3.3.

Effect of incorrect damping gas pressure.

— · — · — · —	$\frac{P_{K_1}}{P_{D_1}}$	ratio low.
—————	" "	correct.
— · — · — · —	" "	high.

Values of $\frac{1}{V}$ are calculated at convenient intervals of x and plotted against x .

c) The integral $\int_0^x \frac{1}{V} dx$ was evaluated graphically, the values being the time required to reach displacement x .

$$\text{i.e. } \int_0^x \frac{1}{V} dx = t_x.$$

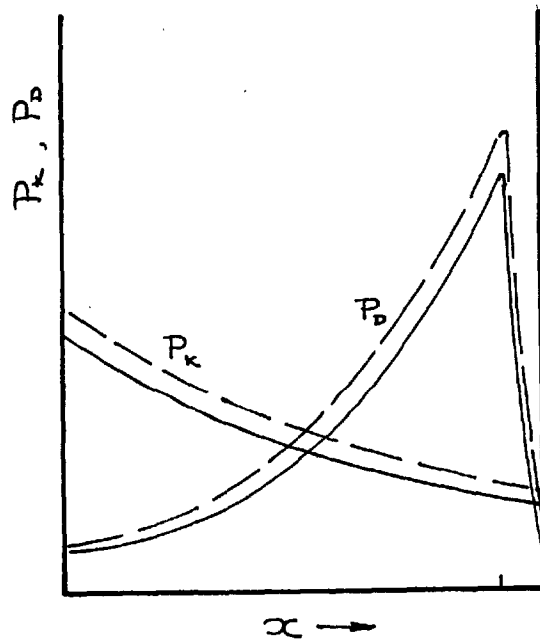
To shorten the compression time of the machine, it is necessary to increase the piston speed. This, as is shown by equation (1), may be achieved by increasing $P_K - P_D$. There are three possible methods by which this can be done:-

- a) By increasing the initial pressures p_{K1} and p_{D1} .
- b) By decreasing the initial damping gas volume.
- c) By decreasing the initial driving gas volume.

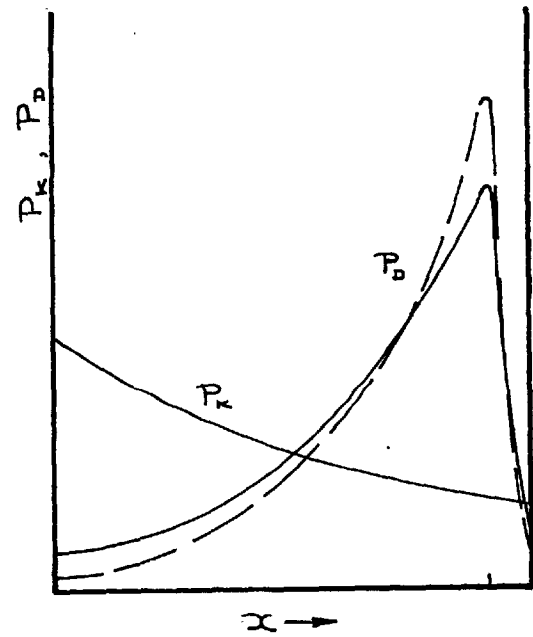
A careful study of the three methods revealed that method a) was the most convenient to carry out in practice. Method b) is to be preferred to method c), firstly, for practical reasons and, secondly, because for the same damping gas force P_D at $x = L$, the magnitude of P_K is decreased as is shown in sketch c) in Fig.3.4. This means that the range of the ratio $\frac{P_K}{P_D}$ necessary to achieve a smooth landing is restricted.

3.3.3. Discharge of damping gas.

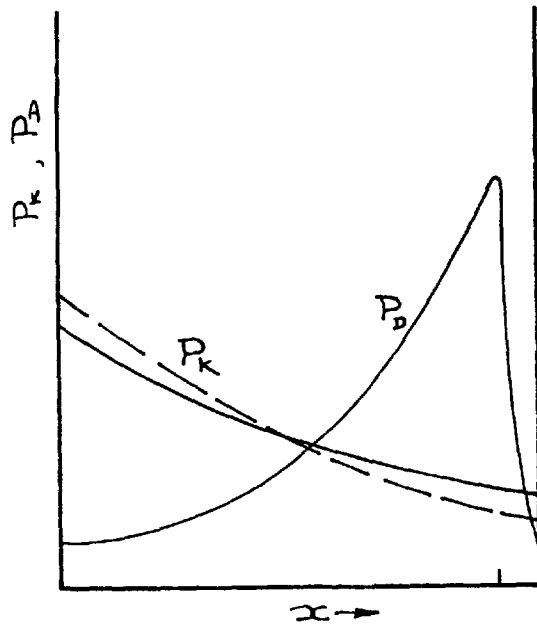
The arrangement of the exhaust port is such that the port height opened at x is $x - x_e$. If the total port height is less than $L - x_e$, then the maximum port height is obtained somewhere before $x=L$.



a). By increasing P_{k1} and P_{d1} .



b) By decreasing initial damping gas volume.



c). By decreasing initial drive gas volume.

Fig. 3.4.

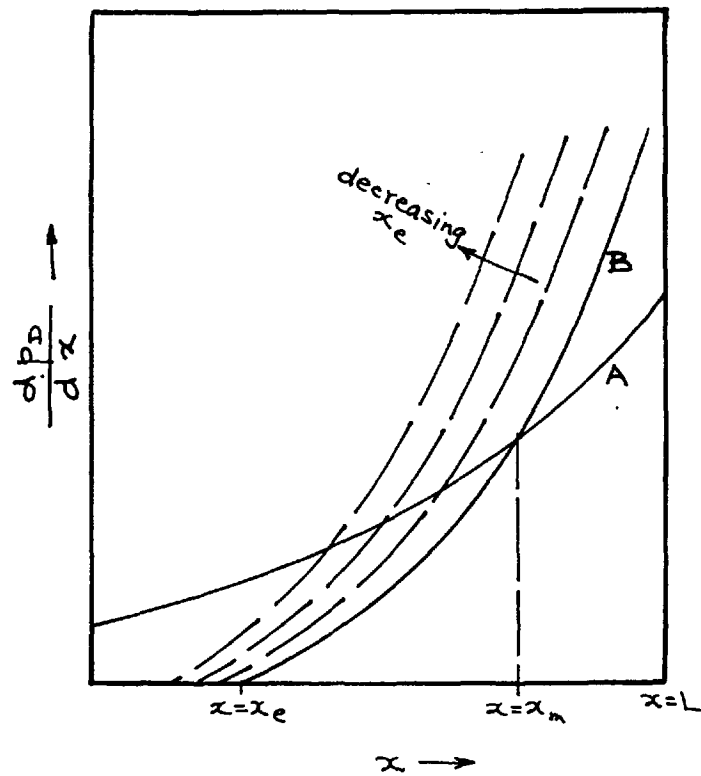
Three different methods for decreasing compression time, as indicated by the dotted lines.

Let the rate of pressure change in relation to piston displacement arising from compression be denoted by $\left(\frac{dp_D}{dx}\right)_{\text{comp.}}$ and that resulting from the escape of gas through the ports to be denoted as $\left(\frac{dp_D}{dx}\right)_{\text{exh.}}$. If at some displacement $x = x_m$, such that $x_e < x_m < L$,

$$\left(\frac{dp_D}{dx}\right)_{\text{comp.}} + \left(\frac{dp_D}{dx}\right)_{\text{exh.}} = 0$$

i.e.
$$\frac{dp_D}{dx} = 0$$

then there will be a peak value of p_D at x_m . As is shown in Fig.3.5., such a condition is quite feasible; at $x = x_e$, $\left(\frac{dp_D}{dx}\right)_{\text{comp.}}$ has a finite and positive value whilst $\left(\frac{dp_D}{dx}\right)_{\text{exh.}}$ is zero because there is no discharge at that instant. If the port sizes are correct, a pressure drop is inevitable between x_e and L , i.e. $-\left(\frac{dp_D}{dx}\right)_{\text{exh.}}$ gradually increases until it is greater than $\left(\frac{dp_D}{dx}\right)_{\text{comp.}}$. Therefore, at some position $x = x_m$ between $x=x_e$ and $x=L$, conditions will be such that $-\left(\frac{dp_D}{dx}\right)_{\text{exh.}} = \left(\frac{dp_D}{dx}\right)_{\text{comp.}}$. Thus, since there is a peak value of p_D between $x=x_e$ and $x=L$, the value of factor K will be greater than 0.5 and may approach 1.0 depending on the chosen values of x_e and the maximum exhaust port size.



$A = \left(\frac{dP}{dx}\right)_{\text{comp}} = \text{pressure change due to compression.}$

$B = \left(\frac{dP}{dx}\right)_{\text{exh.}} = \text{pressure change due to venting.}$

Fig. 3.5. DAMPING GAS PRESSURE CHANGES.

Calculations for exhaust port size and discharge time are dealt with in section 3.4.

Effect of exhaust port size and timing.

To ensure a smooth landing, equation (2) must be satisfied by adjusting the ratio $\frac{P_K}{P_D}$ correctly; and in addition the condition $P_K > P_D > 0$ must be satisfied at the end of the stroke. Cases may arise in which although the first condition is satisfied, the second is not, because the size of the exhaust ports is incorrect. The effect in such cases may be understood by reference to fig. 3.6. The case in which the ports are too small is represented by the broken line. Because the exhaust passages are too small, the damping gas pressure when the piston is near the end of its stroke is such that the upward force on the driving piston P_D is greater than the force P_K exerted by the expanded drive gas. The result of this is that the piston is bounced up again. Meanwhile, more of the damping gas is exhausted so P_D is now less than P_K and the piston remains stationary at the end of its stroke after its second landing.

The case in which the ports are too big is represented by the dash-dot line. Because of the excessive port size, pressure P_D soon drops to below or near atmospheric pressure before the piston reaches the end of the stroke as indicated by point A. Retardation now ceases, and the piston is accelerated again towards the end of the stroke by the force P_K . The consequences of this are that the piston lands with an impact to cause it to rebound, despite the fact that $P_K > P_D$. This state of affairs is to be avoided because of the severe shock and

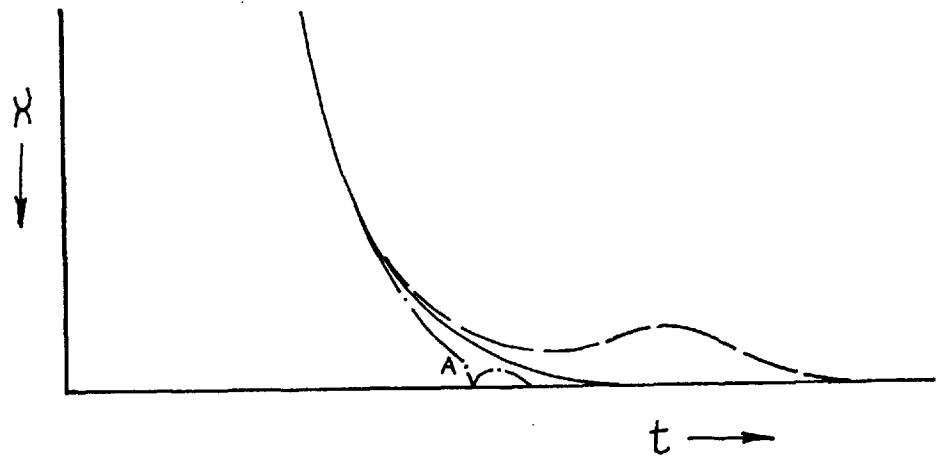
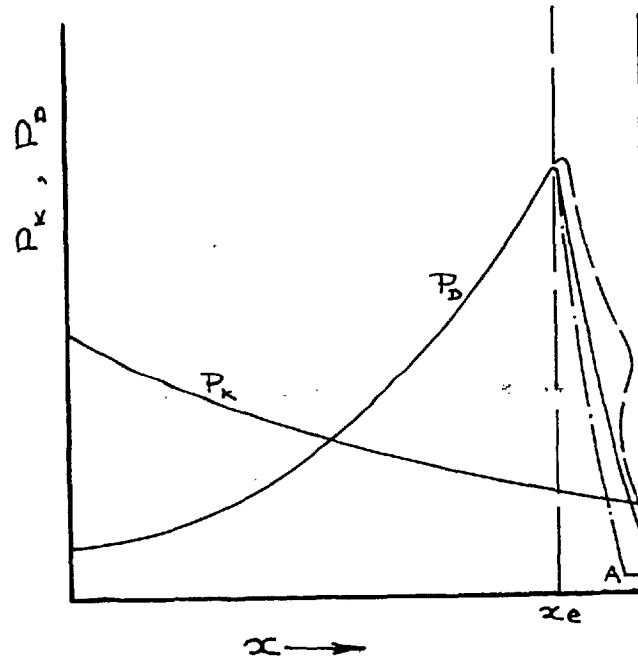


Fig. 3.6.

Effects of different exhaust port sizes.

———— too small.

———— correct.

..... too large.

vibration to the machine.

The effect of changing the time of opening of the ports.

By changing the axial position of the cylinder sleeve the value of x_e may be adjusted. The effects of such a change on the force displacement diagram are illustrated by fig. 3.7 from which it may be seen that they are similar to those caused by incorrect port sizes. To some extent, it is possible to compensate for incorrect port sizes by altering the port opening time. For example, it is necessary to open undersized ports earlier and to delay the opening of oversized ports, if a smooth landing is to be achieved.

3.4. Prediction of Performance.

The essential dimension of the machine arrived at as a result of preliminary design calculations are as follows:

Piston stroke	$L = 5.25$ in.
Exhaust ports open at	$x_e = 4.85$ in.
Length of initial drive gas volume	$l_{K1} = 9.51$ in.*
" " " damping " "	$l_{D1} = 5.52$ in.
Drive piston area	$A_K = 50.30$ in ² .
Damping " "	$A_D = 49.50$ in ² .
Compression piston area	$A_C = 1.765$ in ² .
Mass of moving part	$m = 19.83$ lb. _f .

* It should be noted that the length of the initial drive gas volume l_{K1} includes the volume in the concave top face of the drive piston.

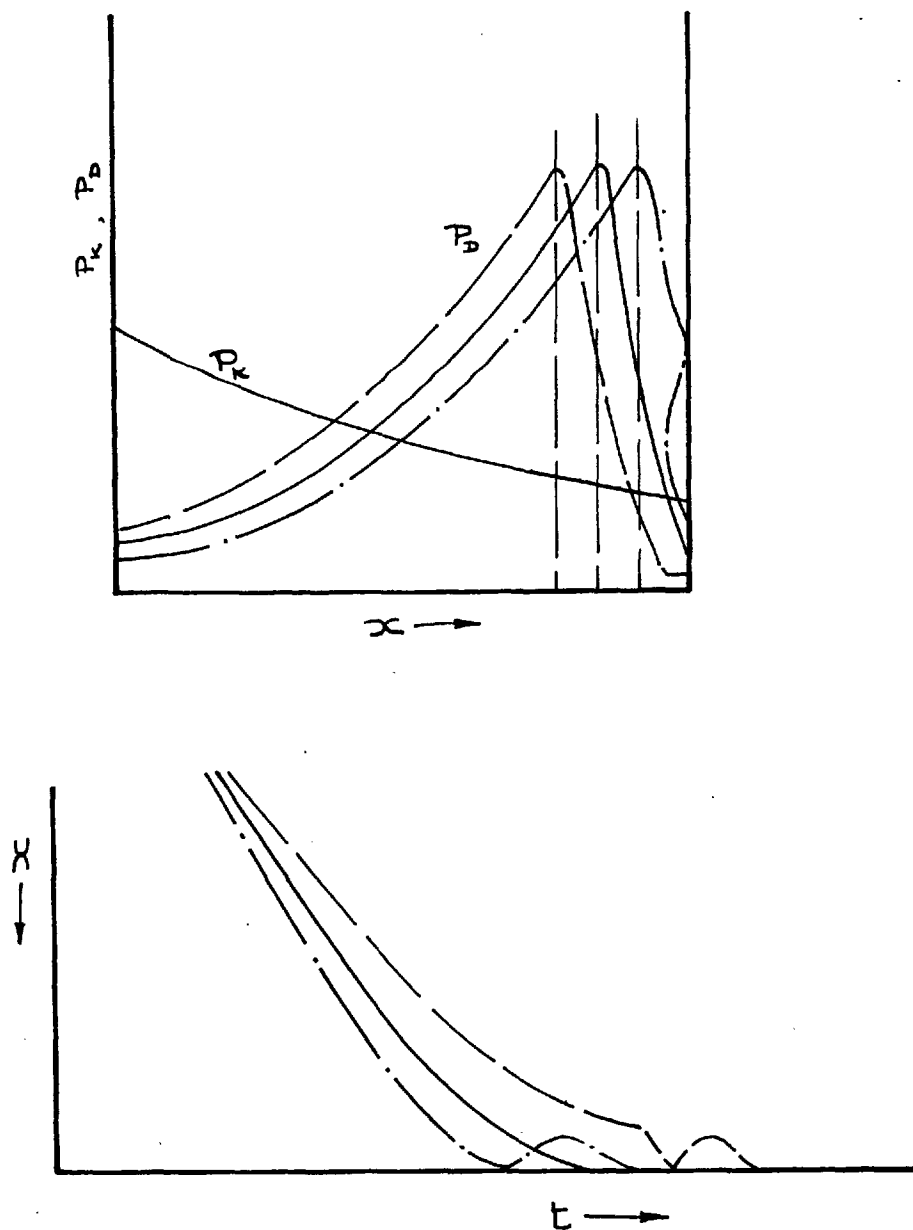


Fig. 3.7.

Effects of incorrect port opening times.

———— early,
 ———— correct,
 — · — · — late.

Because this space is isolated by the release valve, it is not filled to pressure p_{K1} . If it is assumed that the pressure in the space is p_{D1} or atmospheric, the calculated value $\frac{p_{K1}}{p_{D1}}$ should be multiplied by 1.088 to allow for this. (see Appendix D).

To predict the compression time when using a drive gas pressure $p_{K1} = 180$ psia, the correct $\frac{p_{K1}}{p_{D1}}$ ratio is first determined and the calculation procedure described in section 3.3.2. is then followed.

Initial drive pressure to damping pressure ratio.

For nitrogen gas, it was assumed that $\gamma = 1.4$ and from section 3.3.3. that $K = 1.0$. Substituting these and other known values into equation (3), it was found that

$$\frac{p_{K1}}{p_{D1}} = \frac{49.5 \times 5.52^{1.4} \left[5.52^{-0.4} - (552-4.85)^{-0.4} \div 1.0^{\frac{(1.4)(5.25-4.85)}{(5.52-4.85)}} \right]}{50.3 \times 9.5^{1.4} \left[(9.51+5.25)^{1-1.4} - 9.51^{(1-1.4)} \right]}$$

$$= 6.41$$

When a correction is made for the isolated concave volume above the drive piston face, then $\frac{p_{K1}}{p_{D1}} = 6.41 \times 1.088 = \underline{6.95}$

Compression time.

If it is assumed that the driving gas pressure is set to 180 psia, then as soon as the release valve is opened,

$$p_{K1} = \frac{180}{1.088} = 166 \text{ psia}$$

and

$$p_{D1} = \frac{180}{6.95} = 25.9 \text{ psia.}$$

With these values, P_K and P_D are plotted against x by assuming

$$P_K = 50.3 \times 166 \left(\frac{9.51}{9.51+x} \right)^{1.4} \quad \text{from } x=0 \text{ to } x=5.25 \text{ in. and}$$

$$P_D = 49.5 \times 25.9 \left(\frac{5.52}{5.52-x} \right)^{1.4} \quad \text{from } x=0 \text{ to } x=x_e=4.85 \text{ in.}$$

A simplified graph of P_D between $x=x_e = 4.85$ in. and $x = 5.25$ in. sufficed for the purpose of these calculations with the aid of the following three assumptions:

a) The factor K for the P_D graph is 1.0

b) A peak value of P_D is attained after $x=x_e = 4.85$ in.

and before the end of the stroke.

c) The ports are of a size such that $P_D < P_K$ at the end of the stroke.

This graph (see Fig.3.8) consisted of two straight lines between $x = 4.85$ in. and $x = 5.25$ in. having a peak value of P_D at $x = 5.12$ in. The values of the piston velocity V and compression time t were obtained from the force displacement curves using the procedure described in section 3.3.2. The results are plotted as a function of displacement x in Fig.3.8. A knowledge of maximum acceleration and deceleration forces may be obtained from the data in Fig.3.8.

For example, for the conditions:

$$P_{K1} = 180 \text{ psia.}$$

$$P_{D1} = 2.59 \text{ psia}$$

$$x_e = 4.85 \text{ in. when exhaust ports open.}$$

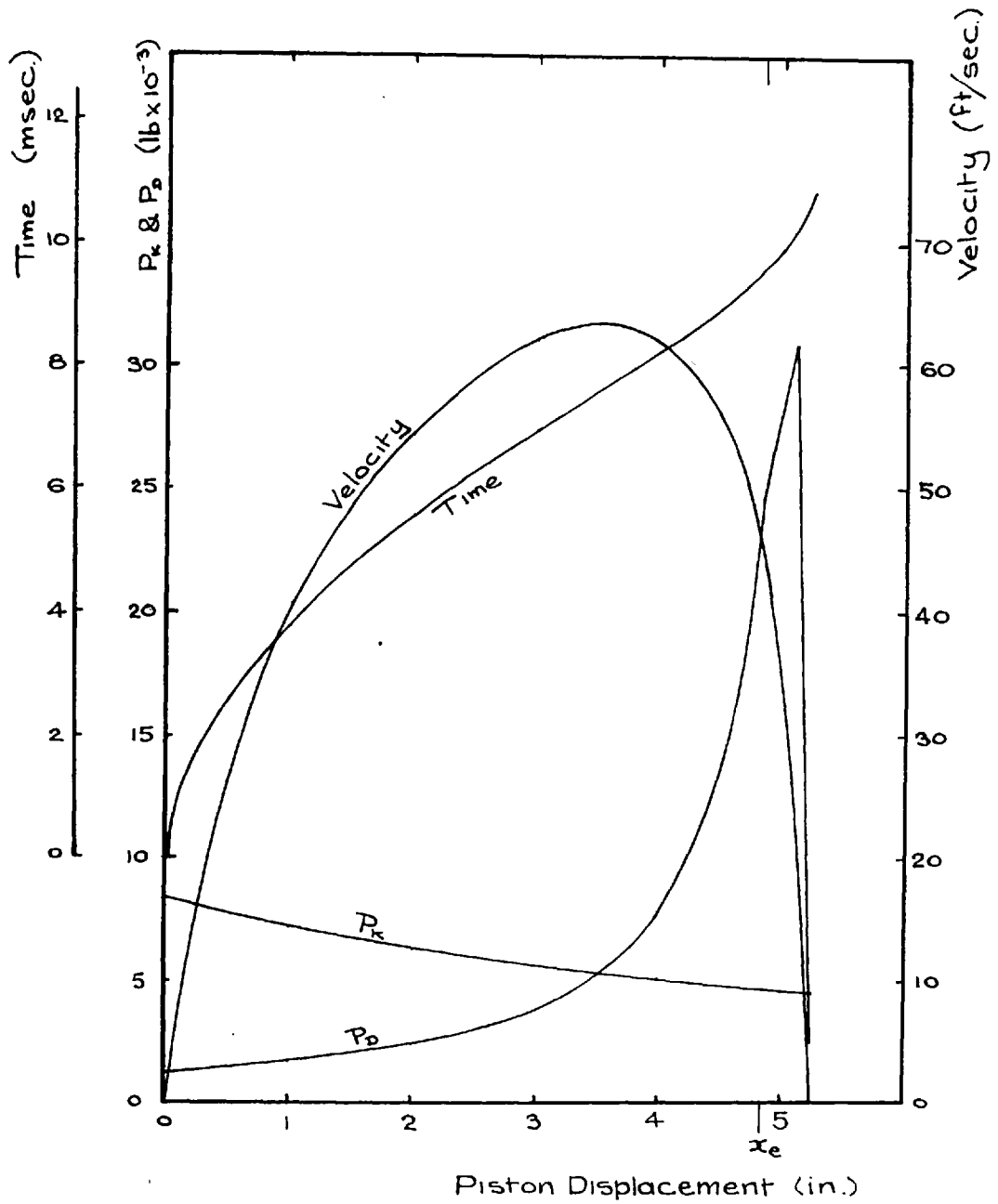


FIG. 3.8. CALCULATED PISTON PERFORMANCE
FOR INITIAL DRIVING GAS PRESSURE = 180 psia

The calculations show that,

Compression time $t = 10.8$ msec.

Max.^m piston velocity = 63.9 ft/sec.

" " acceleration = 352 g.

" " deceleration = 1360 g.

Similar calculations for different values of p_{K1} , indicate that compression time varies from 14 msec. to 6.5 msec. as the initial driving pressure p_{K1} is varied from 90 psia to 500 psia. The results of such calculations are given in Fig.3.9.

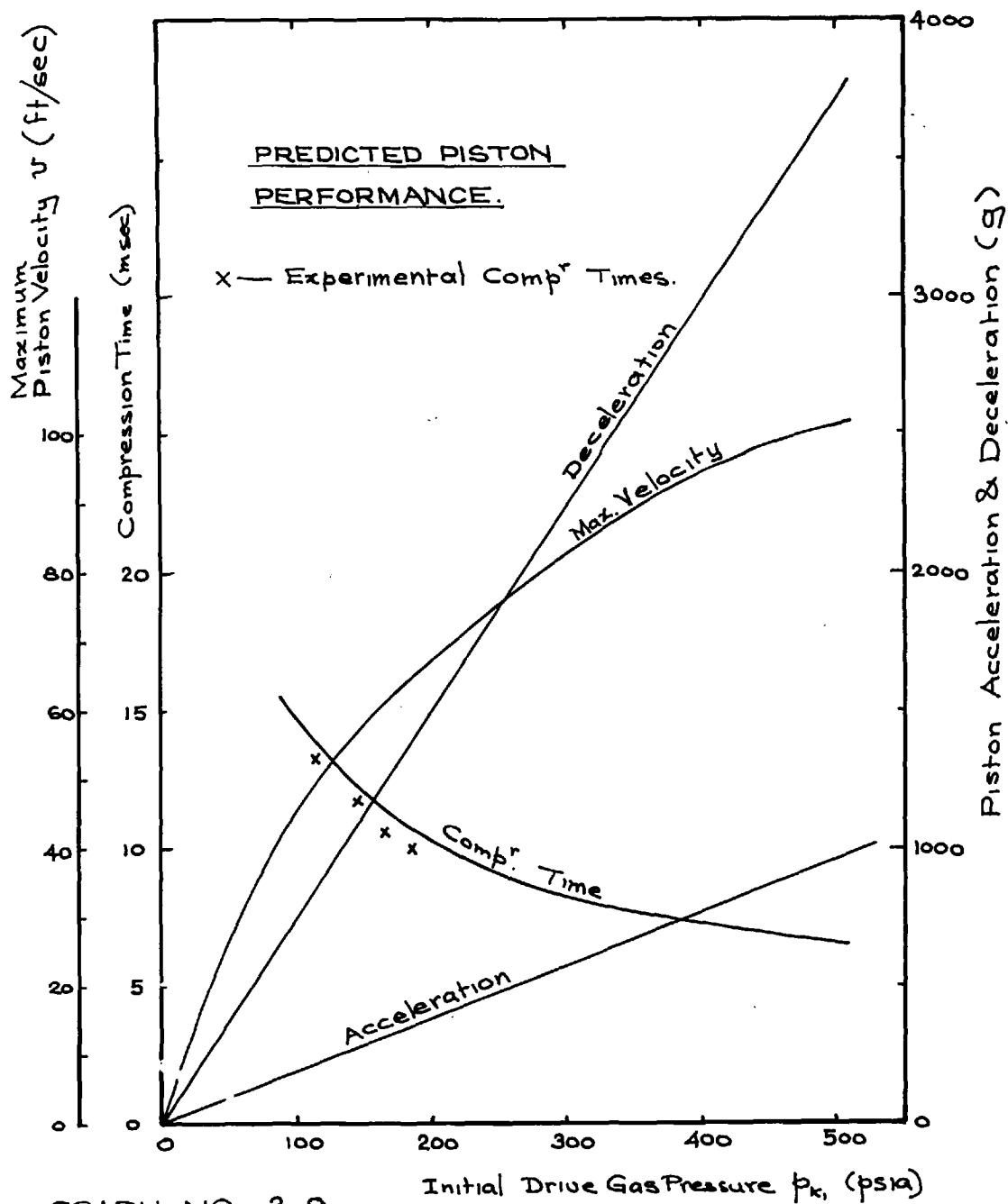
Damping gas discharge time.

The previous calculations of the compression time are based on the assumption that $K = 1$ and that the final damping gas pressure p_D is approximately 70 psia. These calculations are only correct if the exhaust port sizes are sufficiently large to achieve this. An attempt was made to check these assumptions by plotting a port height-time diagram for $x_e = 4.85$ in. and calculate whether or not the discharge could be completed within the predicted period.

Schweitzer⁽³⁶⁾ has shown that

$$\int_{t_1}^{t_2} A dt = \frac{1}{\psi} \cdot \frac{1}{\sqrt{2g}} \frac{V_1}{\sqrt{RT_1}} \cdot \frac{2}{(\gamma-1)} \left[\left(\frac{p_2}{p_1} \right)^{\frac{1-\gamma}{2\gamma}} - 1 \right] \dots (5)$$

for gas discharging from a vessel of volume V_1 through an orifice of area A , at an initial temperature T_1 , pressure p_1 and time t_1 . Where p_2 is the pressure in the vessel at time t_2 ; γ and R are the ratio of specific heats of the gas and the gas constant respectively, and ψ is the Nusselt coefficient which includes the contraction coefficient



GRAPH NO. 3.9.

of the gas jet leaving the orifice. ψ is dependent on the ratio of gas pressure inside the vessel to the outside pressure into which discharge takes place. However, for most pressure ratios above the critical ratio for sonic discharge through sharp edged orifice, the value of ψ is approximately 0.4.

Fig.3.10. is a port height - time diagram plotted from the piston displacement - time relation given in Fig.3.8 and for $x_e = 4.85$ in. The integrated port height time value, obtained from Fig.3.10. is 0.36 in. msec., and since the total port width is 10.27 in. then the left hand side of Equation (5) is given by

$$\begin{aligned} \text{L.H.S.} &= \int_{t_1=0}^{t_2=1.3 \text{ msec.}} \text{Adt} = 0.36 \times 10.27 \times \frac{1}{12^2} \times \frac{1}{1000} \\ &= 2.56 \times 10^{-5} \text{ ft}^2 \cdot \text{sec.} \end{aligned}$$

In order to solve the right hand side of the equation, the condition that $P_D = 25.0 \times 10^3$ lb. when $x = x_e = 4.85$ in. is taken from Fig.3.8. Thus the value of p_1 corresponding to P_D is

$$\begin{aligned} p_1 &= p_{D_{x_e}} = 520 \text{ psia} \\ \text{and } V_1 &= 25.8 \times \frac{1}{12^3} \text{ ft}^3. \\ p_2 &= 70 \text{ psia} \\ T_2 &= 950^\circ \text{R for initial temp. } 520^\circ \text{R} \\ R &= 55 \text{ ft.lb/lb}^\circ \text{R for nitrogen.} \end{aligned}$$

Substituting these in the right hand side of Equation (5)

$$\begin{aligned} \text{R.H.S.} &= 3.38 \times 10^{-5} \text{ ft}^2 \cdot \text{sec.} \text{ is obtained as compared} \\ &\text{with } 2.56 \times 10^{-5} \text{ ft}^2 \cdot \text{sec. for the L.H. side.} \end{aligned}$$

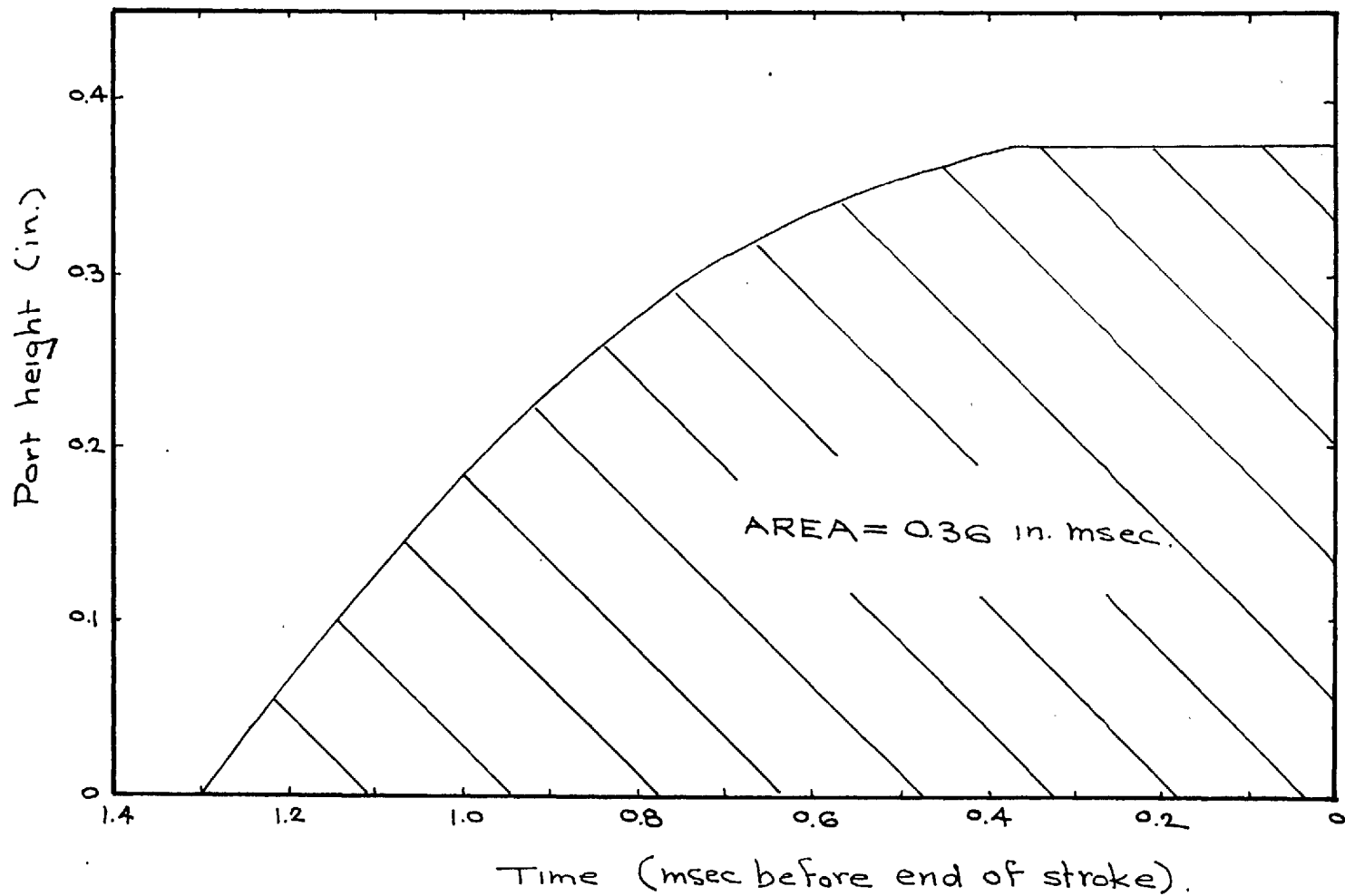


Fig. 3.10. PORT HEIGHT-TIME DIAGRAM (Plotted from Fig. 3.8.).

The discrepancy between the two values indicates that the predicted port area - time ~~area~~ value may be smaller than the actual value.

If it is assumed that $\int_{t_1}^{t_2} A dt = 3.38 \times 10^{-5} \text{ft}^2 \text{sec.}$, then it may be

shown that the time for the piston to travel from $x = x_e$ to $x = L$ is 1.72 msec., instead of previously calculated value of 1.3 msec.

The uncertainties in the predicted time taken for the piston to travel from $x = x_e$ to $x = L$ is small compared with the total compression time predicted to be 10.8 ± 1.5 msec.

Accuracy of the calculation:

The accuracy of the predicted compression times and initial pressure ratios depend largely on the discharge condition of the exhaust ports. An obstructed exhaust passage will decrease the discharge rate and consequently increase the compression time. Another factor which may affect the accuracy of the calculation is the error involved in determining the value of l_{D1} from the dimensions of the machine. Because of the intricate shape of the damping gas space, an error of $\pm 2\%$ in the determination of l_{D1} is possible. This will cause a corresponding error of $\pm 5\%$ in the calculated values of $\frac{P_{K1}}{P_{D1}}$. Although the accuracy in predicting the factor K may be as large as $\pm 10\%$, this only causes an error of approximately 3% in the values of $\frac{P_{K1}}{P_{D1}}$.

It is expected that the compression time achieved in practice will be within 15% of the predicted values.

3.5. Machine details.

The mechanical details and stress calculations of the important machine parts are described in Appendix E.

Accurate alignment of the machine assembly was an important point born in mind during the design stage. Therefore small clearances, large contact areas between parts and the use of 'O' rings instead of gaskets for pressure sealing were adopted.

In order that rotation of the drive piston did not seriously affect the venting of damping gas, a combination of 16 ports in the cylinder and 14 ports in the drive piston enabled the maximum variation of total exhaust port area due to piston rotation to be reduced to 0.68%. (See 'Radial Arrangement of Exhaust ports' in Appendix E.).

The sliding surface of the compression piston was coated with a layer of P.T.F.E. so that no lubricating oil is necessary in the compression cylinder.

CHAPTER 4.4. General layout of the apparatus.

The general outline of the compression machine has already been described in Section 3.2. Fig.4.1 illustrates in diagramatic form the arrangement of the complete apparatus. The drive gas and the damping gas for the machine is supplied from compressed nitrogen gas cylinders '10' and '11' respectively. The initial damping gas pressure is maintained constant in spite of the small leakage of gas past the piston rings of the drive piston '3', by a pressure regulator '13' and a non-return valve '14'.

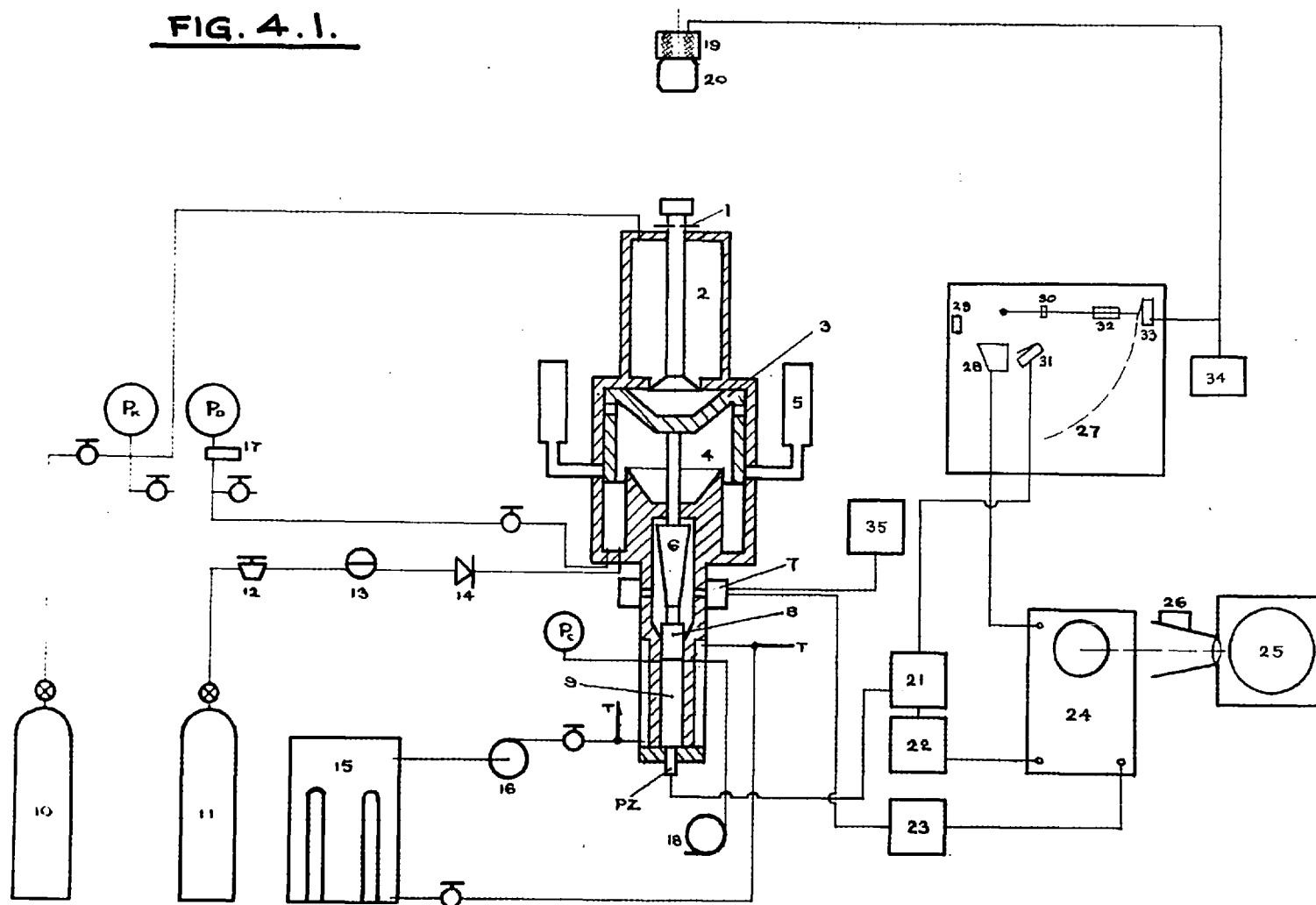
The machine motion is initiated by a falling weight '20' which hits the release valve stem and shears the disc '1'. The falling weight is normally held in its initial position by an electromagnet '19', which is fixed to the top of a removable guide (not shown in the figure).

The electrical signals from the optical unit '7' which measures piston displacement, and the pressure transducer 'PZ' which measures the pressure in the compression cylinder '9', are displayed on a double beam oscilloscope '24' and photographically recorded by a rotating drum camera '25'.

Fig. 4.1. General arrangement of apparatus. (page 56).

1. Shear disc.
2. Drive gas container.
3. Drive piston.
4. Damping gas space.
5. Silencer.

FIG. 4.1.



- | | |
|--|------------------------------------|
| 6. Optical shutter. | 21. Electrometer amplifier. |
| 7. Displacement measuring device. | 22. D.C. pre-amplifier. |
| 8. Compression piston. | 23. Transistor D.C. amplifier. |
| 9. Compression cylinder. | 24. Double beam oscilloscope. |
| 10. Nitrogen gas container. | 25. Drum camera. |
| 11. " " " | 26. Time marker unit. |
| 12. Pressure reducing valve. | 27. Pendulum time switch unit. |
| 13. Pressure regulator. | 28. Contact plate. |
| 14. Non-return valve. | 29. Non-return catch. |
| 15. Temperature controlled water tank. | 30. Contact brush. |
| 16. Centrifugal pump. | 31. Microswitch. |
| 17. Automatic shut-off device. | 32. Bob weight. |
| 18. Vacuum pump. | 33. Microswitch. |
| 19. Electromagnet. | 34. 12V. Accumulator. |
| 20. Falling weight. | 35. Extra high tension power unit. |
- PZ Pressure transducer.
- P_K Driving gas pressure gauge.
- P_D Damping gas pressure " .
- P_C Compression cylinder pressure gauge.

The temperature of the compression cylinder is regulated by a water jacket through which hot water, containing a rust inhibitor, is circulated. The hot water reservoir '15' is heated by two 500 Watt immersion heaters, and the temperature of the fluid is controlled to within $\pm 0.25^{\circ}\text{C}$. with an adjustable bimetallic thermostat.

CHAPTER 5.5. INSTRUMENTATION.5.1. Pressure Measurement:

The chief requirements for the pressure transducer are as follows:

- a. Pressure range up to 3500 psi.
- b. Operating temperature up to 200°C.
- c. A linear pressure/output characteristic.
- d. Minimum sensitivity to flame temperature.
- e. Rapid response time and capability of measuring static pressures also
- f. Minimum sensitivity to vibration.

Two types of transducers were chosen for the work, one of the piezo-electric type and one of resistance strain gauge type.

5.1.1. Piezoelectric pressure transducer.

A piezoelectric transducer type PZ14, made by the Swiss Locomotive & Machine Works (SLM) was used for measuring combustion pressures by mounting the transducer in the cylinder head.

The charge generated across the face of the piezoelectric crystal when the pressure is affected is transferred to a calibrated condenser which then supplies a grid voltage to the electrometer valve in an SLM type PVI7 amplifier. The electrometer has an extremely high input impedance, so that the charge in the transducer can be retained for a considerable period. The output impedance of the amplifier is matched for driving a standard cathode ray oscilloscope. In order to increase the strength of the signal from the electrometer amplifier 10 times before it is fed into the oscilloscope, a D.C. pre-amplifier Type MR.514, made by Southern Instrument Ltd., is used. With reference to Fig.4.1., the transducer,

electrometer amplifier, the D.C. pre-amplifier and oscilloscope are indicated by parts PZ, 21, 22, and 24 respectively. Part 31 is a microswitch which triggers a calibration unit incorporated in the electrometer amplifier signal grid and enables a signal to be displayed on the oscilloscope immediately before an experiment. (see Cp in Run No 298 p.78.)

In preliminary tests, the decay of output signal was found to be less than 2% when the transducer was subjected to a constant hydraulic pressure of 1500 psi. The test for flame sensitivity was carried out by quickly passing a bunsen burner over the pressure sensing face of the transducer. This resulted in a temporary drop equivalent to -20 psi pressure.

The pressure transducer was calibrated statically with hydraulic pressure against a dead weight pressure tester, and the calibration pressures recorded on the drum camera were found to be linear and reproducible to within $\pm 1\%$.

No dynamic calibration of the transducer has been carried out, but it is known that the type PZ 14 transducer has a linear characteristic to within $\pm 1\%$ when calibrated in shock tubes. (38)

The piezoelectric transducer is not suited for mounting inside a moving piston, because the crystal is sensitive to acceleration and also the conducting lead is liable to capacitance as well as inductive interference when it is in motion. Therefore it was planned to use a

resistance strain gauge type of transducer for this purpose.

5.1.2. Resistance strain gauge transducer.

The transducer was originally developed by Ltd (7) (26) and is now manufactured as type PT 30-5 M by Dynamic Instrument Co. Inc. It consists of a cylindrical shaped steel strain generating tube supported at one end, and against the other end rests a thin pressure sensing diaphragm. The outer periphery of the diaphragm is attached to the transducer casing, and as the diaphragm is extremely thin, it rests on the cylinder end in the form of a catenary. Thus when the transducer is subjected to a dynamic pressure change, force in the diaphragm is transmitted to the strain generating tube with very small inertia effect. Strain receiver windings are wound on to the strain generating tube both in the axial and circumferential directions. These windings are connected to form part of a Wheatstone Bridge network which when energised will produce a signal voltage proportional to the sum of the compressive longitudinal and the tensile circumferential strains in the generating tube. The winding is temperature compensated, and as the diaphragm is very thin, combustion flame temperature effects are very small. Figure 5.1. illustrates the arrangement of the connections to the transducer. Rc is a calibration resistor which when switched into the circuit will produce a signal deflection equivalent to a precalibrated pressure. The D.C. preamplifier (Type LR 514) is manufactured by Southern Instrument Ltd. When the transducer is mounted inside the

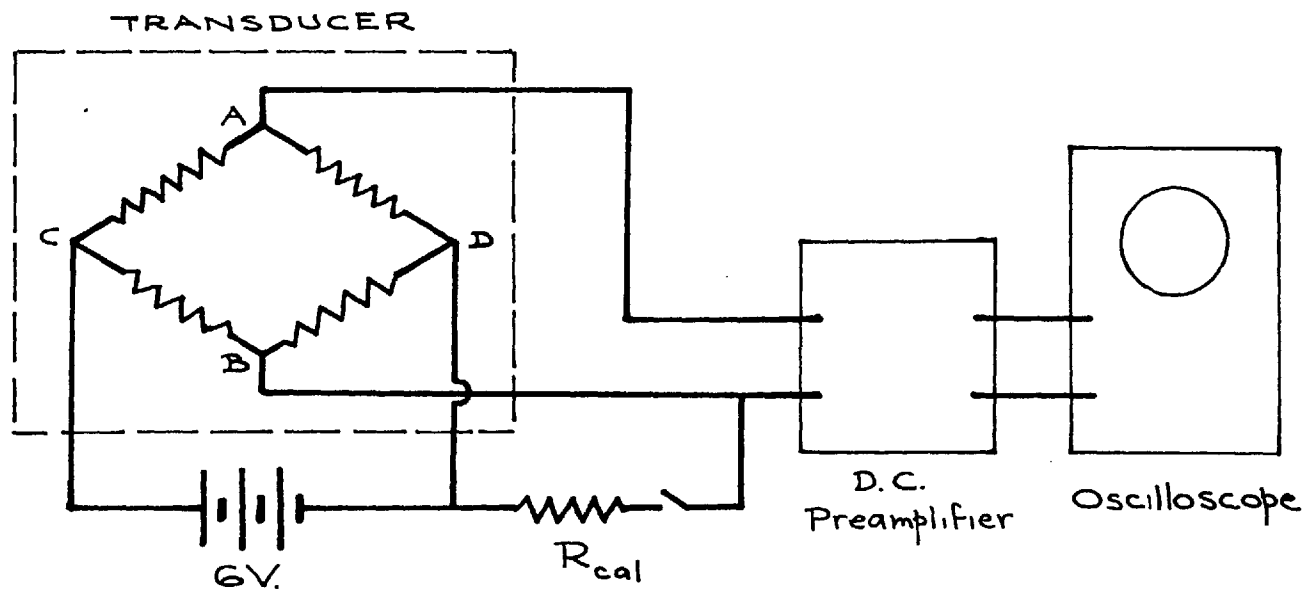
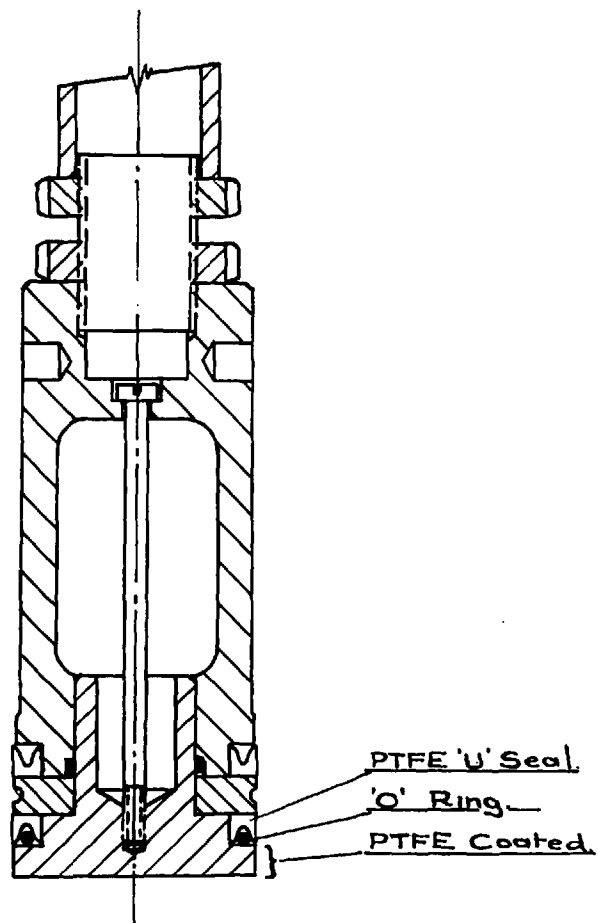
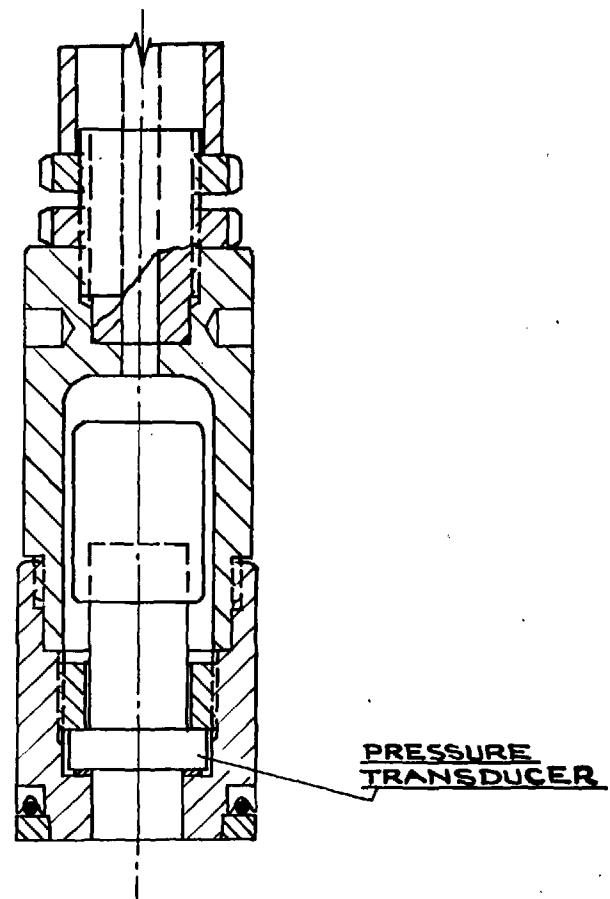


FIG. 5.1. CIRCUIT FOR STRAIN GAUGE PRESSURE TRANSDUCER.



PISTON WITH SEALS
FOR VACUUM & PRESSURE.



PISTON WITH
PRESSURE TRANSDUCER.

FIG. 5.2.

piston in the manner illustrated in figure 5.2, four leads are required to connect the transducer to the ancillary electrical equipment. This is accomplished by using thin screened cables which pass from the transducer through the axial passage inside the connecting rod to the outside of the machine via a side hole in the vicinity of the optical shutter. Since all the leads are connected to the same bridge network, any interference caused by acceleration, inductance or capacitance will be self compensating, provided that the interference occurs to all four leads simultaneously.

5.2. Displacement Measurement.

To measure the displacement of the piston, an optical system is employed which utilises a photosensitive device to convert the displacement into an electrical signal. Figure 5.3. illustrates the layout of the essential part of the system. A 4 volt /1ampere light bulb, made by Cryselco, provided the light source. Two parallel light beams $1\frac{3}{8}$ in apart are deflected from two 45° half inch prisms placed diagonally above and on each side of the bulb. The light from each prism is confined by $\frac{1}{2}$ in diameter collimating lenses of $2\frac{1}{4}$ in focal length. Since the components are arranged so that the light source is at the focal point of both lenses, two parallel beams are obtained. The two beams framed to have rectangular cross-sections, are horizontal and perpendicular to the direction of movement of the piston connecting rod to which is fitted a conically shaped light shutter. They pass on either side of the shutter and are so positioned that the amount of

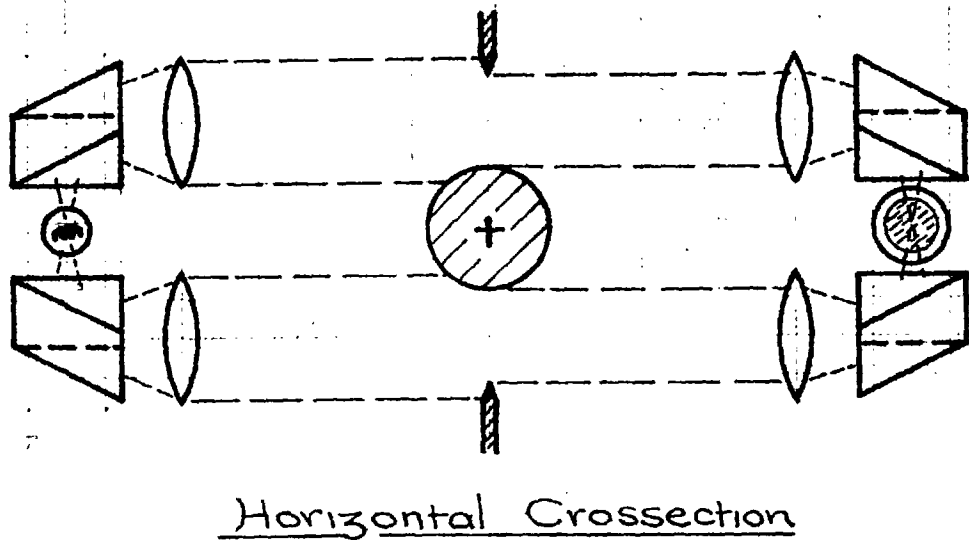
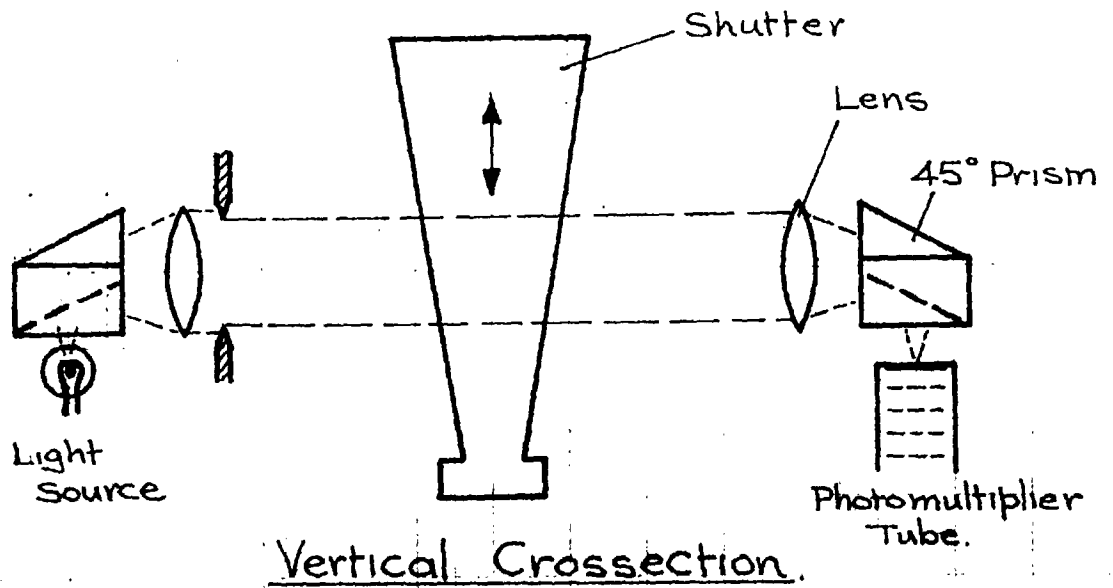
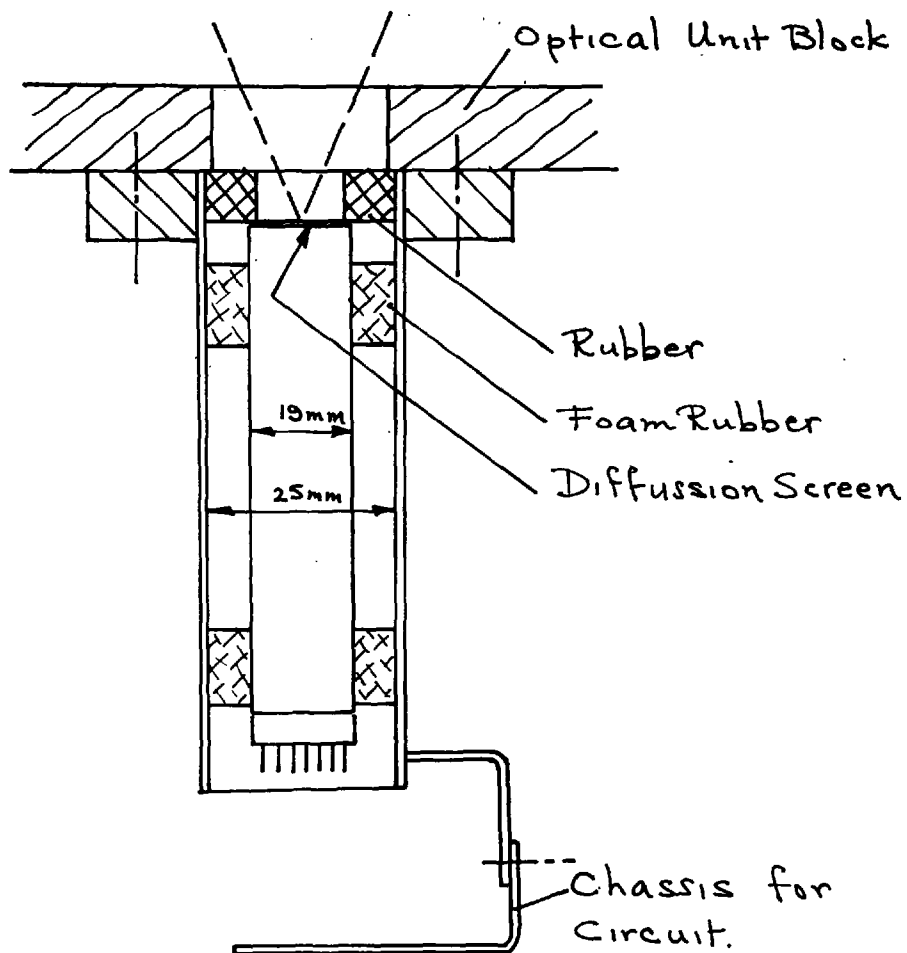


FIG. 5.3. LAYOUT OF OPTICAL SYSTEM.

light obstructed by the shutter is linearly proportional to the displacement of the piston. The split beam arrangement has the advantage of automatically compensating for any lateral vibration of the shutter. The photomultiplier tube receives the light from the two beams via reflecting prisms and lenses and through a paper diffusion screen. All components except the shutter are mounted in two rigid aluminium housings, which are clamped firmly together around the compression cylinder as indicated by part 7 in Fig.4.1. Suitable holes were drilled in the compression cylinder to allow passage of the light beams. The photomultiplier tube is supported elastically by foam rubber packed inside its housing as is shown in Fig.5.4. Both the light source and the photomultiplier tube are placed vertically, as the output signals are least affected by mechanical shock in this arrangement.

The photomultiplier tube Type RBMS 10/14C, manufactured by 20th Century Electronics Ltd., has 10 stages of amplification and it is claimed that it will withstand an acceleration of 100 g. Fig.5.5. shows the circuit diagram of the photomultiplier tube, and a transistor amplifier which amplifies the signal ten times before it is fed into the cathode ray oscilloscope. The circuit was designed so that when the voltage across the photomultiplier tube is 1.35 kV., the maximum and minimum light conditions produced anode current of 0.018 mA and 0.003 mA respectively. This corresponded to output voltages of 8.2 maximum and 1.5 V. minimum to the oscilloscope. The power supply to the photomultiplier



SKETCH OF PHOTOMULTIPLIER TUBE HOUSING.

Fig. 54.

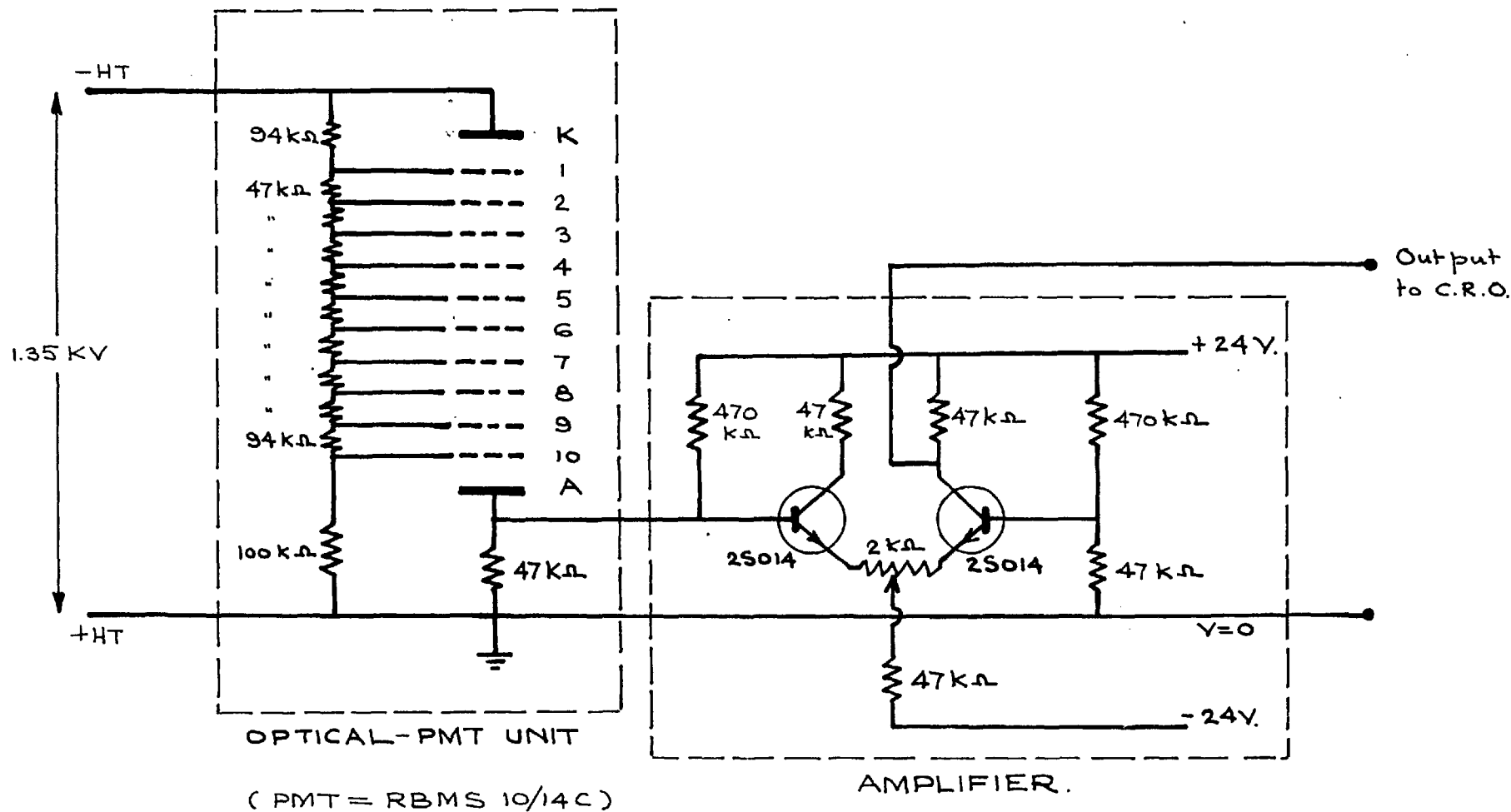


FIG. 5.5. CIRCUIT DIAGRAM OF THE PHOTOMULTIPLIER TUBE AND AMPLIFIER.

unit is from a stabilised variable output E.H.T. power pack, type 2000/5 made by Rank Cintel Ltd. The stabilised ratio of input voltage change to output voltage change is claimed to be better than 100.

The light source is supplied by a 4 Volt accumulator. As the output from the photomultiplier tube is dependent on the applied E.H.T. voltage and the intensity of the light source; and as these quantities may vary from day to day, a special 'step' was built onto the end of the optical shutter (shown in Fig.5.3.), so that the 'step' could provide a calibration mark on the recording trace to indicate the scale of the trace. In Fig.5.6. is a graph showing the linearity of the displacement signal in relation to the actual displacement measured by a dial gauge. It was determined from this graph that the calibration signal provided by the 'step' represented an actual displacement of 2.135 in. A typical displacement record is shown on Run No.298, page 78, in which the straight line marked 'C_D' is the signal provided by the 'step', the vertical distance on the film between C_D and the final level of the trace, after the compression stroke is completed, represent the actual displacement scale of 2.135 in. The calibration mark C_D also serves as an indication of the time when the piston begins to move.

It was estimated that the system is capable of indicating piston displacement to the nearest 0.01 in.

5.3. Recording of observations and synchronization of operations.

Both the pressure and displacement measurements are displayed on a double beam oscilloscope which is coupled to a rotating drum camera.

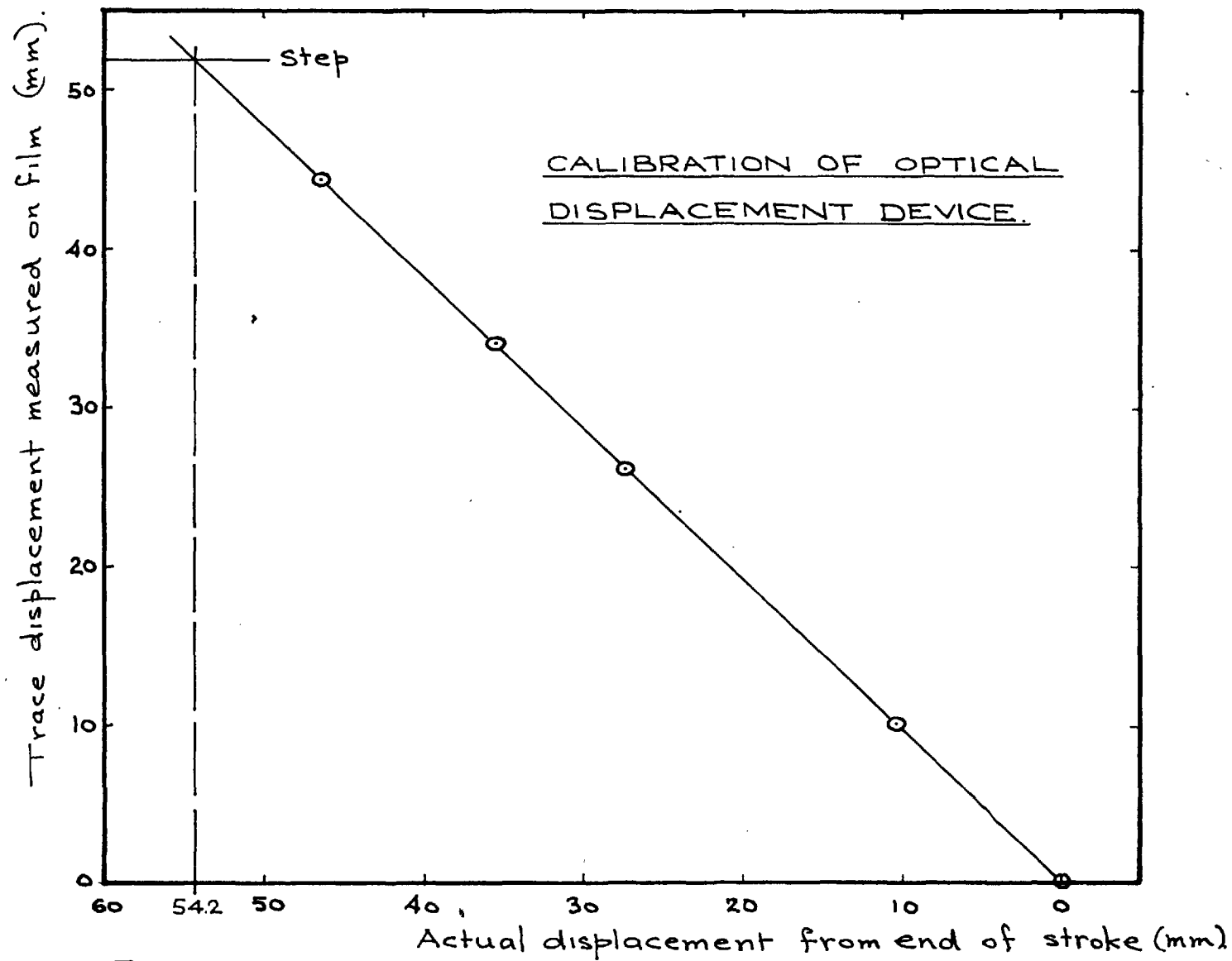


FIG. 5.6.

The drum speed is measured with the aid of a flash type time marker unit, attached to the lens hood, which provides timing marks on the film at 10 msec. intervals. The oscilloscope type M977, drum camera type M371, and time marker unit type M1116 were all supplied by Southern Instrument Ltd.; they are designated as parts 24, 25 and 26 in Fig.4.1, which shows the general arrangement of the apparatus. It is claimed that the frequency response of the oscilloscope is approximately 100 kc., and the accuracy of the time marker unit is within 1%.

A typical sequence of operations is illustrated in Table I, in which the approximate time of the various operations are given.

Table I : Operating sequence of the apparatus.

Time (msec).	Operations.		
0	Release falling weight		Camera drum rotating; record timing marks for drum speed.
250			Pressure and displacement calibration signals on.
270		C.R.O.Beams on.	Record calibration marks Pressure signal return to zero.
300	Weight reaches machine. Machine motion begins Pressure rising.		Record pressure & displacement.
310	Compression completed. Ignition delay. Ignition - further pressure rise.		
370		C.R.O.Beams Off.	Recording completed.

The sequence of operations was controlled mechanically by the pendulum time switch illustrated in Fig.5.7. The operation of the switch is as follows: When the pendulum is in a horizontal position, the microswitch for the electromagnet, which holds the falling weight, is closed. On releasing the pendulum by hand, the microswitch is operated and the falling weight is released. As the pendulum approaches a vertical position, the pendulum rod depresses the microswitch which controls the pressure calibration. When in the vertical position, the contact brush and contact plate close a circuit which triggers the oscilloscope beams, so that the pressure and displacement calibration traces can be recorded by the drum camera. A few milliseconds later, when the microswitch opens, the pressure trace returns to zero. Meanwhile, the oscilloscope beams remain on to record the pressure and displacement characteristics of the experiments until the contact brush reaches the end of the contact plate. The pendulum continues its swing until it is retained by the spring loaded non-return catch.

The pendulum was designed by treating the system as a compound pendulum and arranging for the period of oscillation to be four times the interval of time between the release of the falling weight and the time when the oscilloscope beams are to be switched on. When the unit was made, its actual timing performance was measured and the results are presented in Figs.5.8. and 5.9. It may be seen from Fig.5.8. that the unit can be adjusted so that the pendulum travel time ranges from 0.25 to 0.34 seconds. These times correspond to the times taken for the weight

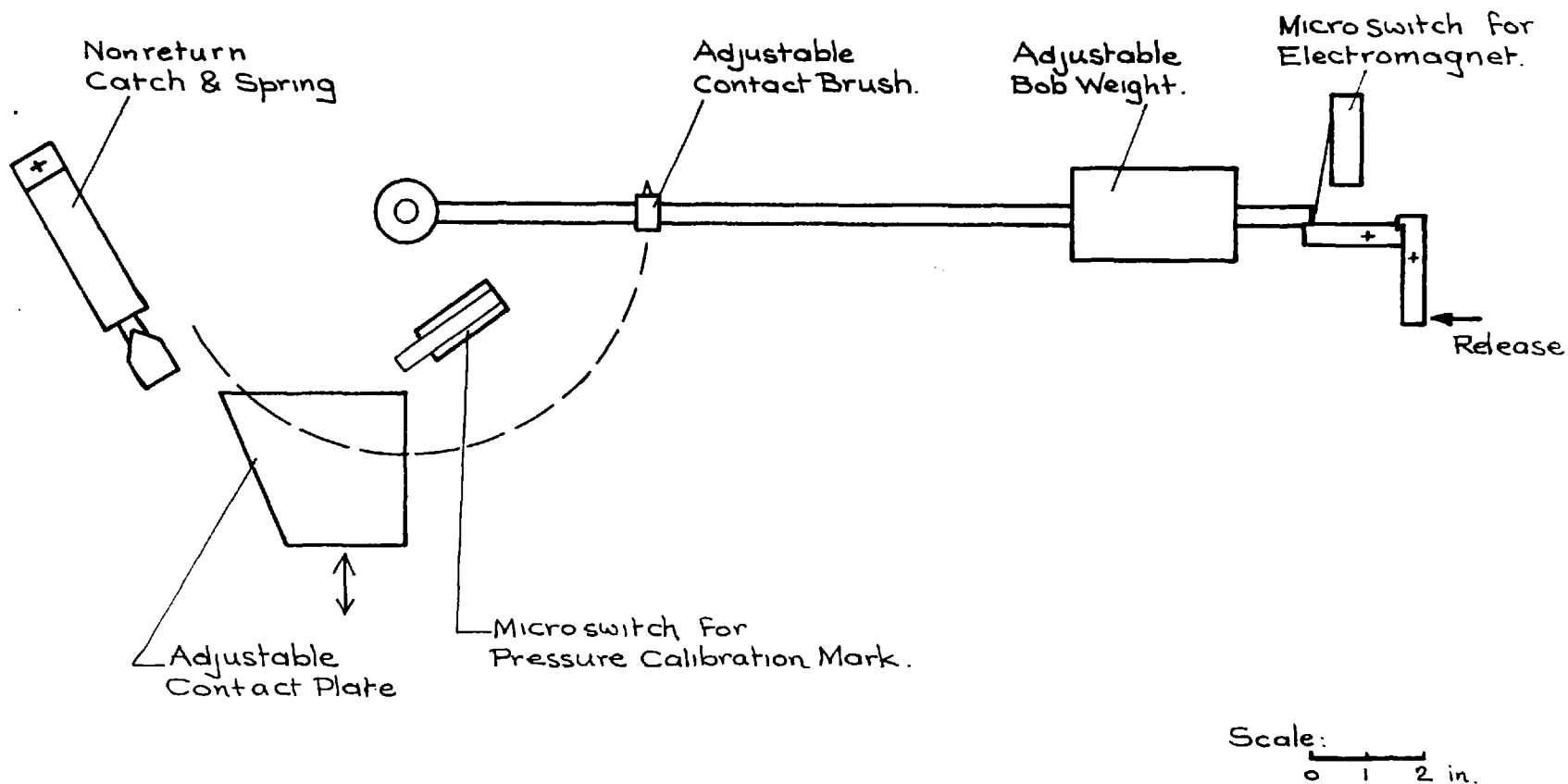


FIG. 5.7. PENDULUM TIME SWITCH

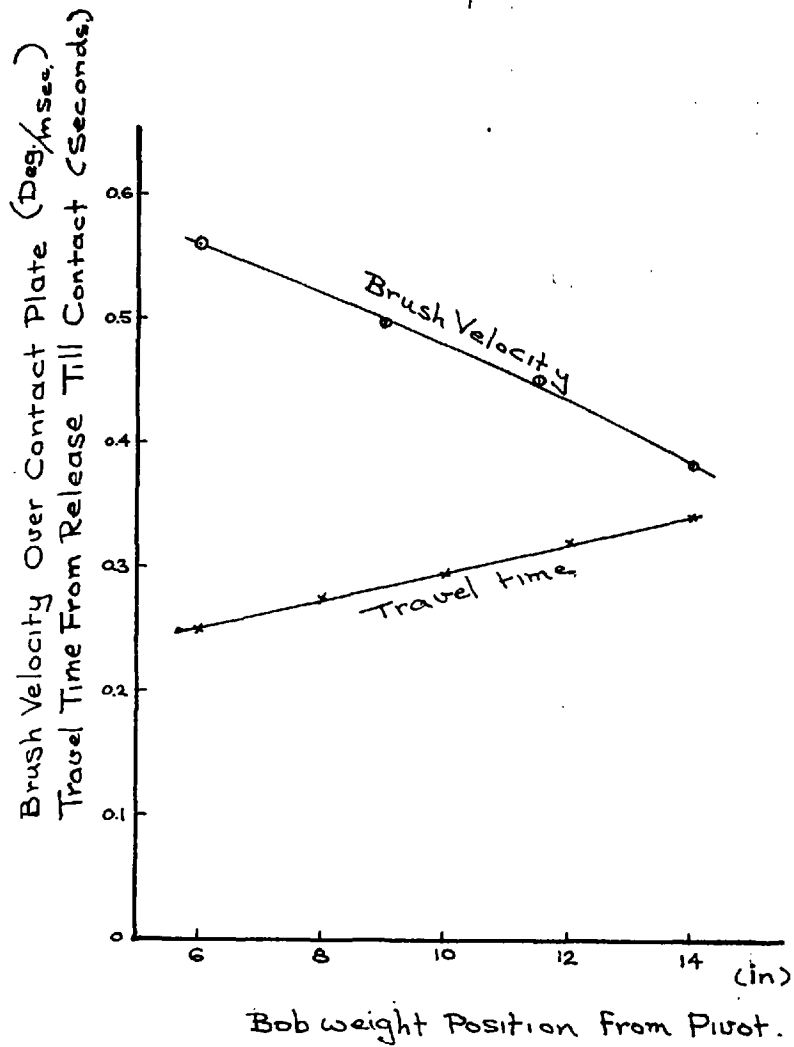


FIG. 5.8.

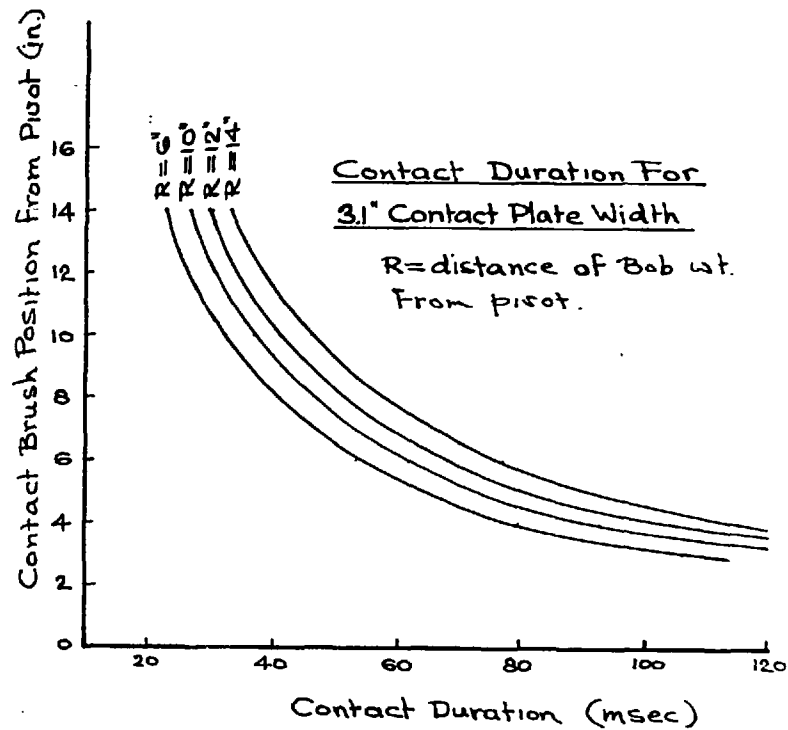


FIG. 5.9.

to fall through a distance of 12 and 20 inches respectively. The contact duration, i.e. the period when the oscilloscope beams are switched on, can be adjusted from 20 to 120 or more milliseconds.

5.4. Pressure controls.

The driving gas and damping gas filling systems are illustrated in Fig.4.1. The pressure gauges were of the Bourdon tube type accurate to $\pm 2\%$. The damping gas pressure gauge is protected by an automatic cut off device (part 17 in Fig.4.1.) which closes the passage to the gauge if pressure exceeds 30 psi. This device was essential to protect the gauge, in case of an omission, to close the isolating valve during an experiment.

In order to compensate for the leakage of gas past the piston rings the damping gas pressure was maintained constant by fitting a pressure regulator and a non-return valve (parts 13 and 14 respectively in Fig.4.1) in the supply line. The pressure regulator, Model 166/11, supplied by I.V. Pressure Controllers Ltd., controlled the pressure to within $\pm 1.3\%$. The gauges and regulator were fixed to an instrument board which was flexibly mounted so as to prevent damage by vibration.

When compressing gas initially at sub-atmospheric pressure, a pressure gauge of the capsule type, supplied by Edwards High Vacuum Ltd., was used. This gauge was calibrated against a mercury column before installation.

5.5. Compression Cylinder temperature control:

The temperature of the compression cylinder is controlled to within ± 0.25 degrees centigrade in the temperature range from $20^{\circ}\text{C}.$ to $100^{\circ}\text{C}.$ Water, mixed with 20% by volume of emulsion oil to prevent rust formation, is stored in an insulated tank and circulated through the cylinder jacket by a centrifugal pump. The water is heated by two electrical immersion heaters and the temperature is controlled by a Sunvic Bimetal thermostat, Type TS3, and a Sunvic hot wire vacuum relay, type 102/3. The mean temperature of the jacket was estimated from the temperatures of the water measured by mercury-in-glass thermometers at the inlet and outlet to the water jacket.

CHAPTER 6.

6. DEVELOPMENT OF APPARATUS.

The development of the apparatus is described in this chapter under separate headings according to the parts concerned, although in practice many aspects of the work were carried out concurrently.

The operating procedure for the machine is outlined in Appendix H.

6.1. Development of the machine.

The main problems in this phase of development concerned the adjustment of the damping gas exhaust ports and reducing the effect of vibration on the instrumentation.

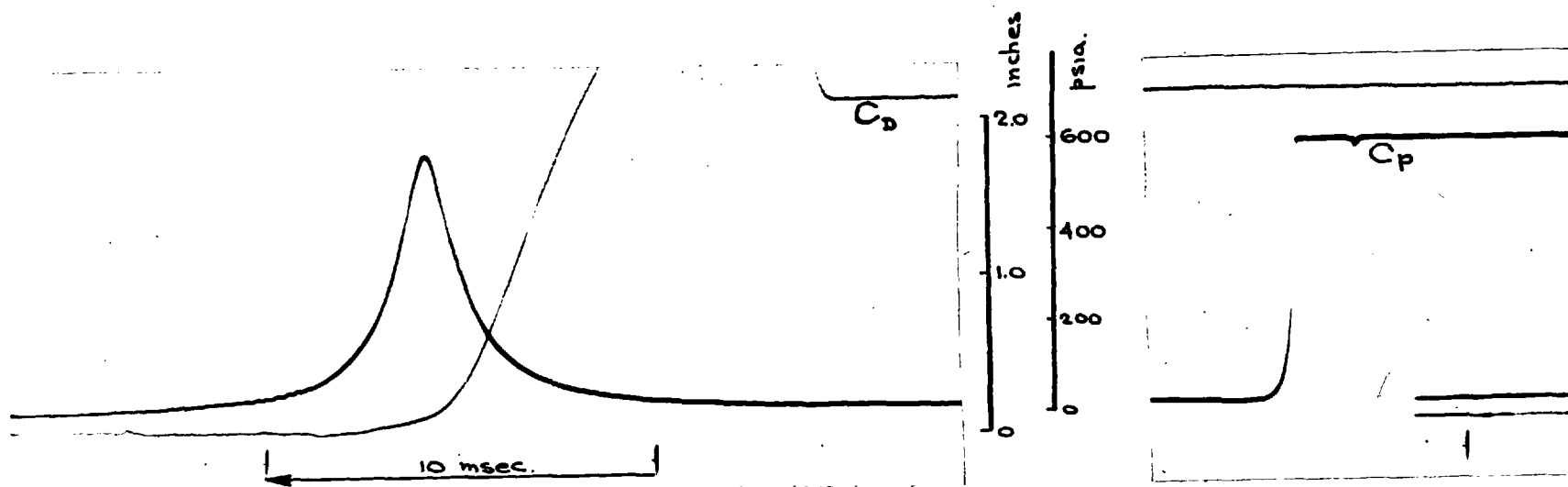
6.1.1. Adjustment of damping gas exhaust ports.

It was thought that the use of small exhaust ports $\frac{7}{32}$ in. high located so that a late port opening of $x_e = 5.03$ in. would cause the piston to decelerate in the shortest time possible. However, it appears that the arrangement did not provide sufficient venting in the time available. Displacing the cylinder liner (part 10 in Fig.3.1.) upwards so that the exhaust ports opened earlier in the stroke at $x_e = 4.905$ in. did not show any improvement. The dip in the pressure trace after reaching a peak value, illustrated in Runs 54 and 86 on page 83, indicate that the piston rebounded before reaching the end of the stroke. The evidence confirmed that the ports were too small. After enlarging the port height in the drive piston and in the liner to $\frac{3}{8}$ in., and adjusting the position of the liner so that the ports opened at $x_e = 4.85$ in., smooth landings were achieved and the pressure records were satisfactory.

According to the theory of the operation of the machine given in section 3.3.3., the damping gas pressure will continue to increase after the exhaust ports begin to open. To determine the piston position at which the pressure was a maximum, the damping gas pressure was measured with the aid of the S L M piezoelectric transducer through a passage in the midflange of the machine. The record of the measurement is Run No. 298 shown on page 78. The passage through which the pressure was measured is 10in long and $\frac{1}{8}$ in. in diameter. After correcting, as in appendix G, for the time required for the pressure wave to travel through the 10in. long passage, the peak damping gas pressure was found to occur at $x = 5.10$ in. which is 0.25in. after the ports began to open. This is illustrated by the damping gas force v piston displacement curve in fig. 6.1. From the area under the curve for the positions when the ports are opened, the value of K in equation (3) in section 3.3.1. is found to be equal to 1.0.

6.1.2. Vibration.

The chief sources of vibration in the machine are the pressure release valve (part 8 in fig. 3.1.) and the drive piston should it land heavily as a result of some maladjustment of the machine. The shocks chiefly affect the instrument mounted on the machine such as the displacement measuring unit, but the shock transmitted to the laboratory floor caused some concern. To determine the extent of vibration, a piezoelectric accelerometer, type E supplied by G.E.C.Ltd, was mounted on the compression cylinder head during operation of the machine and its output signals displayed on the oscilloscope. The accelerometer was not



No. 298

Measurement of damping gas pressure p_d

C_D = Displacement trace calibration mark.

C_P = Pressure trace

$p_d = 12.05 \text{ psig}$

$p_{k1}/p_d = 6.75$

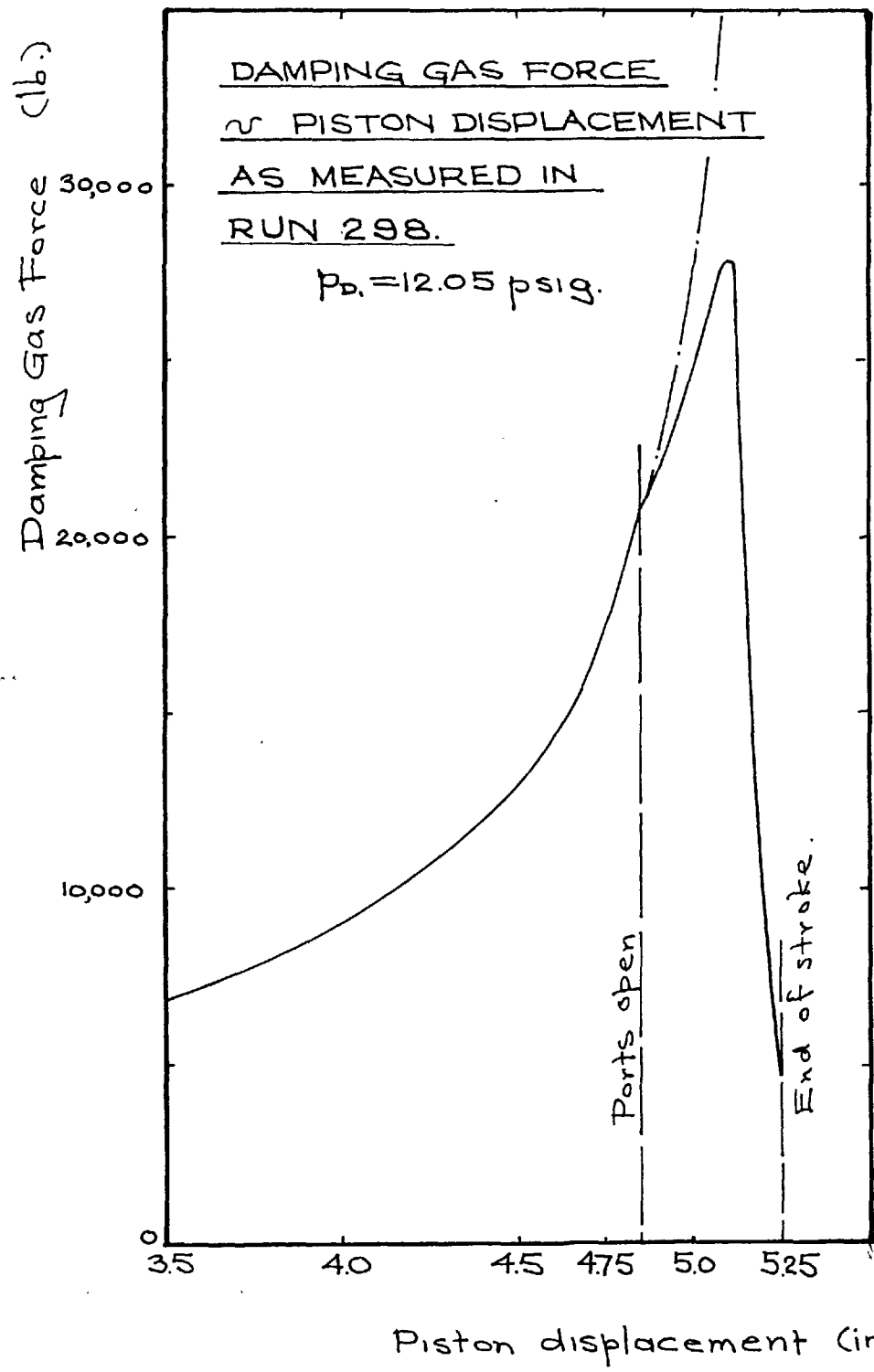


FIGURE 6.1.

calibrated as the signals were only used for the purpose of comparison. When the displacement measuring system had not been perfected, signals recorded from the accelerometer enabled a distinction to be made between a good and a 'hard' landing of the piston. The vibration resulting from the different landing impacts are illustrated in Runs No. 46 and 47 on page 81 . The accelerometer was also used to identify the effects of other sources of vibration, such as those arising from the movement of the drive pressure release valve, and those caused by the impact of the falling weight.

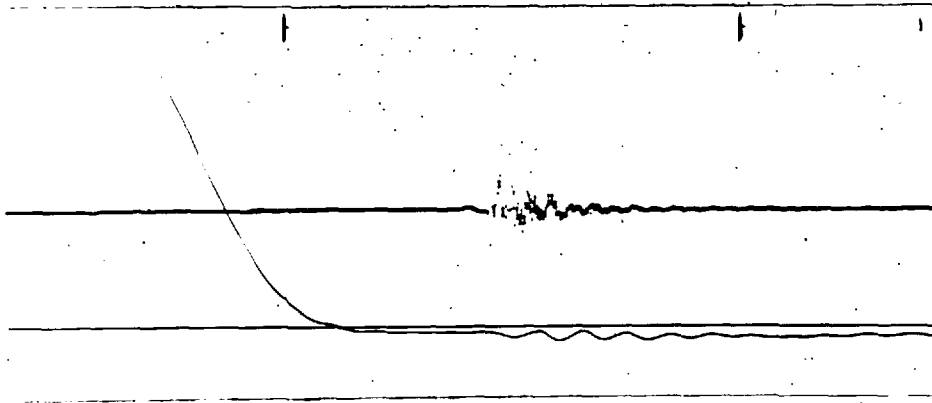
6.1.3. Noise.

The damping gas is released at a pressure of approximately 600 psi into the atmosphere through the exhaust ports within a period of one millisecond. The generated shock wave is audible as a very sharp loud 'bang' in the laboratory. To dampen the noise, cylindrical silencers made of porous polythene and manufactured by Martonair Ltd. were fitted to the exit holes of the cylinder casing. The flow resistance (less than 15% of the full flow) did not appear to affect the machine operation.

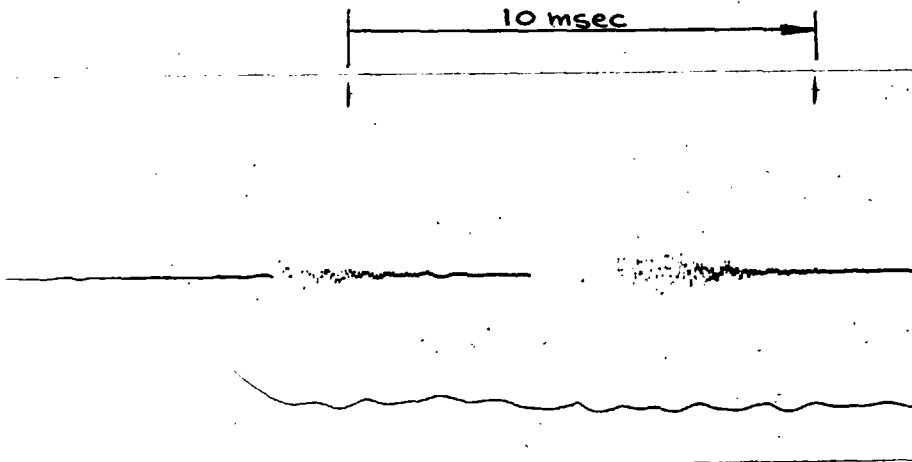
6.2. Displacement measurement unit.

6.2.1. Photomultiplier tube.

The optical layout of the system as shown in fig. 5.3. was used throughout the development work. In the first attempt to employ the system a photocell, type 58 CV manufactured by Mullard Ltd., was connected in series with a 45 V dry battery to the input of the



No. 46



No. 47

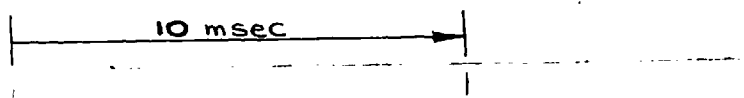
Displacement
trace failed
to indicate
piston bounce.

Displacement traces by photocell and vibration traces
by accelerometer.

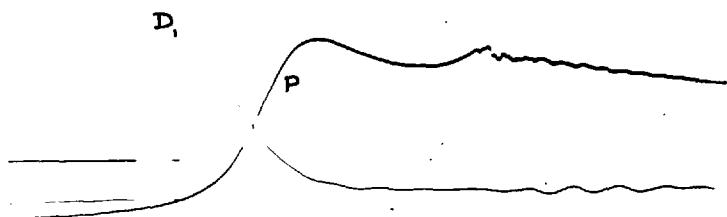
oscilloscope. The unit was calibrated statically and produced a linear displacement characteristic. However, when the piston displacement during an actual experiment was recorded, the trace did not often correlate with that of the pressure signal. In Run No 54 on page 83, the pressure trace indicated a fall in pressure after attaining a maximum value; this could only be explained by a rebound, although no evidence of this was obtained from the displacement trace. This led to the belief that the response time of the photocell unit was too slow. To check this, a Xenon discharge lamp with a flash duration of $5 \mu\text{sec}$. was directed on the photocell and the signal rise and decay times were recorded. It was observed that the decay time of the photocell signal was in the order of 3.5 msec . which explained why the displacement trace did not indicate the piston bounce, the duration of which was probably less than 3 msec .

It was found possible to improve the response time to 0.3 msec by adding to the circuit a transistor amplifier designed to match the output impedance of the photocell with the input impedance of the oscilloscope. However, recordings from further experiments indicated that the photocell was sensitive to vibration even when suspended in foam rubber. This phenomenon was not apparent with the initial circuit because of the sluggish response. The attempt to use the photocell was then abandoned in favour of a photomultiplier tube.

Three different photomultiplier tubes were tried. These were a) 931 A, a 9 stage photomultiplier manufactured by R.C.A. Ltd., b) 9592, an 11 stage photomultiplier tube by E.M.I. Ltd., and c) RH4S 10/14C, a



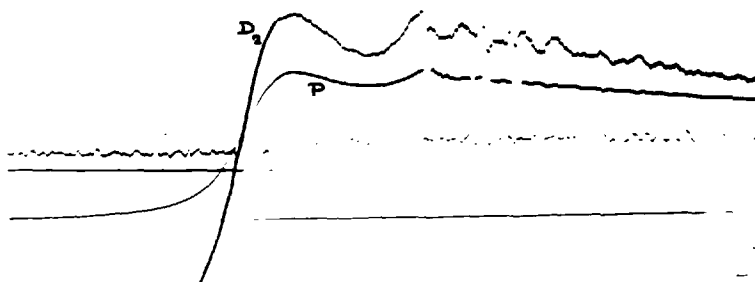
RUN NO. 54



D_1 = Displacement trace
by photocell.

P = Pressure trace by
transducer PZ14

RUN NO. 86



D_2 = Displacement trace
by photomultiplier.

P = Pressure trace by
transducer PZ14.

Piston bounce shown by traces P and D_2 but not
by the photocell trace D_1 .

ten stage photomultiplier of especially rugged construction made by 20th Century Electronics Ltd. The first two tubes were unable to withstand the vibration of the machine and were rejected; but the increased sensitivity of the photomultiplier tubes enabled a response time of 0.15 msec. to be achieved, which was half that obtained for the photocell.

Many methods were tried to deal with the vibration problems. The main approach was to focus the light from the optical unit on to the smallest possible area of a diffusion screen, and to permit movement of the photomultiplier so long as the incident light from the diffusion screen fell on the sensing area of the tube. This was achieved by suspending the photomultiplier tube housing assembly on soft springs, so that the shocks from the machine were damped out although a vertical movement up to $\frac{1}{8}$ " occurred between the photomultiplier tube and the machine. This method was only partially successful, as may be seen from Run No.86, p.83, which is an example of a displacement trace using this arrangement. The displacement trace marked D indicated the rebound of the piston satisfactorily; but the drifting base line of the displacement trace was caused by the movement of the photomultiplier tube. Vibration disturbance from the machine is still present.

The final arrangement employs tube REMS 10/14C, which is claimed by the manufacturers to be able to withstand an acceleration of 100 g. The tube is held by friction with foam rubber inside a rigid housing as sketched in Fig.5.4. The performance of the new tube, which was used from Run No.133 onwards, appeared to be very satisfactory. The base line of the signal which indicates the end of the stroke is undisturbed by vibration, unless the foam rubber inside the tube housing loses its resilience. For example, in Run No.275 on p.110, the disturbance on the base line is caused by lack of resilience and, in these circumstances, it is necessary to repack

the foam rubber. This is usually required only after approximately 40 or 50 runs.

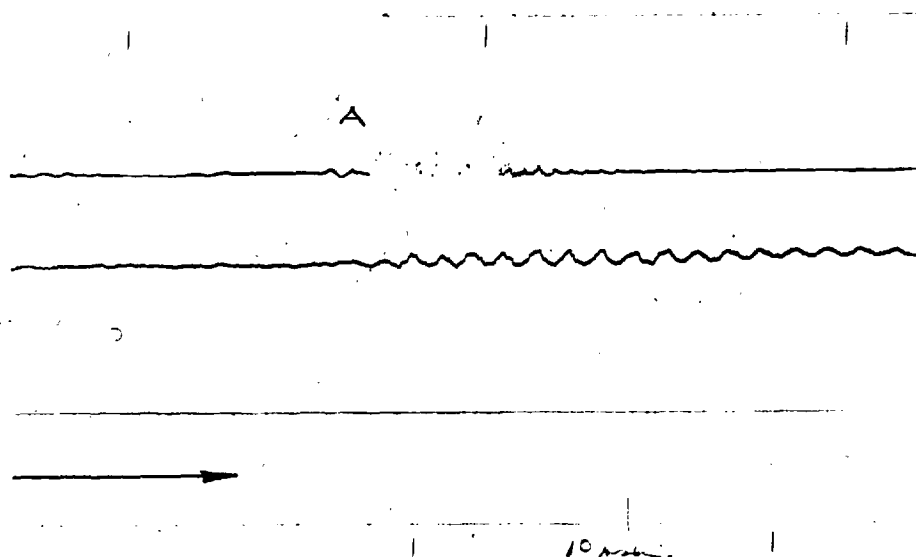
6.2.2. Elimination of vibration effect on the light source.

During the process of eliminating various vibration ripples in the signal, a shutter of uniform gap which did not cause any signal deflection during a piston stroke was found very useful. Runs No.66 and 69 on page 86 are two examples of the use of such a device. Points A in the accelerometer traces indicate the vibration generated by the piston on reaching the end of the stroke. In Run No.66, the light source was disturbed by the vibration. After modification to the light source, no further disturbance was recorded, as shown in Run No.69. In the process of modifying the light source, the only type of small filament light bulb, which was found suitable, was type SES manufactured by Cryselco; and only when the bulb was fixed vertically inside the holder by means of grub screws instead of the usual layout arrangement, did it become free from vibration.

6.3. Pressure recording.

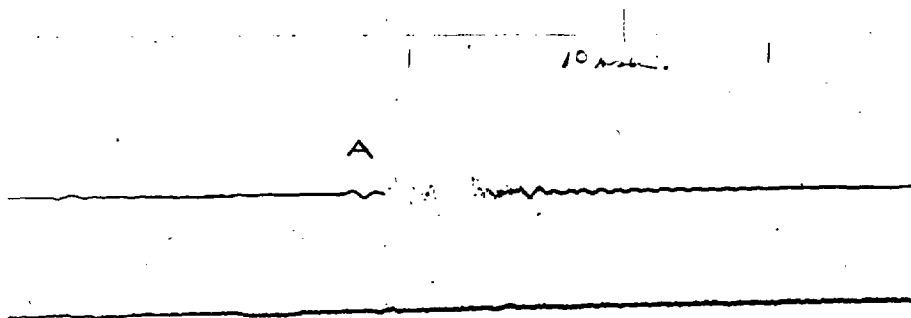
The SLM PZ14 piezoelectric transducer appeared to perform satisfactorily when mounted into the machine, as shown in Fig.6.2. However, a test regarding its vibration sensitivity was carried out.

In the special test, the transducer was rigidly fixed to the machine in the vicinity of the cylinder head, so that when compression took place it was subject to the same acceleration without any increase in gas pressure. The results showed that, when the machine is driven by a drive gas pressure of $p_{K1} = 150$ psi, a negative pressure signal



No. 66

Light source
vibrating.



No. 69

Modified
light source

Elimination of disturbance caused by vibration of the light source.

'A' is vibration indicated by an accelerometer when the piston reached the end of stroke. A photomultiplier was used in conjunction with a straight optical shutter to detect light source vibration.

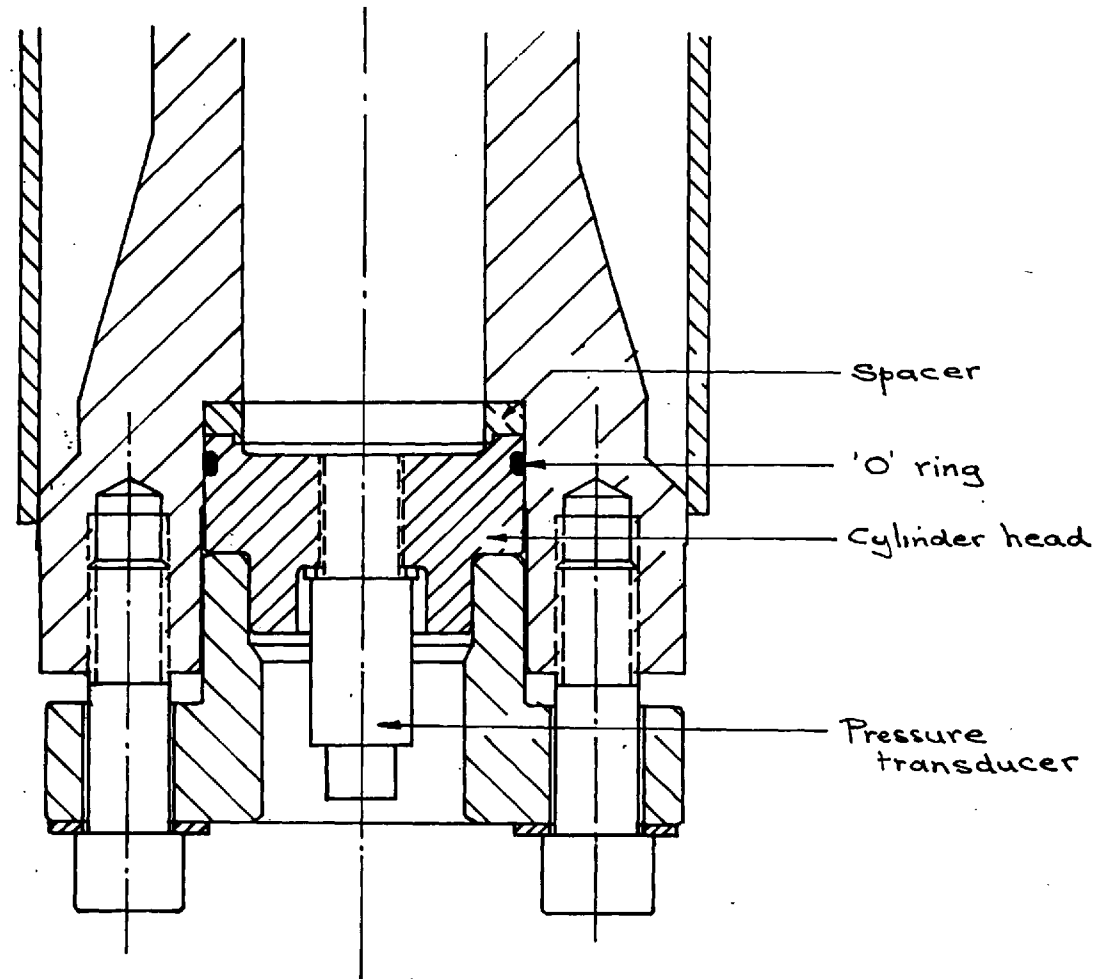


Fig. 6.2. ASSEMBLY OF CYLINDER HEAD.

deflection occurs during the period 2 msec. before and 2 msec. after the piston landing. The magnitude of the deflection was about - 3 psi at atmospheric pressure.

When measuring pressures below 300 psi, the vibration of the machine was found to cause a high frequency ripple of approximately 15 kc on the pressure trace at the instant when the piston landed. As its duration was usually less than 4 msec., it did not obscure the pressure record and no attempt was made to eliminate it.

The pressure calibration mark is switched by the 'external calibration' circuit of the electrometer amplifier. Because of stray capacitance effect in the external leads of the circuit, the pressure trace required approximately 7 msec. to return to zero after calibration. Hence, in all the pressure recordings, the zero level was measured at a position corresponding to a 10 msec. time delay after the calibration signal.

6.4. Release of falling weight by electromagnet.

The electromagnet (part 19 in Fig.4.1.) has a continuous rating of 12 Volts D.C. and 0.2 Amp. It was tested to hold a weight of 50 lb., which was released when the voltage was reduced to 1.5 Volts. In the present system, a falling weight of seven pounds provided sufficient energy to release the valve. The comparatively small mild steel weight was easily magnetised permanently by the electromagnet. Consequently, difficulties were experienced in the synchronisation of the machine operation because of the release delay.

The problem was overcome by placing a layer of paper 0.014" thick between the electromagnet and the weight so as to reduce the intensity of the magnetic flux. In addition, a $\frac{1}{2}$ micro Farad condenser was connected in series with a 2.7 ohm resistor across the control switch so as to prevent self-induction in the magnet winding and secure a rapid release. The release time was reduced to between 30 and 50 milliseconds, a period which is easily allowed for by adjustment of the pendulum switch unit.

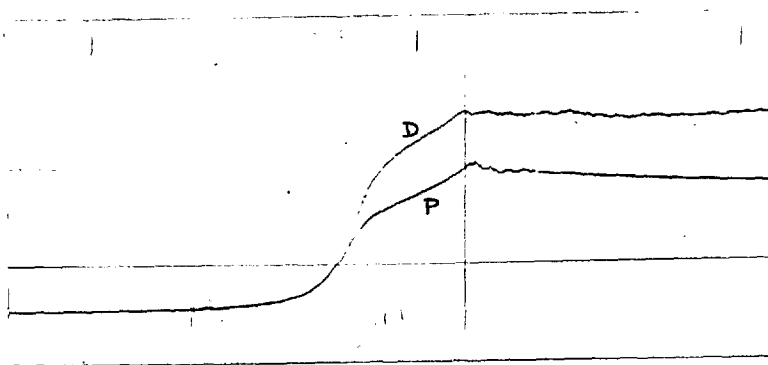
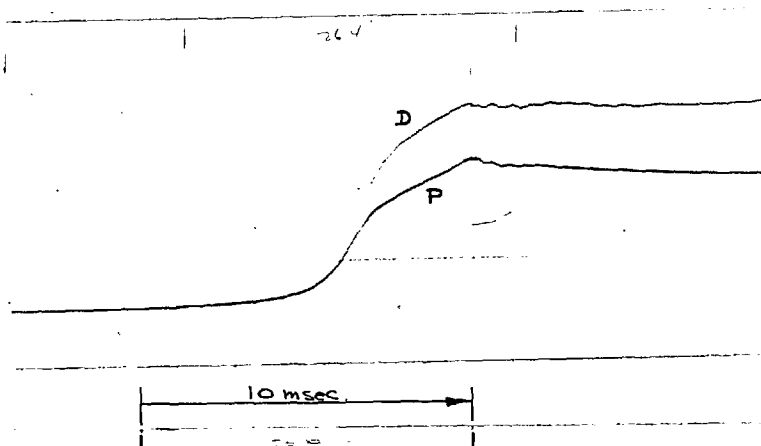
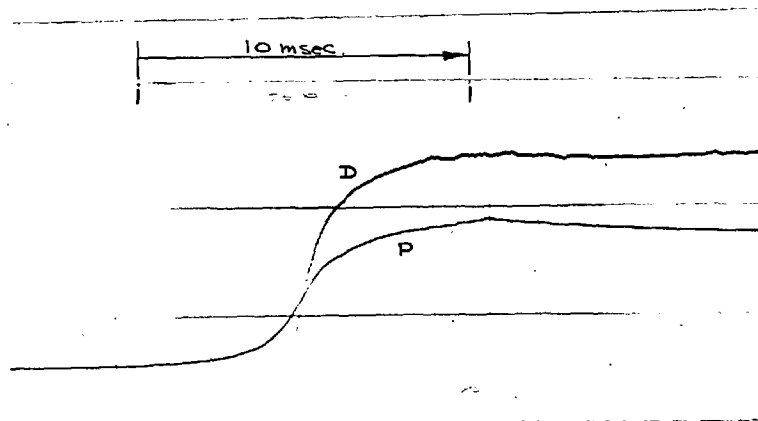
CHAPTER 7.

Evaluation and discussion on the performance of the compression machine.

7.1. Compression times.

The machine has not been operated by driving pressures in excess of 200 psi, as compression times of 12 msec. were found to be adequate for the tests. In this range, the compression times were found in most cases to be within $\pm 10\%$ of the predicted values, nevertheless durations of up to 17% greater have been recorded. A few typical experimental values are plotted in Fig.3.9. to show the agreement between the experimental values and those predicted in Section 3.4. The occasional large deviation from the practical value is thought to arise from the lubricating grease which causes the top face of the drive piston to stick against the bottom face of the drive gas container. At times it was noted that forces of approximately 200 lb. were necessary to separate the two components.

It was found that, by restricting the exhaust passage of the damping gas, i.e. by closing two or more of the valves '11' in Fig.3.1., it was possible to adjust the time taken for the piston to cover the last 0.1 in. of the stroke without any alteration to the initial drive and damping gas pressures. The traces of Run Nos.136, 137 and 138 on page 91, and Table I illustrate this effect.

No. 1368 silencers
open.No. 1377 silencers
open.No. 1385 silencers
open.

Effect on stroke time by restriction of damping gas
exhaust passage.

$$p_{d1} = 6.0 \text{ psig}$$

$$P_k / p_{d1} = 6.75$$

$$C.R. = 25.2$$

D = displacement trace.

P = pressure trace.

Table I. : Effect of damping gas exhaust passage size
on compression time.

Run No.	P_{KL} (psig.)	P_{KL}/P_{DL}	Compression ratio.	Damping gas exhaust passage.	Compression time. (msec.).
136	125	6.75	25.2	8/8	12.6
137	"	"	"	7/8	13.1
138	"	"	"	5/8	13.9

7.2. Initial drive gas and damping gas pressure settings.

The ratio of correct initial drive gas pressure to damping gas pressure setting $\frac{P_{KL}}{P_{DL}}$ was found to be below the value of 6.97 calculated in Section 3.4. The reason for the difference is thought to arise from errors in the estimated internal volume of the damping gas space; and in the assumed value of K in Equation (3).

The experimentally determined values of $\frac{P_{KL}}{P_{DL}}$ to secure good piston landings, were found to lie between 6.75 and 6.85 for any compression ratio up to 24:1 when using drive gas pressures below 200 psig. In practice, the ratio $\frac{P_{KL}}{P_{DL}}$ can be maintained within these limits without difficulty; for example if P_{DL} is 12 psig. exactly, the pressure with which it is necessary to set P_{KL} is 167 psig. ± 1.5 psi. However, when operating with a compression ratio of 30:1 using drive gas pressure below 200 psig., the value for $\frac{P_{KL}}{P_{DL}}$ becomes greater and its limits for

successful operation narrows : in this case the limits are between 6.86 and 6.89. This results from the fact that much higher compression pressures are produced with the higher compression ratios. For example, when the piston is within 0.1" of the end of its stroke, and the damping gas is escaping from the ports, the upward force arising from the compression pressure may increase from 1400 lb. to 2700 lb.; this force is sufficiently large to influence the piston motion. It is expected, however, that when drive gas pressures above 300 psig. are used, the conditions for operating at a 30:1 compression ratio will become less critical, because of the greater driving force present.

7.3. Compression of Air.

In assessing the performance of the machine from a compression point of view, a series of runs in which air was compressed was carried out. The leakage rate, the reproducibility of pressures from identical initial conditions, the apparent adiabatic exponent γ and the proportion of heat lost due to heat transfer through the cylinder walls were determined.

A special cylinder head, made to accommodate a short gas filling passage and a pressure gauge, was used to test the seals on the compression piston for leakage. A newly assembled seal was found to contain a gas pressure of 1500 psi without any leakage for periods of more than 15 minutes. After 80 runs, the same test was repeated and the leakage rate then found was of the order of 0.6 psi/second.

Results of various experiments in which air was compressed under various initial conditions are given in Table II. The apparent adiabatic exponent γ^* , calculated from the final and initial air pressures and the compression ratio, appears to decrease with increasing compression ratio. This is because the surface to volume ratios are slightly greater at higher compressions, and also because the temperature gradients in the cylinder walls are steeper. Both of these factors increase the proportion of heat lost. Variations in the initial cylinder and gas temperature do not appear to influence the values of γ , as was first discovered by Kroger and Gondermann⁽¹⁹⁾. The ratio of the absolute temperature after compression, calculated from the experimentally recorded pressures and the theoretically calculated temperature for isentropic compression, as shown in Appendix I is termed the adiabatic efficiency. Since this is a measure of the heat lost, it is lower at higher compression ratios as expected. Tests on the reproducibility of the final compression pressures and temperatures show that, in general, the results are reproducible to within $\pm 1.5\%$, but at compression ratios above 20:1 the limits broaden to $\pm 3.5\%$. This increase in the random experimental error at higher compression ratios may be caused by the slight movement of the neoprene 'O' ring seals in the cylinder head during compression. The uncertainty introduced by such an effect will be more important at high compression ratios. On the other hand, the estimated accuracy of the pressure measuring system, using the piezoelectric pressure transducer, is within $\pm 1\%$.

* Strictly speaking, the term γ should be called the exponent for polytropic compression, but as the compression is found to be very nearly adiabatic, it is called 'apparent adiabatic exponent' throughout this thesis.

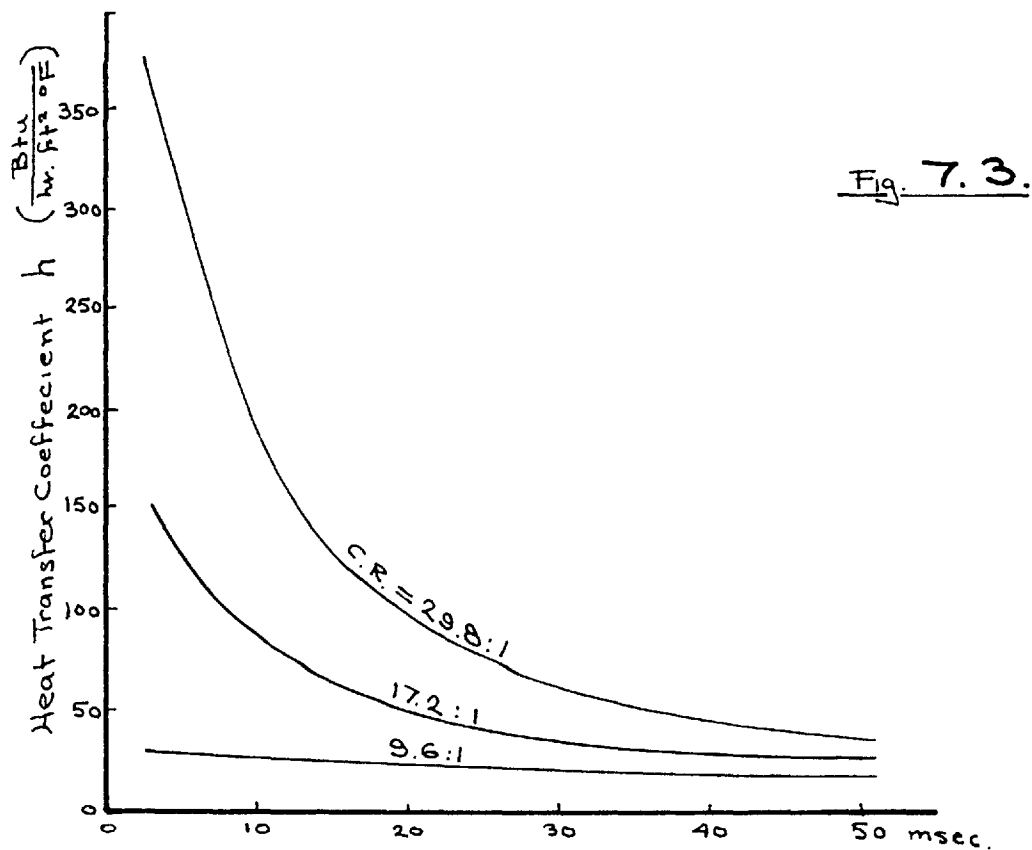
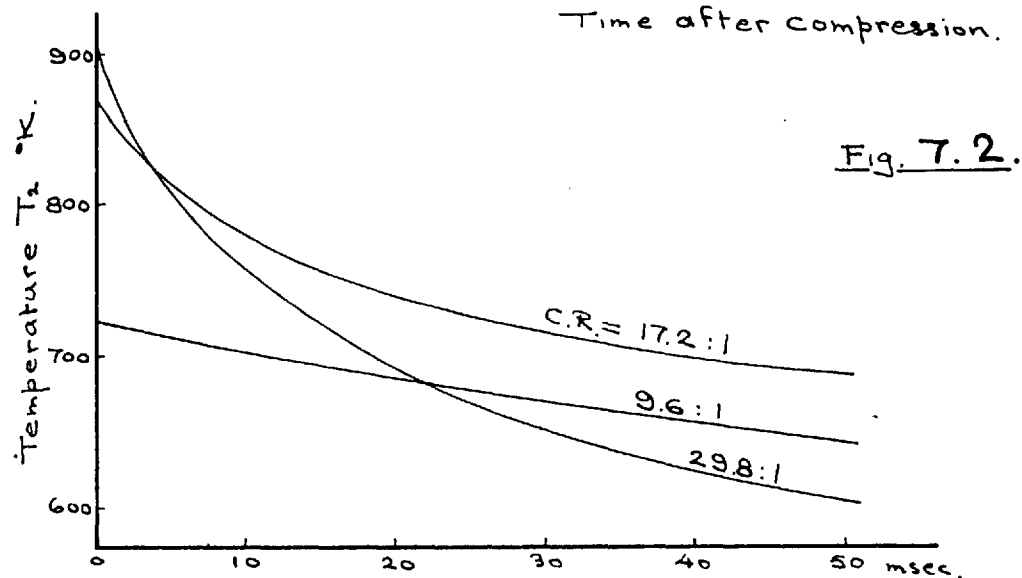
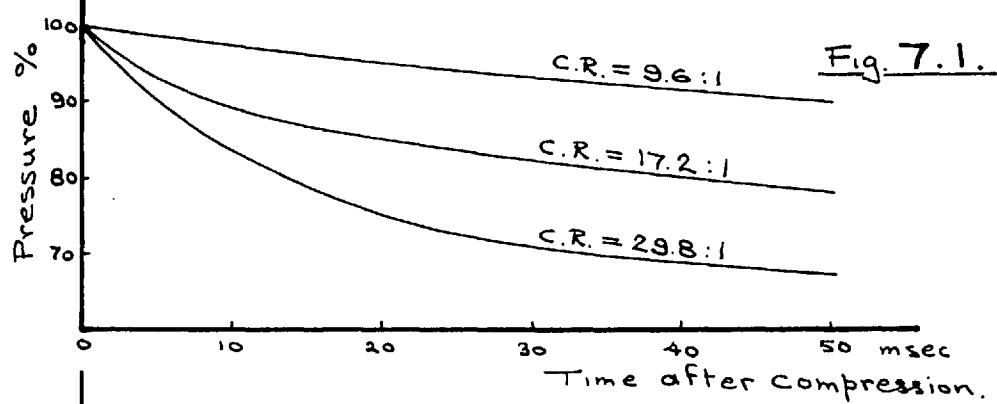
Table II. : Compression of Air.

Run No.	Compression Ratio.	Initial gas & cylinder temperature °C.	Drive gas pressure P_{K1} psig.	Initial air pressure, P_1 psia.	Final air pressure, P_2 psia.	Apparent adiabatic exponent γ	Adiabatic efficiency. %
159	8.58	58	171	14.8	268	1.350	93.8
157	"	45	176	14.8	276	1.361	96.5
153	"	24	179	14.7	276	1.365	95.7
160	"	19	168	14.8	271	1.352	93.0
246	9.94	40	168	14.7	338	1.368	96.8
330	11.0	17	154	14.9	401	1.370	96.9
279	14.2	40	159	14.7	519	1.342	91.5
247	17.2	40	167	14.8	701	1.358	95.3
161	20.0	20	171	14.8	862	1.359	95.5
154	22.8	18	173	14.7	999	1.351	92.9
155	"	48	172	14.7	919	1.324	87.0
156	"	61	172	14.7	937	1.331	88.7
248	23.5	40	166	14.8	985	1.333	89.0
251	29.8	40	167	14.8	1301	1.320	85.4
281	"	40	160	14.8	1382	1.337	90.6

Thus the same order of accuracy applies to the measurement of pressure and the calculation of temperature in each run.

After compression is completed, the maximum pressure is observed to fall gradually. Since the leakage rate from the cylinder is practically zero, the observed pressure drop must be caused by the heat lost through the walls. In Fig.7.1. the pressures at the end of the piston stroke are plotted as a function of time. It may be noted that the percentage drop for a given time is greater at higher compression ratios. This is because of the steeper temperature gradients near the walls, and the greater surface to volume ratios when compression ratios are high. An estimate of the amount of heat lost during compression was made by measuring the rate of pressure drop after compression, and by taking into consideration the fact that the surface to volume ratio as well as the difference between the gas temperature and wall temperature are continually changing during the compression period. By this method, it was found that the loss of pressure during compression is approximately 0.7% for compression ratios up to 31:1. (See Appendix J). The amount of heat lost in this way constitutes only a small proportion of the heat loss as indicated by the adiabatic efficiency tabulated in Table II. The unaccounted loss is thought to be due to frictional effect in the gas under turbulent conditions.

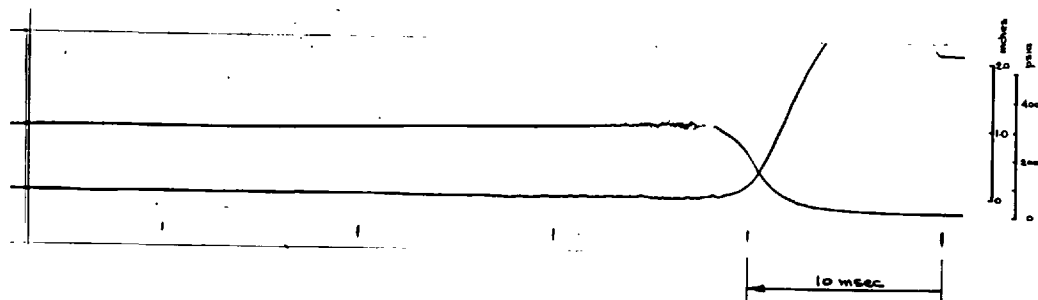
Fig.7.2. shows the cooling curves of air after compression; the temperatures being calculated from the pressure measurements. From the instantaneous rates of cooling, the apparent overall heat transfer



coefficients 'h' between the air and the walls were calculated and are plotted in Fig.7.3. It appears that values of h increase with the compression ratios. 30 to 50 msec. after compression, h decreases to a steady value between 30 and 40 Btu/ft.hr.[°]F., irrespective of the compression ratio. Higher values of h are generally associated with turbulence. Comparing the heat transfer coefficient for compression ratios of 30:1 and 17:1 with those calculated, according to an equation by Annand⁽¹⁾, for heat transfer across engine cylinder walls, the values correspond to the conditions of 'high swirl' and 'low swirl', respectively, in a motored engine. (See Appendix K).

Livingood and Leary⁽²⁷⁾ took schlieren photographs of compressed dry air through a transparent window in the cylinder head of the M.I.T. rapid compression machine, and recorded a sequence of photographs at various time intervals after compression was completed. With conditions of C.R. = 12.6:1, $p_1 = 15.5$ psia and $T_1 = 150^{\circ}$ F., the photographs showed that the compressed air was not optically homogeneous, indicating that there were temperature gradients persisting for at least 250 msec. During the first 28 msec. after compression, hot spots were seen near the edge of the chamber; after 86 msec. the hot spots became less obvious and the temperature more uniform. The rate of decay of the temperature gradients is comparable to the rate of decrease of the heat transfer coefficients calculated in this project.

The final rate of temperature rise in the cylinder is found to be very similar to the rates achieved by Martinengo⁽³²⁾. In Run 246, on page 100, where air was compressed under a compression ratio of 9.94:1 from an initial temperature of 313°K to reach a final temperature of 730°K in 12 msec., the temperature rise during the last millisecond was 36°C.



No 246

Air only.

C.R. = 294

$P_1 = 14.7$ psia

CHAPTER 8.

Compression and ignition of n-heptane air mixtures.

To assess the performance of the machine, an investigation of the compression ignition of n-heptane air mixtures was undertaken with the object of comparing the results obtained with those of Taylor⁽³⁹⁾, Martinengo⁽³¹⁾, Rögner⁽³³⁾, and Scheuermeyer & Steigerwald⁽³⁵⁾.

The analysis of the n-heptane fuel is shown in Appendix L.

8.1. Preparation of n-heptane air mixtures.

The mixture was prepared in a stainless steel vessel, having an effective internal volume of 483.8 ml., utilising the partial pressure technique. The general arrangement of the apparatus is shown in Fig. 8.1a. A known volume of liquid fuel is injected through a $\frac{1}{16}$ in. diameter hole in the vessel 'A', at atmospheric pressure, with the aid of a hypodermic syringe, after which the hole is sealed. Air from a compressed air cylinder 'B', dried by passage through a tube containing cupric chloride, is fed into the vessel. The pressure in the vessel is measured by a mercury manometer, and the temperature by a thermometer in the thermostatically controlled water bath surrounding the vessel. The passage between the vessel and the manometer consists of a 1 mm. bore capillary tubing 40 cm. long; since the passage is narrow, very little fuel vapour is expected to diffuse out of the vessel during the short period needed to make the pressure measurement.

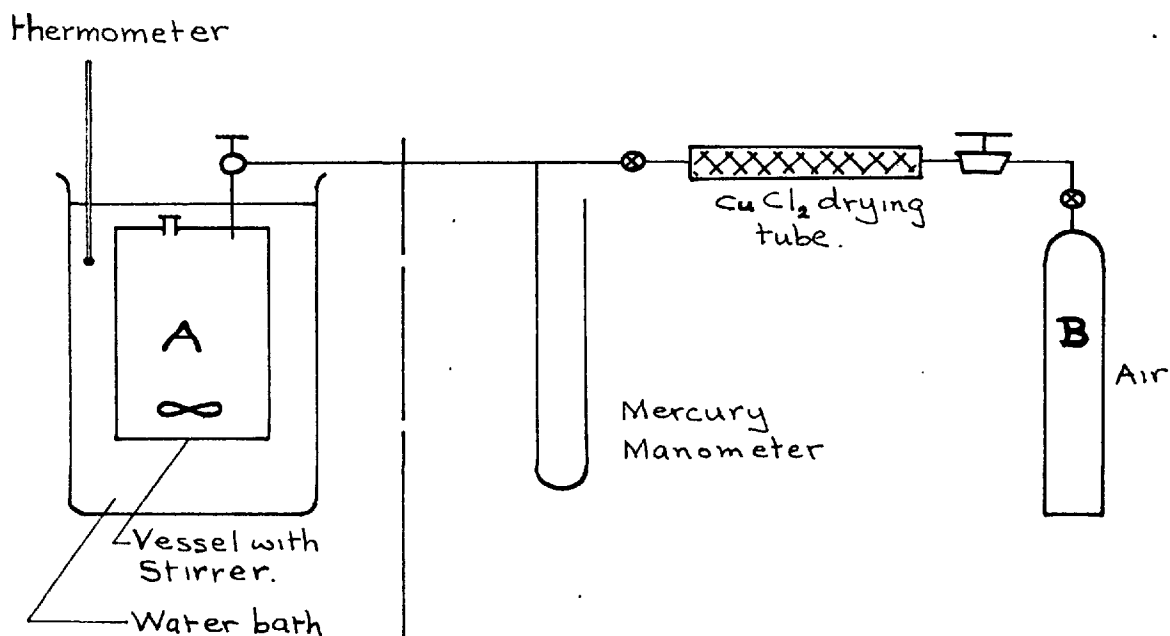


Fig. 8.1a.

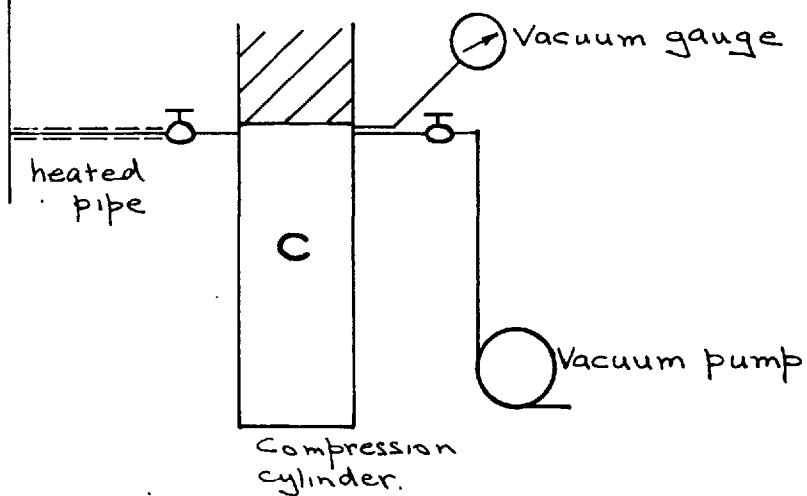
a). Preparation of vapour mixtures.

Fig. 8.1b.

b). Filling the machine with the prepared vapour mixture.

The fuel/air ratio by weight is :

$$\frac{w_f}{w_a} = 3.45 \frac{p_f}{p - p_f} = 0.0667 \lambda$$

where p_f , the partial pressure of the fuel, is found by :

$$p_f = 0.88 V_{inj} T \quad \text{mm.Hg.}$$

V_{inj} is the volume of fuel liquid injected in ml., T is the temperature of the water bath in $^{\circ}\text{K}$, and p is the total absolute pressure in mm. of Hg. in the container. (See Appendix M). For a stoichiometric mixture, $\lambda = 1.0$.

Figure 8.1b. shows the arrangement used for filling the compression cylinder. Before vessel A was connected to the apparatus, it was shaken by hand to allow the stirrer inside the vessel to mix the gases thoroughly. After the compression cylinder had been evacuated, the vapour mixture was allowed to flow into the cylinder until the required pressure had been attained. To avoid condensation of the fuel, the copper filling pipes were electrically heated to a temperature a few degrees above that of the vapour. Although the vapour mixtures were prepared at the same temperature as that of the compression cylinder, at least 10 minutes were allowed to elapse after filling before a compression ignition experiment was carried out. After each experiment, the walls of the combustion space were cleaned by wiping with dry cotton wool, so that residuals of combustion products did not affect the following run.

8.2. Apparent adiabatic exponents.

Since the specific heats of n-heptane are greater than those of air, the adiabatic exponent γ should decrease with increased mixture strengths. In a series of experiments with compression ratios of chiefly 8.66:1 and 9.60:1, this was found to be so. In addition, it was noted that for the same mixture strength, the lower the gas pressure before compression, the lower the apparent γ value. The probable explanation for this observation is that with a smaller mass of gas in the cylinder, the same heat loss to the walls represents a larger percentage of the total available heat. Hence the adiabatic efficiency is decreased and, as a result, the apparent γ value is decreased.

The values for γ given in table I were calculated from the measured pressures; those for initial gas pressures between 14.5 and 15.0 psia are shown graphically in fig. 8.2. Quantitative measurement only commenced after Run No. 160, and the technique of pressure measurement had not progressed to the stage of reliability until Run No. 200. It is for this reason that there is a considerable scatter of the values of γ obtained from runs prior to No. 200. However since the operating conditions of the machine were essentially constant during the runs, the actual temperature and pressure conditions at the end of the stroke should not have changed. Therefore in order to analyse the ignition delay data obtained from the earlier runs, the corresponding pressures and temperatures after compression were calculated from the values of γ given in

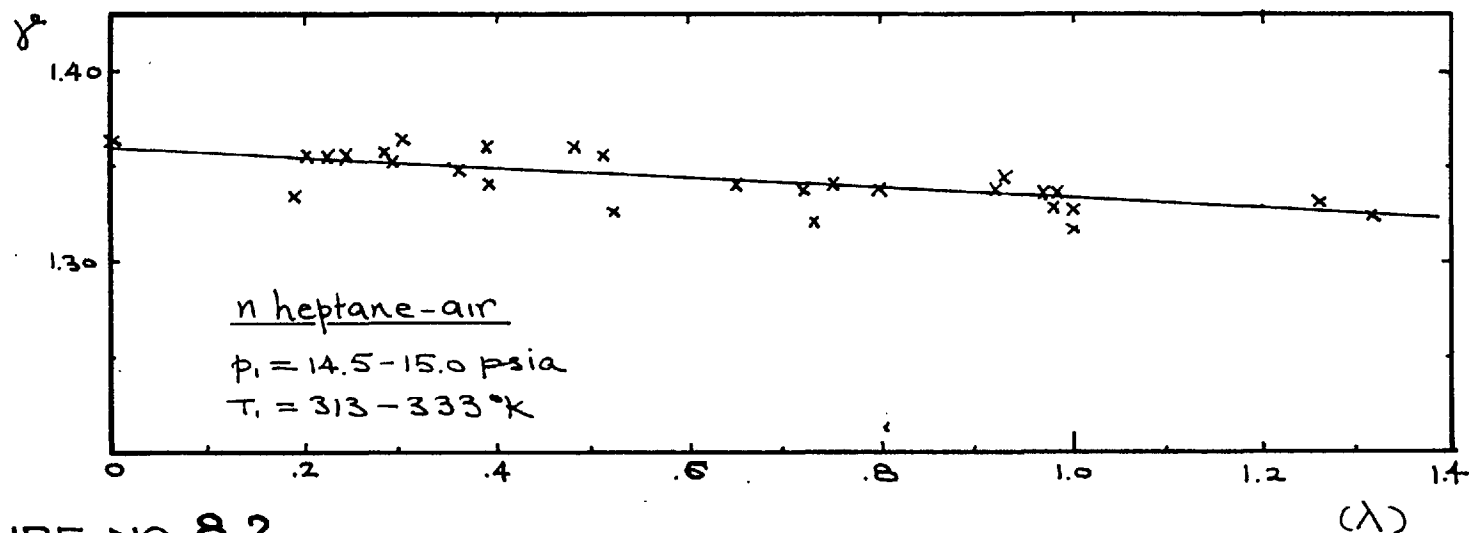


FIGURE NO. 8.2.

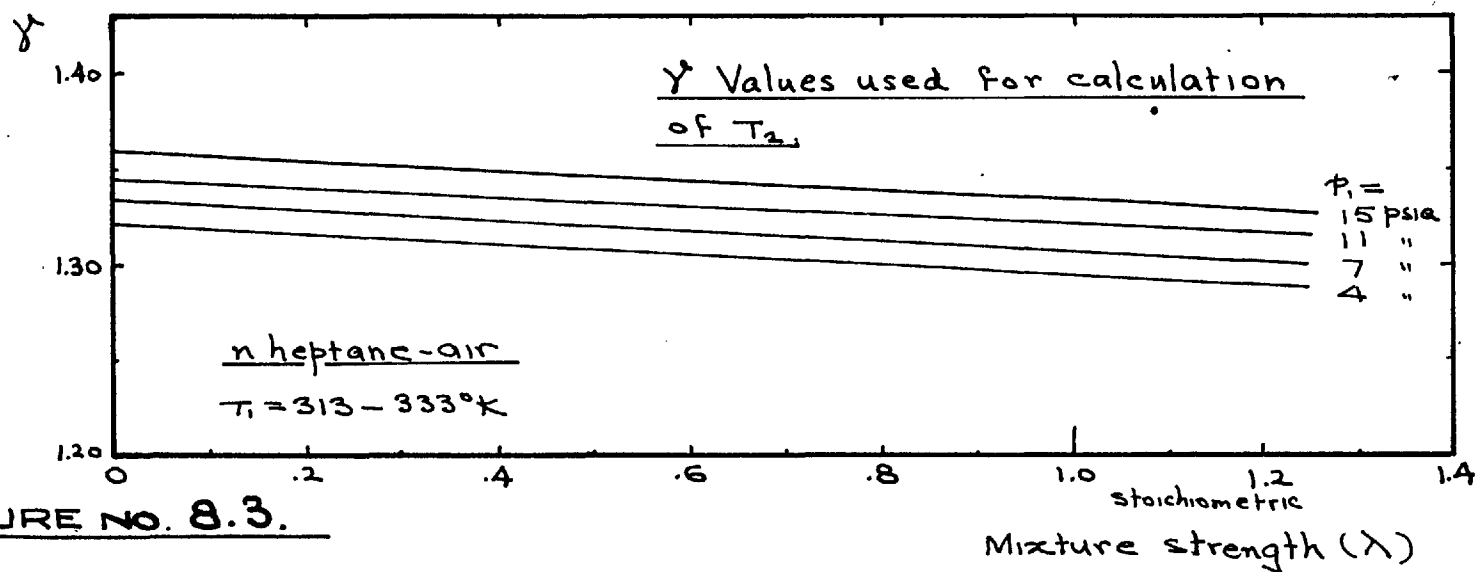


FIGURE NO. 8.3.

fig. 8.3, which was derived by the least squares method from graphs of values plotted from 66 later and more accurate runs at various initial pressures p_1 .

8.3. The two stage ignition phenomenon and ignition delay measurements.

The phenomenon of the two stage ignition is known to occur when a n-heptane-air mixture is subject to a condition which is above the critical pressure and temperature values for ignition to occur. ⁽⁴⁾⁽¹⁴⁾⁽³⁹⁾
(31)(24)

In the case when the mixture is adiabatically compressed to the pressure and temperature condition at which ignition may occur, the phenomenon is usually accompanied by two successive ignition delay periods τ_1 , and τ_2 , as illustrated in fig. 8.4.

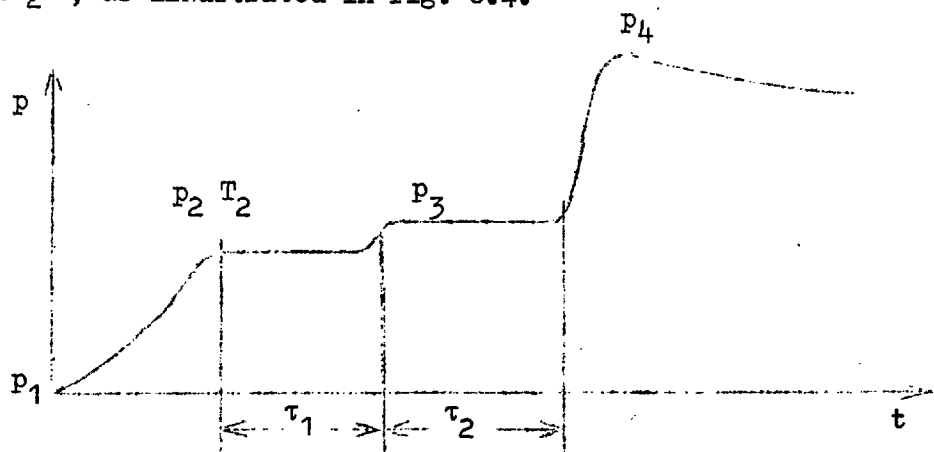


Fig. 8.4.

The first period τ_1 , is measured from the time compression is completed until such time that the derivative of pressure with respect to time, $\frac{dp}{dt}$, is a maximum. Little energy is released during τ_1 , except near the end of the period when cool flames occur ⁽²³⁾ ⁽²⁴⁾ and between 10% to 27% of the total heat of combustion is released ⁽⁴²⁾.

Intermediate products accompanying the cool flames in the first stage ignition have been found to consist mainly of peroxides and aldehydes (4) (6). The second period τ_2 is the difference in time between the end of the first period and the time at which the second derivative of pressure with respect to time, $\frac{d^2p}{dt^2}$, has a maximum positive value. Immediately following this period, the main explosion occurs and the remaining heat of combustion is released.

The two stage ignition process is generally thought to be a phenomenon of the process of building up the concentration of chain carriers necessary for the reaction; at the first stage some relatively stable products are formed which eventually become oxidised in the second stage.

The experiments carried out included the following:

- a. The effect of mixture strength λ on the ignition delay periods.
During these experiments, the compression ratio, the initial pressure p_1 and temperature T_1 were unchanged.
- b. The effect of final compression pressure p_2 on the ignition delay for a constant mixture strength, compression ratio and initial temperature T_1 . since the seals on the compression piston enabled subatmospheric pressures to be contained in the compression cylinder before the commencement of the stroke, different values of p_2 could be achieved by readjusting the initial pressure p_1 . Whilst the value of T_2 remained more or less constant.

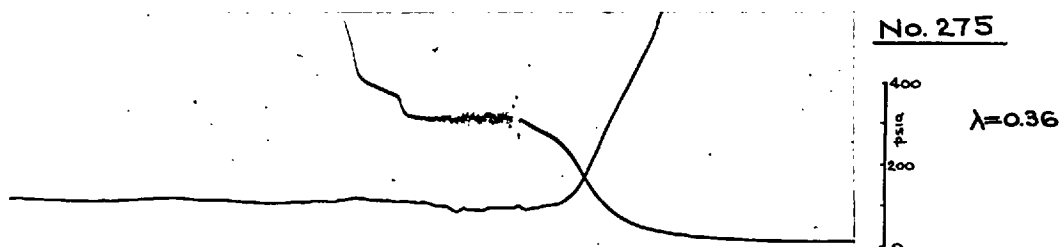
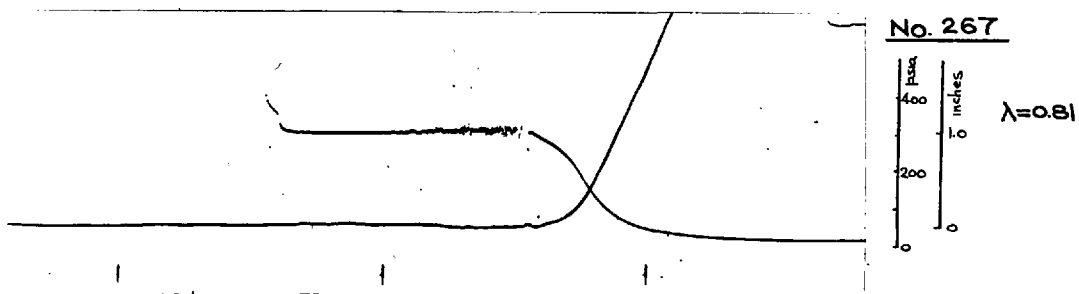
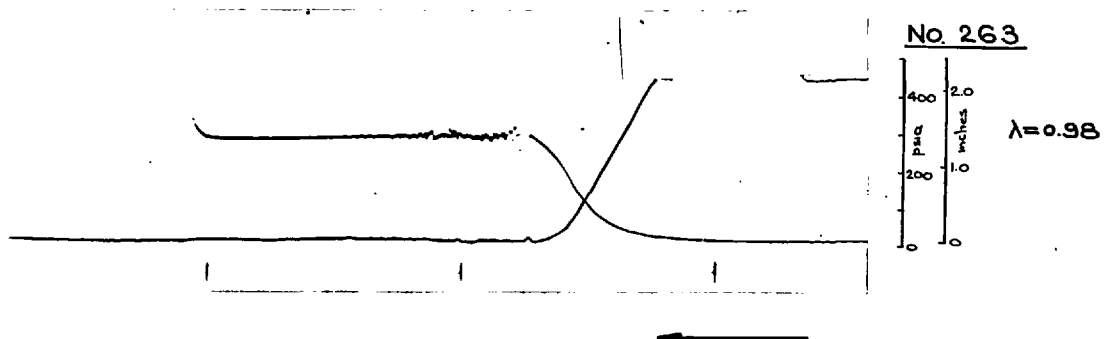
Table I : Ignition delay periods of n-heptane/air mixtures.

Run No.	C.R.	λ	P ₁ psia.	T ₁ °K.	P ₂ psia	γ (Apparent)	T ₂ °K.	τ_1 msec.	τ_2 msec.	$\tau_{tot.}$ msec.
167	8.66	.20	14.7	313	273	1.354	673	8.2	18.1	26.3
164	"	.24	14.8	"	276	1.356	672	9.3	9.0	18.3
166	"	.30	14.8	"	282	1.364	670	7.8	5.8	13.6
169	"	.39	14.7	"	276	1.360	667	9.9	1.7	11.6
170	"	.48	14.7	"	276	1.340	667	11.6	1.8	13.4
165	"	.75	14.8	"	267	1.340	655	25.3	1.2	26.5
173	"	.97	14.7	"	263	1.337	648	44.2	-	44.2
168	"	.98	14.7	"	258	1.330	648	42.0	-	42.0
190	8.66	.19	14.8	325	262	1.332	698	3.1	16.1	19.2
191	"	.29	14.5	"	268	1.353	696	3.8	3.9	7.7
192	"	.39	14.5	"	260	1.340	692	5.4	1.8	7.2
193	"	.52	14.5	"	253	1.325	688	7.8	1.0	8.8
194	"	.73	14.5	"	250	1.320	680	12.5	0.5	13.0
195	"	.93	14.5	"	264	1.345	673	17.5	0.5	18.0
196	"	1.26	14.7	"	260	1.331	661	30.2	-	30.2
220	9.60	.22	14.5	313	309	1.352	697	2.4	6.9	9.3
221	"	"	11.6	"	243	1.350	676	2.4	10.5	12.9
222	"	"	7.55	"	162	1.355	657	4.0	31.0	35.0
223	"	"	5.22	"	106	1.332	645	5.2	-	5.2
186	9.60	.28	14.8	313	319	1.357	695	4.2	3.6	7.8
275	9.60	.36	14.7	313	309	1.348	692	4.5	1.7	6.2
276	"	"	11.6	"	241	1.340	574	5.3	2.3	7.6
277	"	"	7.8	"	155	1.320	653	6.4	5.5	11.9
278	"	"	4.62	"	90	1.311	635	10.8	37.0	47.8

Table I - contd.

Run No.	C.R.	λ	p_1	T_1	p_2	γ	T_2	τ_1	τ_2	τ tot.
242	9.60	.51	14.7	313	315	1.354	689	5.7	0.8	6.5
243	"	"	11.6	"	230	1.322	667	6.0	1.1	7.1
244	"	"	7.27	"	141	1.310	646	7.6	2.7	10.3
245	"	"	4.39	"	84	1.302	632	10.9	10.5	21.4
271	9.60	.65	14.5	313	300	1.340	684	6.1	0.7	6.8
272	"	"	11.6	"	237	1.338	667	7.6	0.9	8.5
273	"	"	7.42	"	145	1.312	639	10.3	2.0	12.3
274	"	"	4.38	"	85	1.308	621	16.1	9.3	25.4
216	9.60	.72	14.8	313	304	1.337	680	8.5	0.5	9.0
267	9.60	.80	14.7	313	317	1.337	676	9.4	0.5	9.9
268	"	"	11.6	"	230	1.321	654	11.6	0.8	12.4
269	"	"	7.41	"	149	1.328	632	16.9	1.7	18.6
270	"	"	4.39	"	84	1.305	619	44.7	-	44.7
224	"	0.92	14.7	313	303	1.338	672	12.9	-	12.9
263	9.60	0.98	14.7	313	299	1.332	670	13.0	-	13.0
264	"	"	11.6	"	230	1.320	651	19.0	-	19.0
265	"	"	7.62	"	151	1.320	627	39.6	-	39.6
266	"	"	5.43	"	104	1.305	616	>50	-	>50
255	9.60	1.00	14.5	324	285	1.318	690	9.1	0.4	9.5
257	"	"	11.6	"	224	1.312	674	10.2	0.6	10.8
259	"	"	7.8	"	146	1.294	655	12.7	1.1	13.8
261	"	"	4.92	"	95	1.308	639	15.9	4.7	20.6
260	"	"	14.0	"	274	1.317	690	7.9	0.4	8.3*
256	"	"	11.6	"	217	1.295	674	8.2	0.5	8.7*
262	"	"	6.15	"	115	1.295	641	11.3	2.5	13.8*
258	"	"	4.92	"	89	1.280	639	14.6	3.5	18.1*
384	11.0	1.00	12.8	333	308	1.325	727	1.0	0.5	1.5

*Combustion chamber not cleaned.



IGNITION OF n-HEPTANE AIR MIXTURES AT
VARIOUS MIXTURE STRENGTHS.

C.R. = 9.60 $T_i = 313^\circ\text{K}$ $P_i = 14.7$ psia

Timing marks at 10 msec intervals.

$\lambda = 1$ for stoich^c mixture

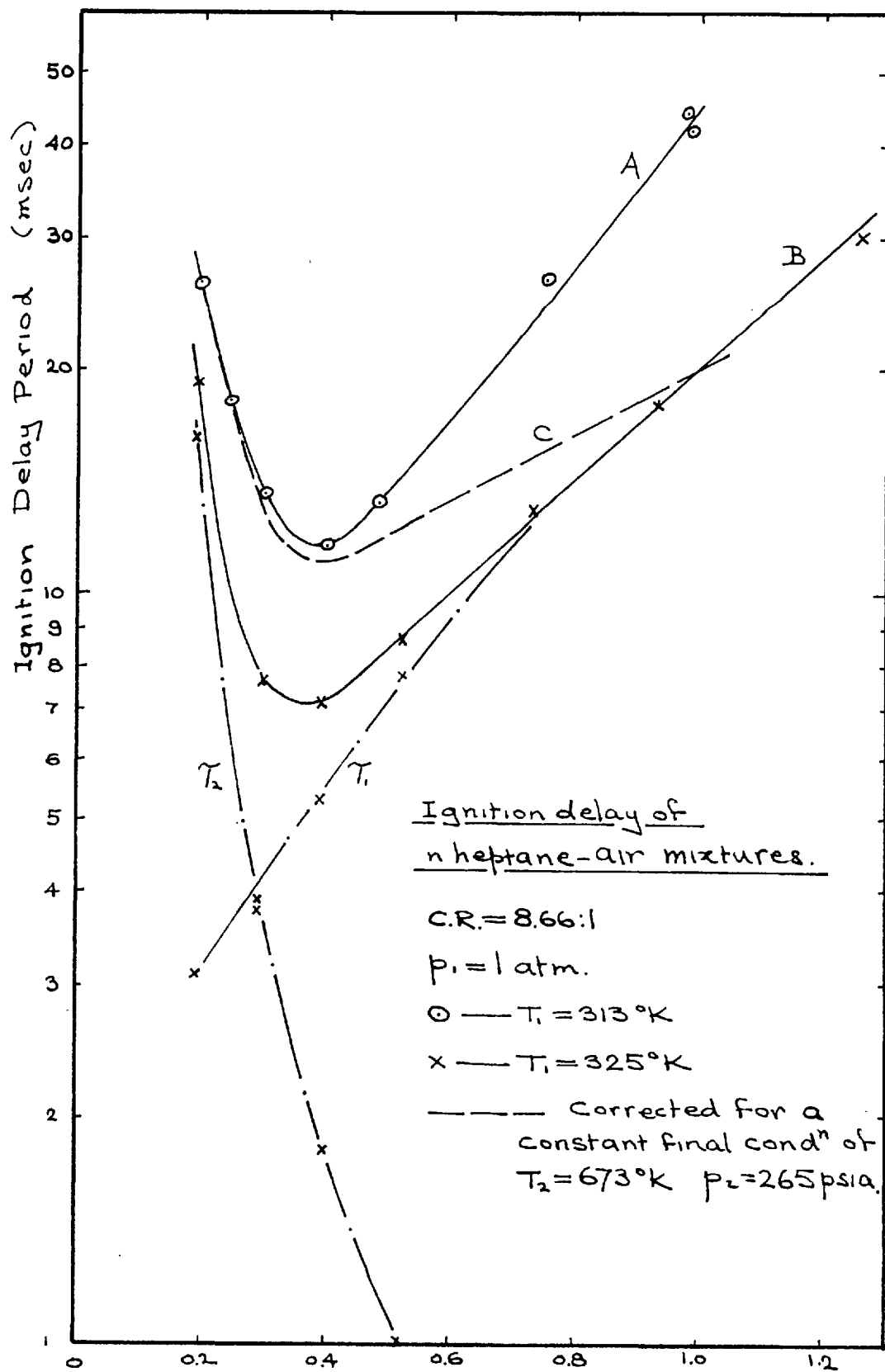


FIGURE NO. 8.5.

Mixture Strength (λ)

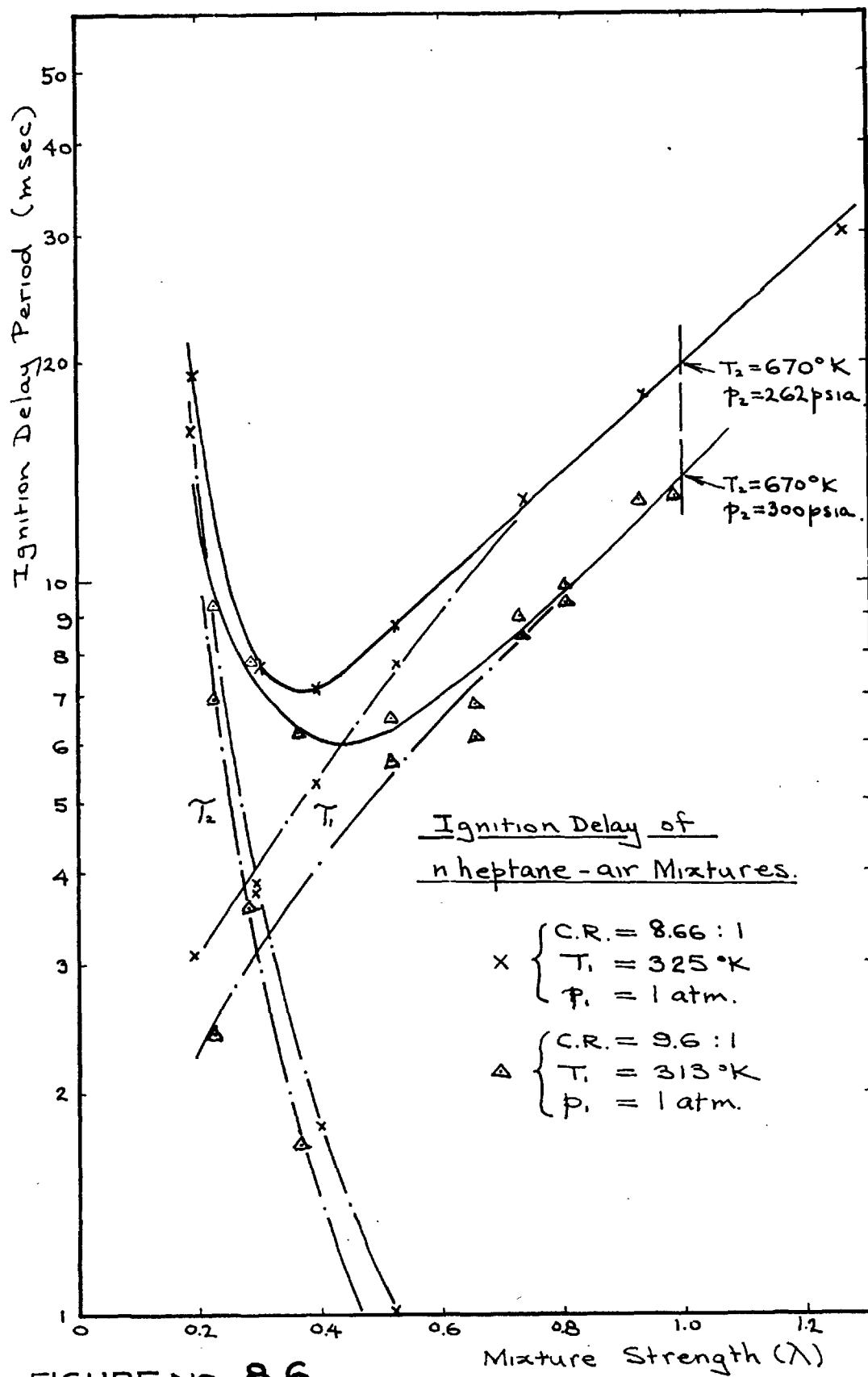


FIGURE NO. 8.6.

The ignition delays, plotted in figures 8.5 and 8.6. as a function of the mixture strength, were not measured necessarily under identical final compression conditions (p_2 and T_2), because adiabatic exponent γ also varies with mixture strength. Thus in Fig. 8.5. the experimental data, represented by the continuous lines A and B, has been interpolated with the aid of the following equation suggested by Rogener ⁽³³⁾ to give the ignition delay at constant p_2 and T_2 represented by the broken line C.

$$\tau = a p^{-n} e^{\frac{E}{RT}}$$

where τ is the ignition delay time

a is a constant (time)

p pressure after compression is completed

T temperature " " " "

E activation energy

R universal gas constant

The data confirm Martinengo's results that, for constant conditions of p_2 and T_2 , the first ignition τ_1 increases, and the second delay τ_2 decreases as mixture strength is increased. The sum $\tau_1 + \tau_2$ is a minimum when the mixture strength is approximately 0.4 x stoichiometric (i.e. when $\lambda = 0.4$). In Fig. 8.5, the difference between the curves A and B at any particular value of λ is mainly because of the difference in the T_2 values. To illustrate the effect of pressure on the ignition delay as a function of λ , the two series of results in Fig. 8.6. are plotted at equal temperatures T_2 but different pressures p_2 . The same effect is

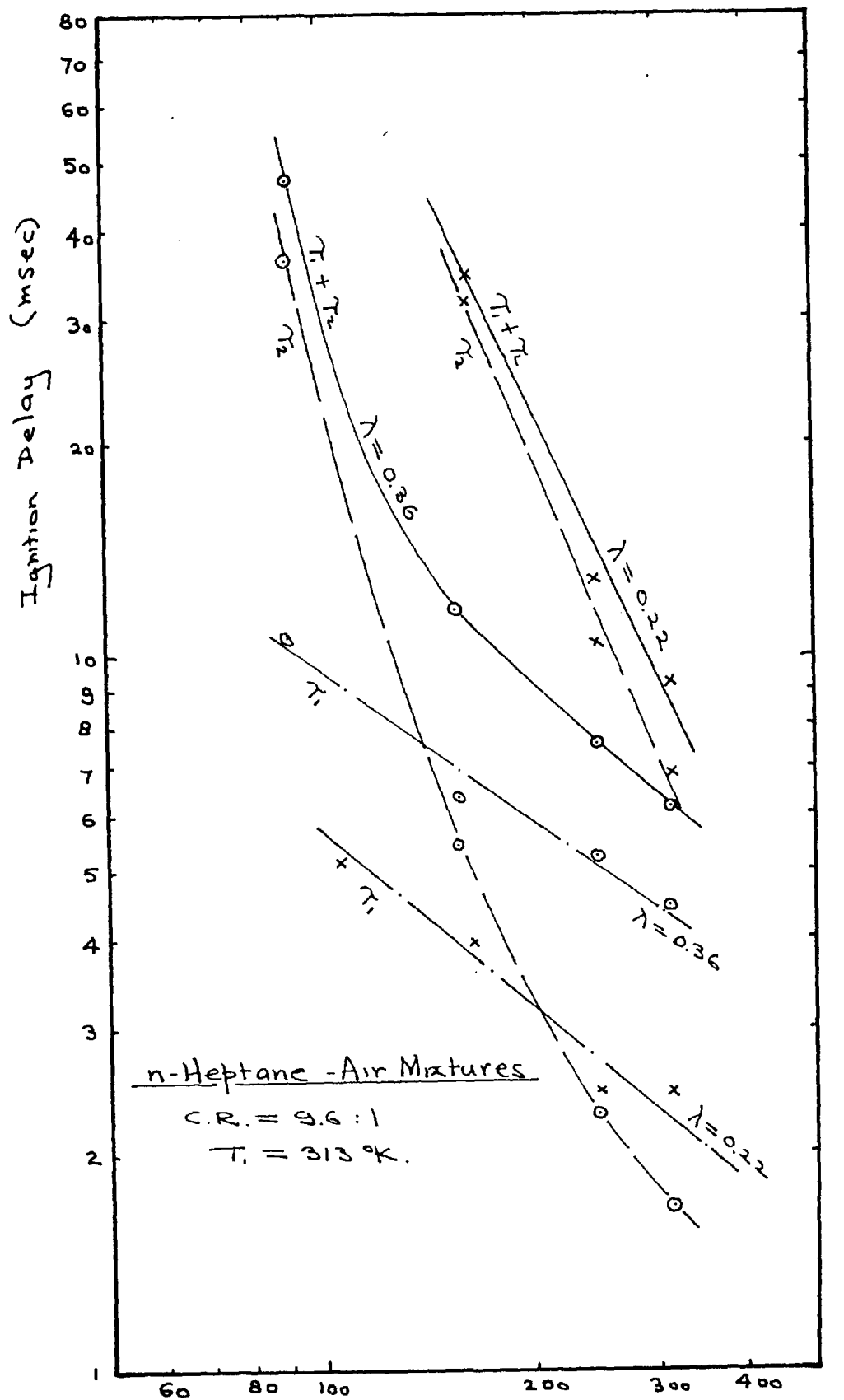
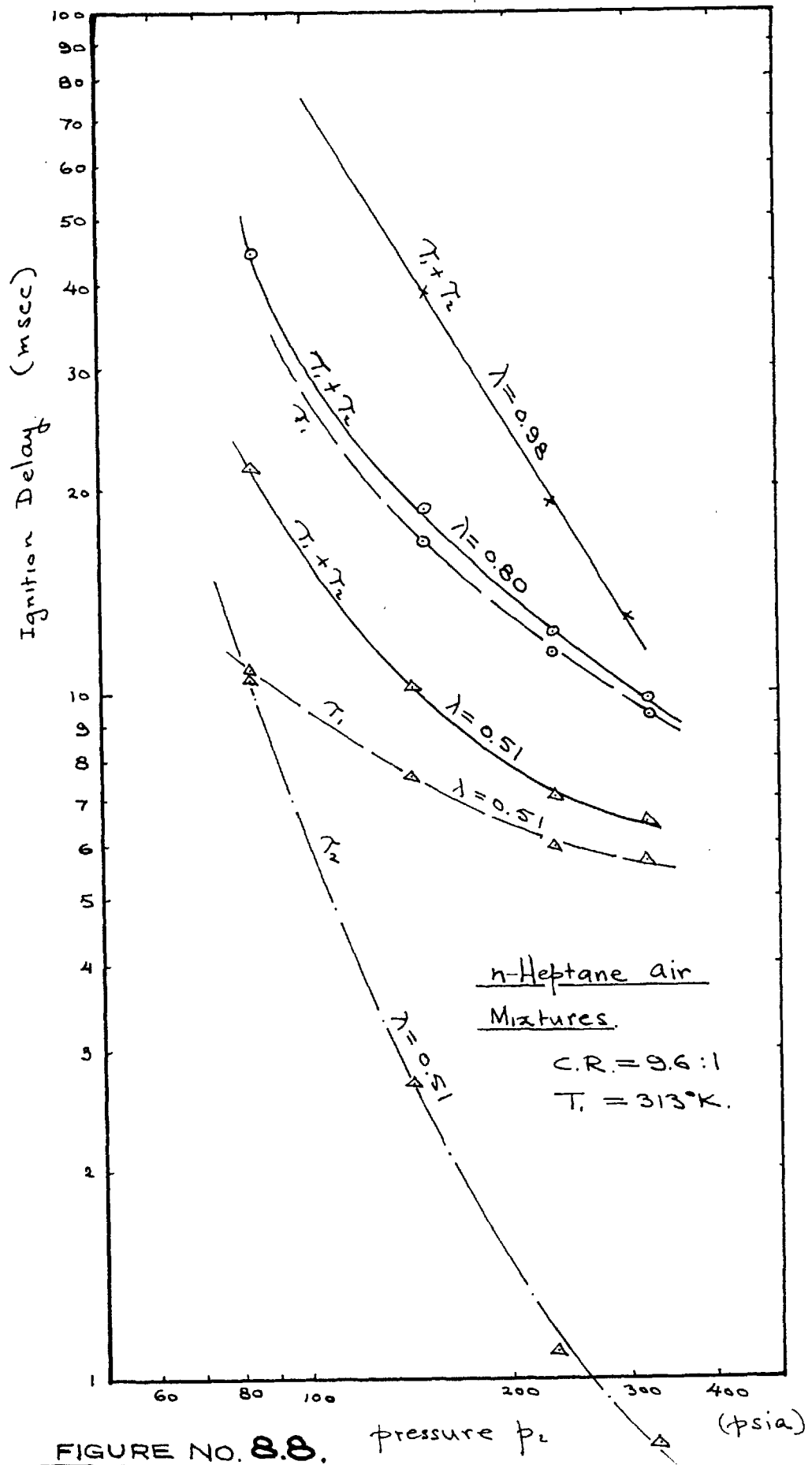
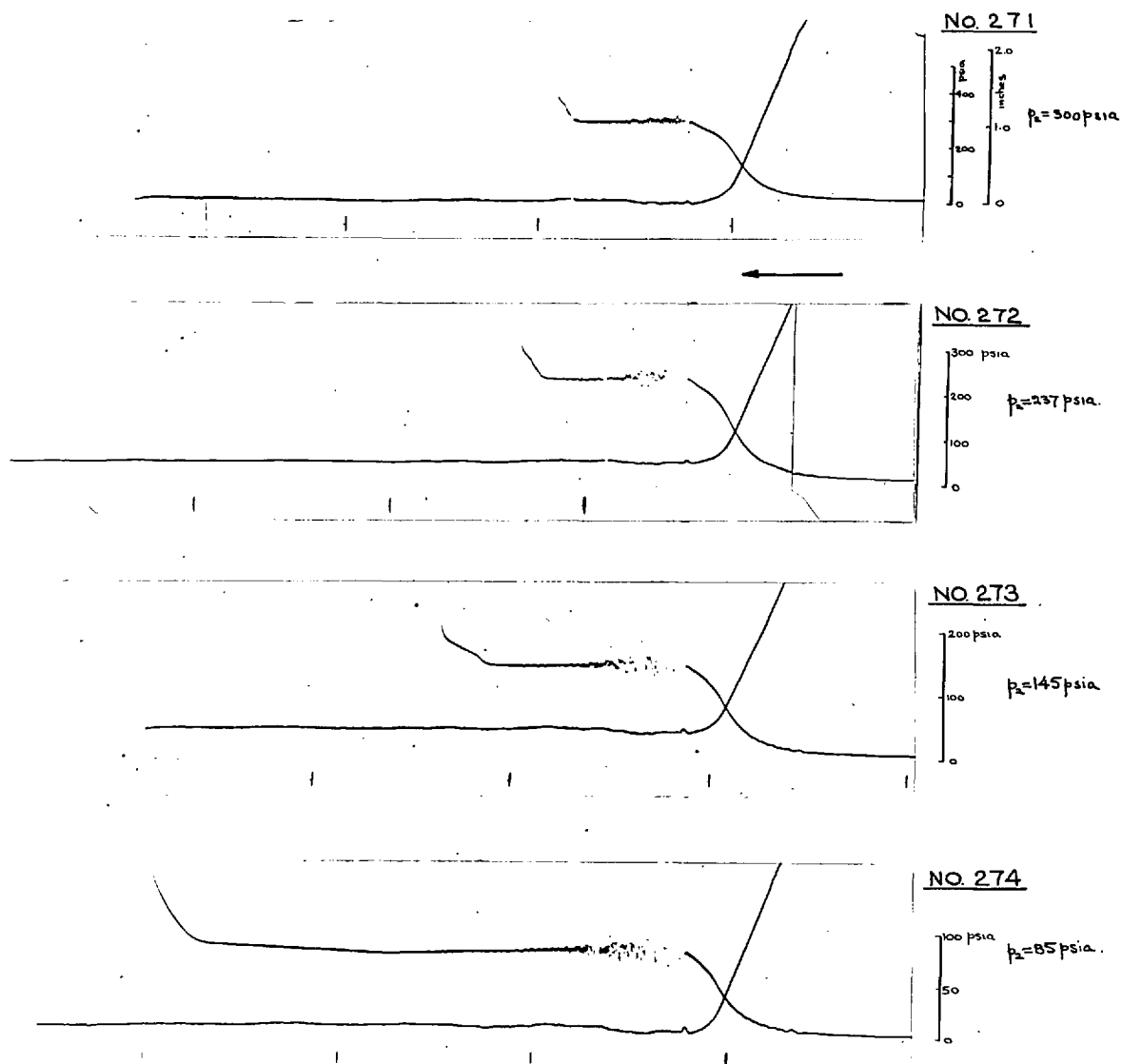


FIGURE NO. 8.7.

Pressure p_2

(psia)





IGNITION OF n-HEPTANE AIR MIXTURES AT VARIOUS
 COMPRESSED PRESSURES p_2

C.R. = 9.60 $T_i = 313^\circ \text{K}$ $\lambda = 0.65$

Timing marks at 10 msec intervals.

illustrated more clearly in Figs.8.7. and 8.8. where the delay periods are plotted directly against p_2 for fixed mixture strengths.

8.4. Discussion :

8.4.1. The effect of pressure and temperature on ignition delay.

The observed influence of mixture strength λ on the first and second ignition delay periods τ_1 and τ_2 when the mixture is compressed to constant final values of temperature T_2 and pressure p_2 , is in good agreement with that found by Martinengo⁽³¹⁾. Also, the effect of an increase in the final temperature T_2 causing a reduction in the delay periods, was found to be in accord with his results. As is shown in graph 8.6, the magnitude of p_2 has a significant effect on delay periods.

The results plotted in Fig.8.7. and 8.8. indicate that, when p_2 is reduced both delays τ_1 and τ_2 increase, but τ_2 increases at a faster rate. The probable explanation of this is that at lower pressures and concentrations, the frequency of collision between molecules is smaller, and hence longer ignition delay times result. At a constant temperature T_2 , the delay times τ are proportional to p_2^{-n} , where n is a constant. The observation is in quantitative agreement with Rogener's equation :

$$\tau = a p^{-n} e^{\frac{E}{RT}} .$$

After correcting the graphs in Figs. 8.7. and 8.8. to a constant temperature $T_2 = 680^\circ\text{K}$, the values of n given in Table II were determined for the two ignition delay periods as a function of mixture strength.

Table II. : Values of n derived for the relation $\tau \propto p^{-n}$.

λ	- n	
	τ_1	τ_2
.218	0.73	2.18
.357	0.57	1.52
.511	0.25	1.41
.805	0.56	-
.979	1.32	-

From the logarithmic plot of τ_1 and τ_2 versus pressure p_2 , given in Fig.8.7. and 8.8., it may be seen that deviations from linearity are more pronounced at lower pressures when the delay periods usually exceed 20 msec. In this range of delay times, it is possible that loss of heat through the cylinder walls may account for the departure from linearity.

From most of the results, the two-stage ignition phenomenon was found to occur in weak mixtures. However, in Table I, results of Runs 255, 257, 259, 261 and 263-266 indicate that two-stage ignition may occur with mixtures richer than stoichiometric when combinations of higher temperatures T_2 and lower pressures p_2 exist. For this reason, and also because τ_2 increases faster than τ_1 when p_2 is decreased for the same mixture strength, the minimum total delay time $\tau_1 + \tau_2$ may not always occur at a mixture strength of $\lambda = 0.4$, as it does in Figs.8.5. and 8.6. Indeed, the value of λ at which $\tau_1 + \tau_2$ is minimum, is likely to be dependent on temperature and pressure.

Hsieh and Townend⁽¹⁴⁾ found that increased pressure widens both the upper and lower mixture concentration limits for ignition. Runs 220 to 223, tabulated in Table III, were concerned with the compression of very weak mixtures ($\lambda = 0.22$) to see if reducing the pressure p_2 enabled the limit of inflammability to be exceeded. In Table III, p_2 is the pressure reached after compression; p_3 is the pressure resulting from the increase of temperature accompanying the cool flame; and p_4 is the maximum pressure which occurred during the second stage of ignition.

The ratios p_3/p_2 and p_4/p_3 indicate, to a first approximation, the amount of heat released by the cool flame and by the second stage ignition respectively.

Table III : Ignition of weak mixtures ($\lambda = 0.22$).

Run	p_2	p_3/p_2	p_4/p_3	T_2	τ_1	τ_2	τ tot.
220	psia. 309	1.22	1.40	^o K. 697	msec. 2.4	msec. 6.9	msec. 9.3
221	243	1.19	1.30	676	2.4	10.5	12.9
222	162	1.14	1.10	657	4.0	31.0	35.0
223	106	1.13	1.00	645	5.2	-	

It may be seen from Table III that, as p_2 is reduced towards a critical pressure, the second stage ignition ceased to occur. The temperature and pressure at which only the cool flame exists for a n-heptane/air mixture of $\lambda = 0.22$ are 645°K (372°C.) and 106 psia (7.2 atm.). McCormac and Townend⁽³⁰⁾ found that, for a mixture of strength $\lambda = 0.65$ at 372°C. , only

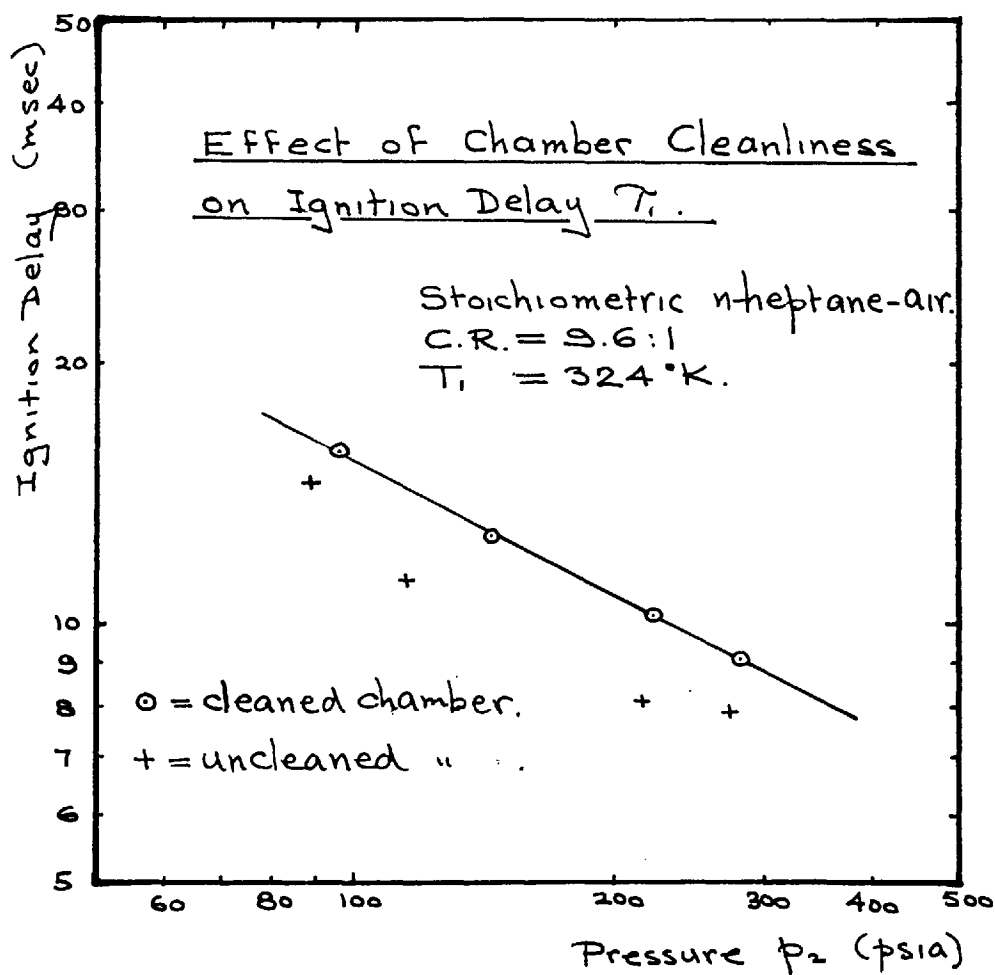


FIGURE NO. 8.9.

cool flames existed in the pressure range between 0.3 and 3.5 atm.

Because of the limitation of their instruments, they were not able to detect delay times much shorter than $\frac{1}{2}$ second. The results obtained in this investigation show that it is possible for cool flames to exist in weaker mixtures at higher pressures.

8.4.2. The effect chamber cleanliness.

When vapours of stoichiometric strengths were ignited, carbon and water were deposited on the cylinder head and walls. It was necessary to remove these products by cleaning the inside of the cylinder with dry cotton wool if reproducible results were to be obtained. The effect of the products was to shorten the ignition delay periods as shown in Fig. 8.9. In general, the reproducibility of ignition delay periods was found to be within $\pm 7.5\%$, provided that the walls were conditioned. For example, if after a series of runs with a different fuel, an experiment with an n-heptane/air mixture was repeated, the new results were found to differ from the previous ones by as much as 50%. Thus, it was essential to 'condition' the compression cylinder firstly by evacuating it for at least half an hour and followed by at least four or five compressions of the new fuel mixture.

8.4.3. Comparison of results.

In order to compare the ignition delays obtained in this investigation with those published by Taylor et al⁽³⁹⁾ using the M.I.T. compression machine, Martinengo⁽³¹⁾, Rogener⁽³³⁾ and Scheuermeyer & Steigerwald⁽³⁵⁾, the results for stoichiometric mixtures are plotted in Fig. 8.10. with

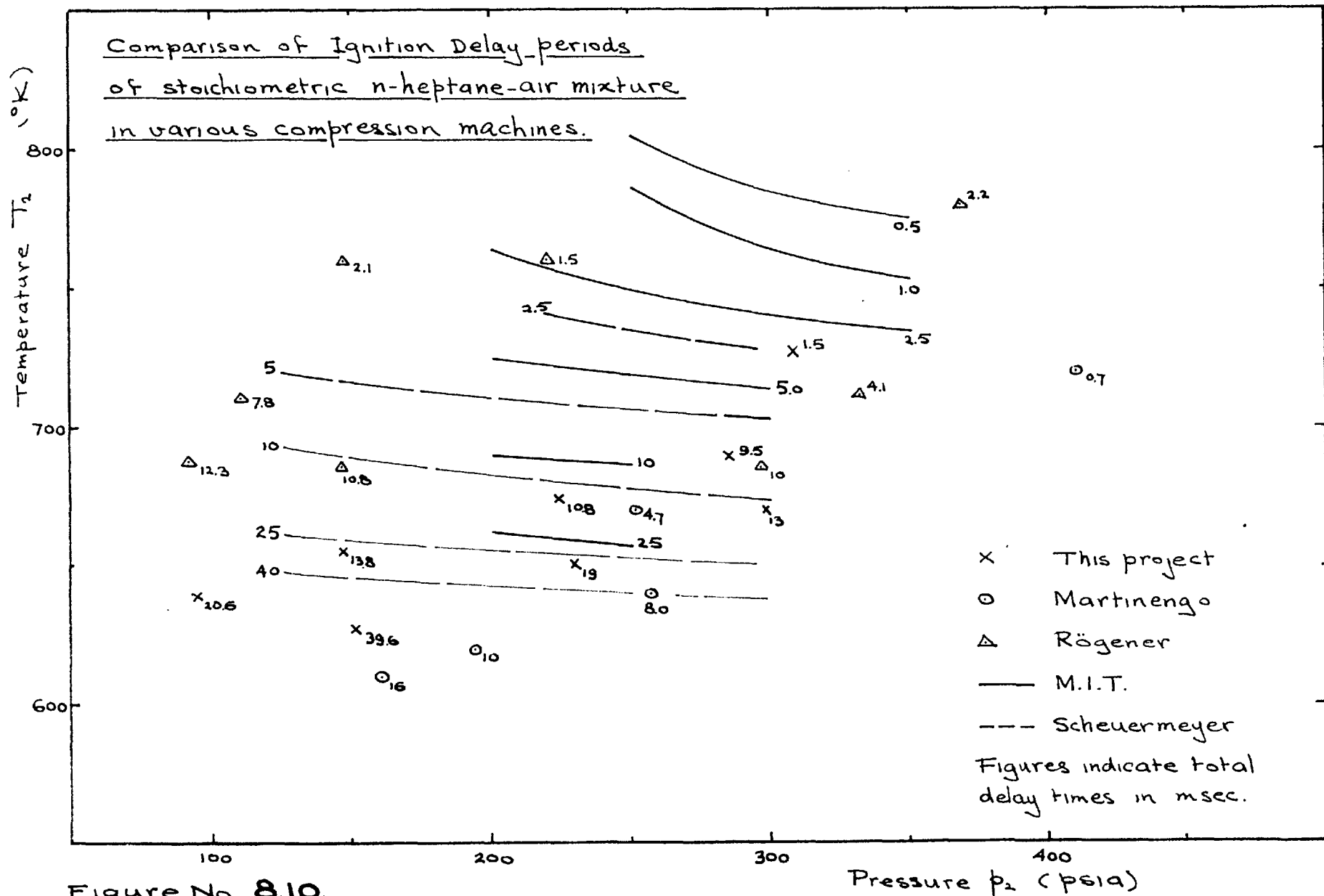


Figure No. 8.10.

temperature T_2 as ordinate and pressure p_2 as abscissa. The solid lines of isodelay contours are results from the M.I.T. machine published by Livingood⁽²⁷⁾, and the dotted contours are results obtained by Scheuermeyer & Steigerwald. For delay periods of the order of 10 msec., the present results agree well with those obtained at the M.I.T. and by Scheuermeyer and Rögner. However, for ignition delays exceeding 20 msec., the data obtained at the M.I.T. and those of Scheuermeyer are longer than the results of this project and those of Martinengo. In general, the M.I.T. machine appears to give longer delay periods than all others under the same conditions.

Table IV : Comparison of some of the characteristics of the various compression machines and the ignition delay periods for stoichiometric n-heptane/air mixtures obtained with the machines.

Machine	Piston dia. (in.)	Surface volume (in ⁻¹)	Min. ^m distance between walls (in.)	Compression time. (msec.)	Ignition Delay Periods (msec.)		
					$p_2=275$ psia $T_2=680^\circ\text{K}$	$p_2=250$ psia $T_2=740^\circ\text{K}$	$p_2=225$ psia $T_2=660^\circ\text{K}$
M.I.T.	2.0	7.33	0.34	6	12	3	25
This project.	1.5	5.72	0.59	11	10	1.5*	17
Martinengo	1.18	3.77	1.18	16*	48	1*	6*
Rögner	2.0	3.3*	1.6*	18*	9	2	15*
Scheuermeyer	3.15	2.10	2.41	19*	9	2	20

* Estimated Values.

A comparison of the characteristic of the various machines for the same compression ratio of 9:1 shows that the M.I.T. machine has the largest surface to volume ratio after compression; the shortest distance between walls; and the shortest compression time. All these factors may contribute to the longer ignition delay periods.

With a large surface to volume ratio the proportion of heat lost to the walls is greater, the energy available to sustain reaction is reduced, and consequently the delay period is lengthened. Although the heat loss depends on the rate of heat transfer, it is thought that the effect will be more pronounced as the actual delay period is increased. Except for the results of Scheuermeyer, those results tabulated in Table IV support this contention.

When the compression time is reduced, the subsequent ignition delay period should be increased because time is available for reaction during compression. This effect is illustrated in Table V by the experimental results of Scheuermeyer & Steigerwald and also by applying Livingood & Wu's⁽²⁸⁾ theory in predicting ignition delay times.

Table V : The effect of compression time on ignition delay for a stoichiometric n-heptane/air mixture.

1. Prediction according to Livingood and Wu (see Appendix <u>N</u>).				
C.R. = 9:1		$p_1 = 1 \text{ atm.}$		
	$T_2 = 680^\circ\text{K.}$		$T_2 = 750^\circ\text{K.}$	
Compression time (msec.)	6	18	6	18
Ignition delay (msec.)	11.75	11.50	3.76	3.30
2. Experimental Results by Scheuermeyer & Steigerwald.				
	$T_2 = 710^\circ\text{K.}$			
Maximum piston velocity (m/s)	52		16	
Ignition delay (msec.)	3.8		3.3	

An interesting fact to notice from Table V is that, when the ignition delay is longer (see values below $T_2 = 680^\circ\text{K}$), the duration of the delay is less affected by the compression time as compared with the same effect on the shorter ignition delay periods (see values below $T_2 = 750^\circ\text{K}$). Therefore, as is seen in Fig.8.10 and Table IV, for ignition delay periods longer than 10 msec., the differences between the delay periods collected from the various machines are greater than the differences which could be contributed to the effect of varying compression times only. For this reason, it is apparent that different compression times as well as different values of surface to volume ratios influence the ignition delay period.

If consideration is confined to the effect of compression time and surface to volume ratio, an ideal machine should give longer delay periods in the higher temperature and pressure region; and shorter delays in the lower temperature and pressure region when compared with the results of actual machines such as those collected in Fig.8.10.

Very little information about the surface conditions of the compression cylinders of the various machines is available. This factor may have had an important influence on the delay times, as it affects the reaction kinetics.

In conclusion, the delay times obtained with the machine described in this thesis are in good agreement with the mean of the widely scattered values obtained with other machines for the same conditions and fuel.

CHAPTER 9.

Compression and ignition of films of fuel in contact with air.

When a volume of air immediately above a film of liquid is suddenly compressed, the resultant increase in the air temperature may cause the fuel to ignite. To carry out a preliminary investigation of this phenomenon, the liquid film was confined in a recess in the cylinder head. The recess was in the shape of a ring, which was concentric with the pressure transducer. It had an O.D. of 1.03 in., I.D. of 0.60 in. and a depth of 0.01 in. The maximum capacity of the recess was 90.8 μ l, and when filled with 80 μ l of liquid, as was done in the experiments, the thickness of the film was 0.009 in.

9.1. Films of n-decane.

n-decane was chosen as the fuel because its vapour pressure is such that at temperatures below 28°C., the saturated vapour in the air above the film is less than 0.2 x stoichiometric, i.e. below the limit of ignition⁽³¹⁾. Thus, by varying the temperatures of the combustion cylinder between 15°C. and 60°C., it was possible to investigate the effect of rapid compression on a film with a non-ignitable vapour above it, as well as that of a film in contact with an ignitable vapour.

The liquid was introduced into the machine at least 15 minutes before compression to enable the liquid film to attain a steady temperature, and for the vapour to come into equilibrium with the liquid.

The temperature of the film was determined in a separate series of experiments under similar temperature conditions using a thermocouple.

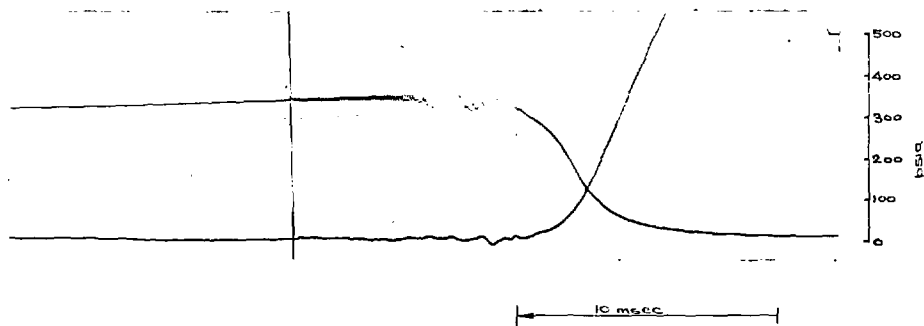
The results of the compression tests are tabulated in Table I, in which T_1 is the initial vapour and compression cylinder temperature; T_{1f} the film temperature; and T_2 the calculated adiabatic temperature of the vapour after compression. In all experiments, the initial pressures were atmospheric. To calculate T_2 , the saturated vapour mixture strength before compression was first estimated. Knowing the value of λ and compression ratio, the apparent value of γ was then determined from the graph in Fig. 9.1. based on information obtained from the compression of n-heptane/air mixture. The value of T_2 so estimated is thought to be accurate to within $\pm 10^\circ\text{C}$. The results of visual inspection of the products after compression were classified into three types; those experiments in which soot was formed, those in which a partial formation of soot was found and those in which no soot was found. The first classification means that the cylinder head was covered with soot completely; and the second, that a few particles of black soot were found in the remaining liquid. Because of the very slow rate of combustion in some experiments, the pressure increase was difficult to detect and it had to be determined by a comparison of the pressure-time records of the experiment with that of similar runs in which only air or nitrogen was compressed. In the column headed 'Pressure rise' in Table I, the symbols 'D.C.' and 'A.C.' are used to denote the cases where the pressure rose as a result of ignition during compression and after compression respectively. The case where no ignition or pressure rise occurred is denoted by the symbol 'N'.

Table I : Compression and ignition n-decane films.

Run No.	C.R.	Vapour temp. T_1 (°C.)	Film temp. T_{1f} (°C.)	Temp. after compression. T_2 (°K)	Soot Formation			Pressure Rise.	Type of Result.
					Yes	Partially	No		
373	30.3	17	17	888		✓		N.	A
305	"	18	18	891	✓			D.C.	D
300	"	31	31	894	✓			D.C.	D
375	"	32	32	897	✓			D.C.	D
372	23.8	17	17	840			✓	N	A
306	"	18	18	843	✓			D.C.	D
388	"	25.5	25.5	852	✓			D.C.	D
390	"	40	39.5	852	✓			D.C.	C
386	"	54	53	846	✓			D.C.	C
309	17.4	18.5	18.5	776			✓	N	A
310	"	19	19	777			✓	N	A
313	"	24	24	797			✓	D.C.	B
314	"	28	28	797			✓	D.C.	B
301	"	31.5	31	797		✓		D.C.	B
391	"	41	40.5	797	✓			D.C.	C
307	12.7	18	18	709			✓	N	A
312	"	23.5	23.5	723			✓	D.C.	B
316	"	28.5	28	725			✓	D.C.	B
302	"	31.5	31	727			✓	D.C.	B

Table I - contd.

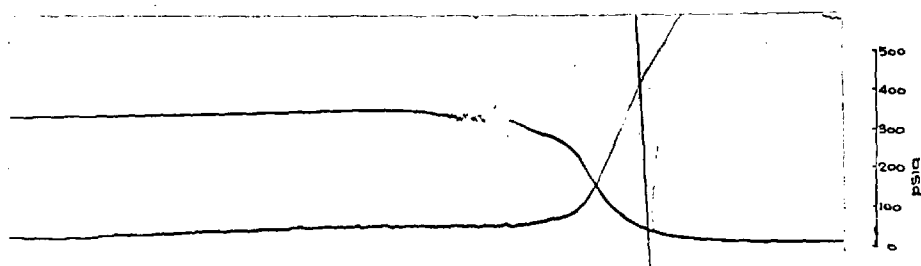
Run No.	C.R.	T ₁	T _{1f}	T ₂	Soot Formation			Pressure Rise.	Type of Result.
					Yes	Partially	No		
308	9.60	18	18	650			✓	N	A
389	"	24.5	24	661			✓	A.C.	B
315	"	28.5	28	666			✓	A.C.	B
311	"	30	29.5	670			✓	A.C.	B
303	"	40	38	668			✓	A.C.	B
304	"	40	38	668			✓	A.C.	B
394	"	45	43.5	666		✓		A.C.	C
387	"	54	51	672		✓		A.C.	C
392	9.60	29	-	668	(Vapour only, no film).			A.C.	
376	30.3	32	32	897 ²	(Film + N ₂ gas).				
						✓			

**RUN 311**

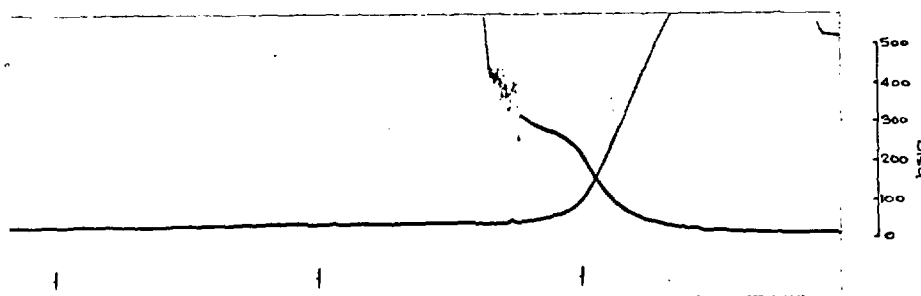
n decane film.

 $T_i = 30^\circ\text{C}$. C.R. = 9.6:1

No soot formed.

**RUN 392**Saturated ndecane
vapour only. $T_i = 30^\circ\text{C}$. C.R. = 9.6:1

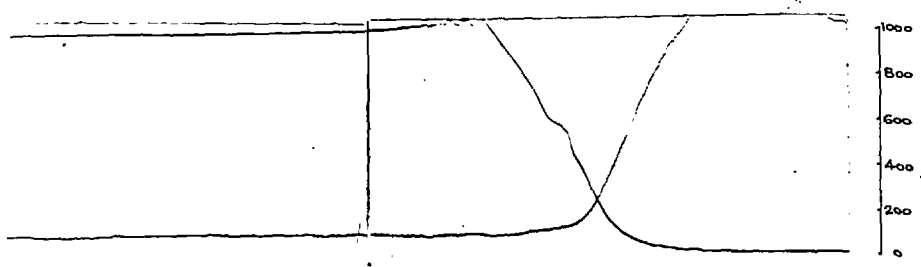
No soot formed.

**RUN 387**

n decane film.

 $T_i = 54^\circ\text{C}$. C.R. = 9.6:1

Some soot formed.

**RUN 391**

n decane film.

 $T_i = 41^\circ\text{C}$. C.R. = 17.4:1

Soot formed.

Ignition during
compression.

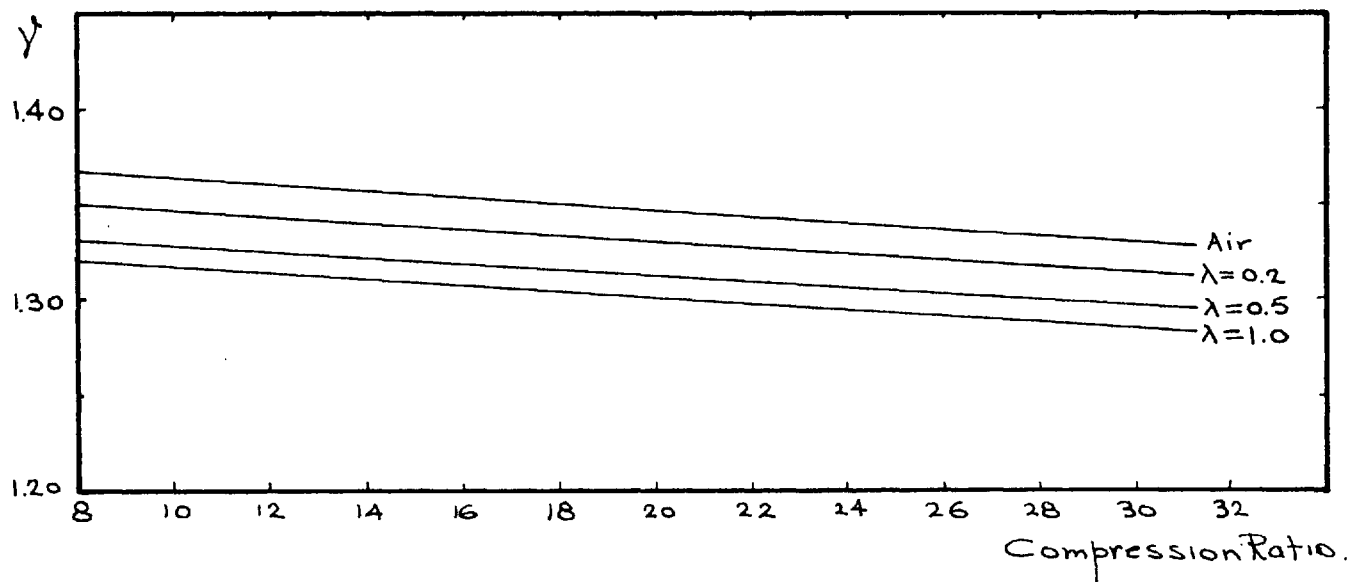


Fig 9.1. Values of γ at various Compression Ratios.

Experiments at various temperatures and compression ratios provided basically four different kinds of results, represented by the areas A, B, C and D in Fig.9.2. In area A, the final temperature T_2 of the vapour above the liquid was not high enough to cause ignition; no pressure rise was recorded, and the liquid remained unaffected. Results in Area B exhibit a small pressure rise of less than 15% of the compression pressure (see Run 311 on page 130) after delay of a few msec., the actual delay depending on the compression ratio used. The liquid remaining after compression was clear and no soot was detected. In Area C, the records showed sharp pressure rises either immediately after or during compression and a large amount of soot was found. Results in Area D showed pressure increases which were slow and difficult to detect, except for a gentle rise approximately 90 msec. after compression, as shown in Fig.9.4. In these cases, inspection of the cylinder head after the experiment revealed large quantities of wet soot.

To show whether the formation of soot, in the experiment represented by Area D, was due to combustion or cracking of the fuel, an experiment (Run 376) was carried out with nitrogen instead of air. This resulted in the deposition of a few very fine specks of carbon in the liquid, but the amount was much less than that produced in the corresponding Run 375 with air. This suggests that, although cracking of the fuel did take place to a limited extent, it was not the factor directly responsible for the formation of soot in Run 375. Fig.9.4. shows the pressure time records

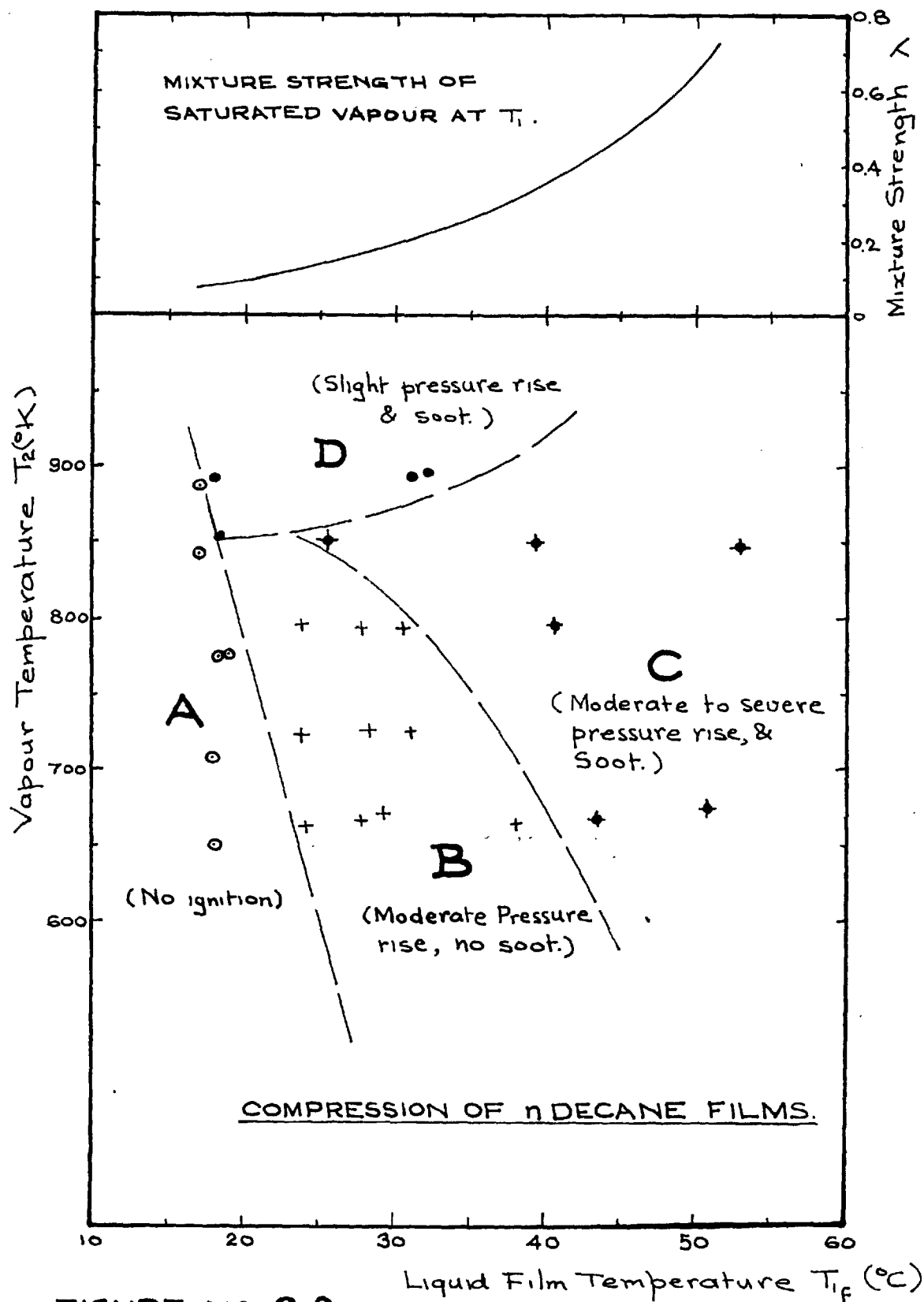


FIGURE NO. 9.2.

ADIABATIC COMPRESSION OF nDECANE
FILMS. $p_1 = 14.7$ psia.

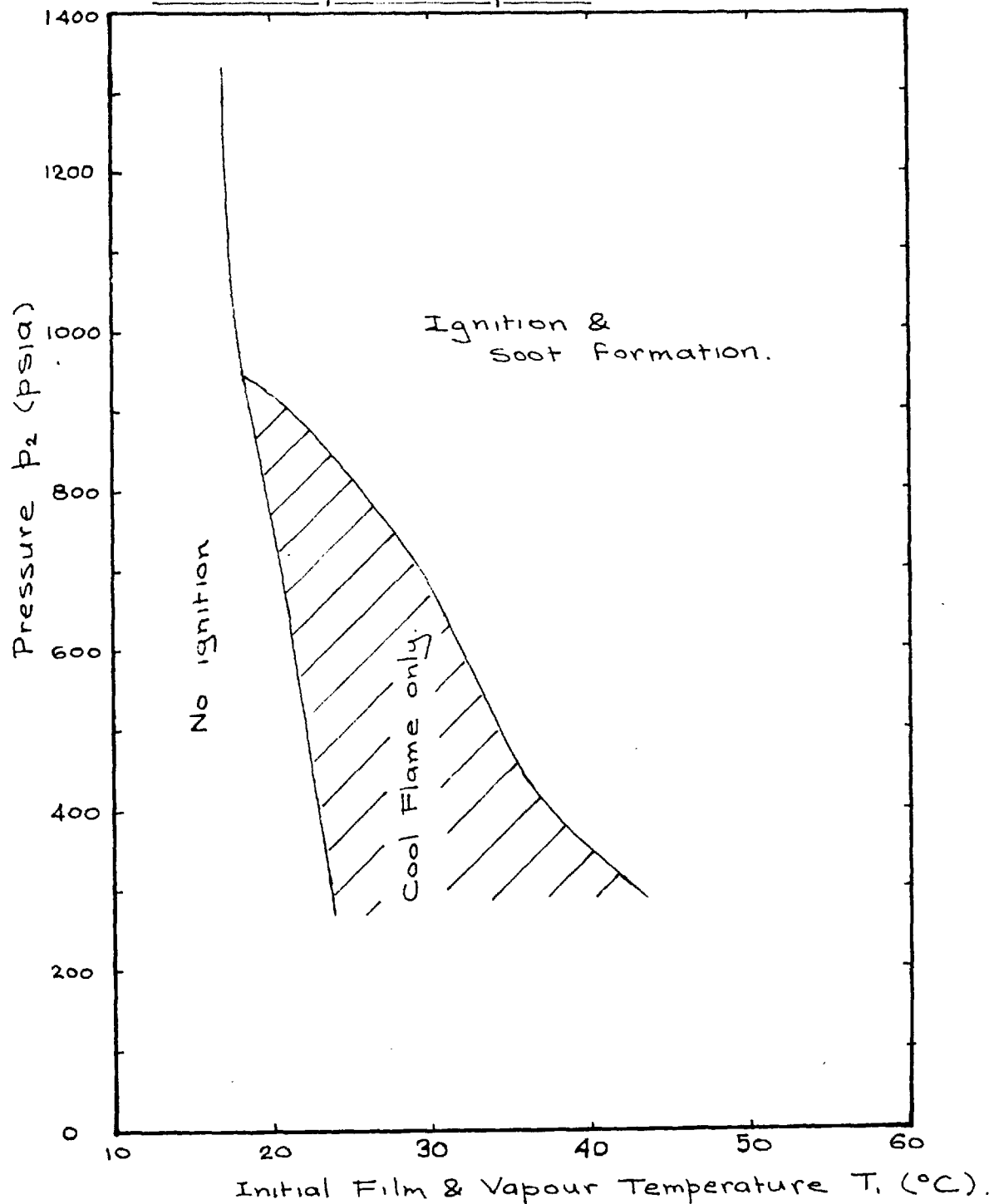


FIGURE NO. 9.3.

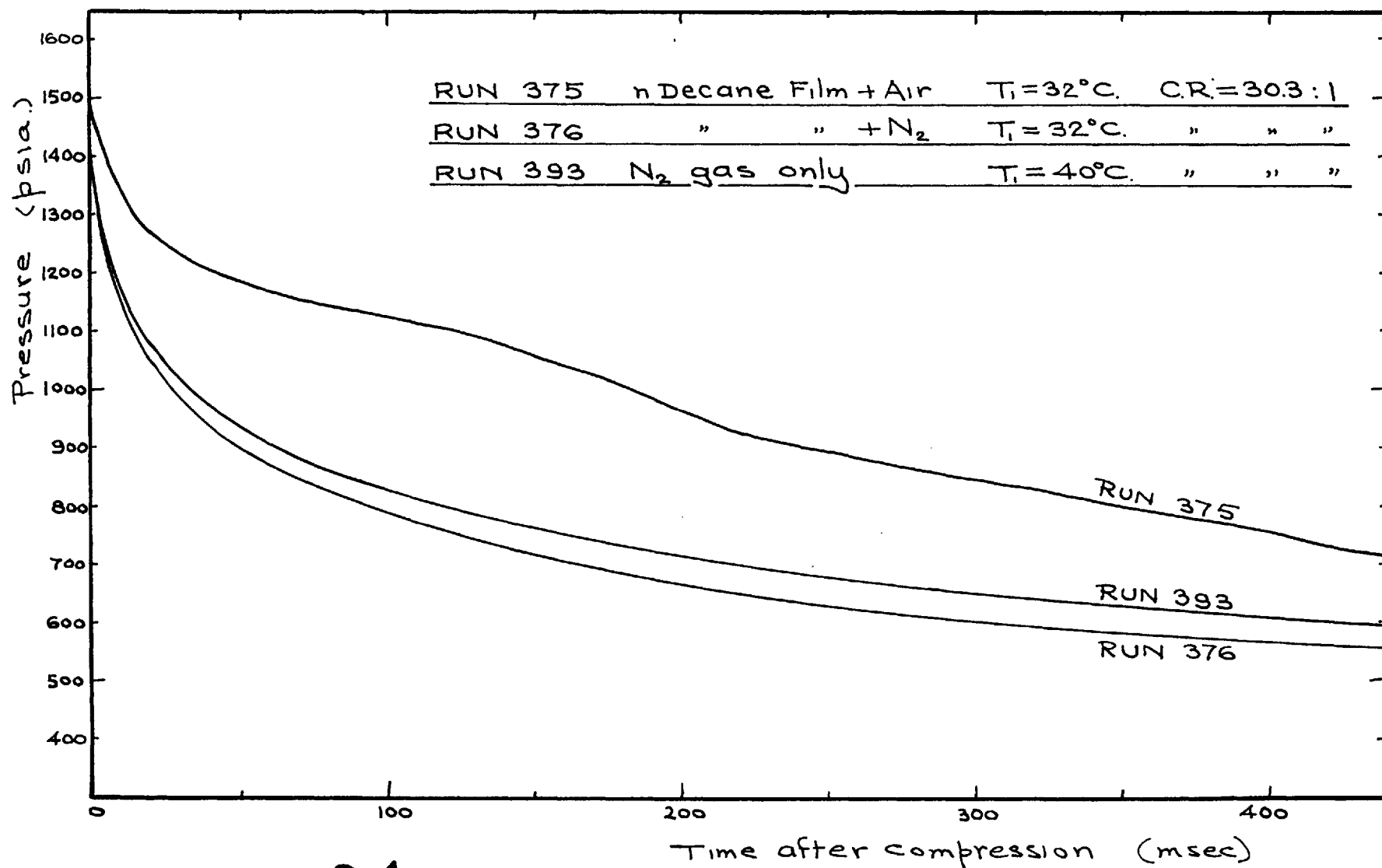


FIGURE NO. 9.4.

of Runs 375, 376 and Run 393 after compression. Run 393, in which nitrogen alone was compressed, served as a standard of comparison. Thus, the area between the records of Run 375 and 393 represent the heat released in Run 375.

9.2. Discussion.

An explanation of the four different types of behaviour shown in Fig.9.2. is as follows:

No ignition occurred in the Area A, because the initial temperature T_1 was not sufficiently high to provide a saturated vapour rich enough to ignite at the temperature T_2 at the end of the compression stroke. Further evaporation after compression was presumably insufficient to provide an ignitable mixture during the period that the temperature remained sufficiently high to cause ignition.

In Area B, the temperature T_1 was greater than 24°C. , and the saturated vapour was rich enough to cause a mild ignition. This was shown by Runs 311 and 392 on page 130. In Run 311, $80\ \mu\text{l}$ of n-decane was placed in the cylinder and compressed, whereas in Run 392, only $2.5\ \mu\text{l}$ was injected into the cylinder. The smaller quantity of liquid, when evaporated, should provide a saturated vapour of $\lambda = 0.17$ at a temperature $T_1 = 40^{\circ}\text{C.}$

Table II : Comparison of pressure rises in Runs 311 and 392.

Run	τ_1 msec.	p_1 psia.	p_2 psia.	$\frac{p_{\text{max.}}}{p_2}$
311	2.8	14.7	331	1.06
392	2.4	14.7	332	1.07

The results of the experiment are given in Table II, in which $p_{\text{max.}}$ is the peak value of explosion pressure. It may be seen from this table that both experiments gave almost identical pressure records. These results indicate that the pressure increase in Run 311 arises solely from the ignition of the saturated vapour above the liquid film. The type of pressure-time record obtained was very similar to that from Runs 222 and 223, in which very weak n-heptane/air mixtures near the ignition limit were compressed. (See Table III in Section 8.4.1). It was shown that in the case of n-heptane, combustion was confined to the formation of cool flame. The fact that the excess n-decane liquid in Run 311 was left uncontaminated with soot suggests that, as a result of its low temperature and luminosity, the cool flame could neither inflame the liquid, nor cause sufficient further evaporation to provide a rich burning mixture.

Results of type C appear when T_1 is greater than 38°C . and the saturated vapour enables mixture strengths greater than $\lambda = 0.33$ to be attained. In this zone, a second explosion definitely follows the cool flame stage with the result that faster and greater pressure rises were recorded, as seen from Runs 387 and 391 (page 130). Taylor⁽³⁹⁾ and Levedahl & Howard⁽²³⁾ found that the second explosion was accompanied usually by high intensity luminosity. This radiation, together with the high temperature, tended to evaporate more fuel and form a richer burning mixture, which explains the formation of the large quantity of soot.

Finally, the results in area D were produced with compression ratios greater than 23:1, so that pressures between 980 psia and 1350 psia, and temperatures between 840°K and 900°K were generated. The formation of soot arises to some extent from the cracking of the fuel, as was proved by Run 376, in which nitrogen was used instead of air. It may be seen from Fig.9.4. that, at the instant compression was completed, Run 375 reached a higher pressure than Run 393, in which only nitrogen was compressed. This indicates that some heat was liberated from combustion reactions occurring during the compression stroke. Ignition during compression is to be explained by two reasons. Firstly, it is shown in Section 8.3 and also by the work of Martinengo⁽³²⁾, that ignition delay time is shortened by increased pressure and temperature. Secondly, it is well known from the work of Hsieh & Townend⁽¹⁴⁾, and Lewis & von Elbe⁽²⁵⁾ that the limits of ignition are widened as pressure is increased. The combined effect of the two factors hastened the ignition and consequent pressure rise so that it occurred during the compression. This also explains why in Fig.9.2., the boundary line between Area A and Areas B and D has a negative slope. Further, in Fig.9.4., it is observed that approximately 20 msec. after compression, the rate of pressure drop of Run 375 was slower than that of Run 393; after 80 msec. a distinct hump was noted in the pressure record. The higher temperature enabled further evaporation to form a rich burning mixture, the combustion of which resulted in the hump in the record. Continued burning, especially at higher pressures, would also promote thermal cracking of the fuel. These combined effects may account for the large quantities of wet soot formed.

Since further evaporation was necessary to maintain combustion, the flames in type D results must have been closer to the liquid surface than those in the case of type B, which did not exhibit sustained burning maintained by further evaporation.

At higher compression ratios and pressures, the difficulties in detecting ignition increased, for several reasons. Firstly, as shown in Appendix C, for a fixed quantity of heat liberated during combustion, the resulting percentage increase in pressure decreases with increasing compression ratio. Secondly, higher pressures favour the formation of soot and polymers⁽¹⁰⁾, thus, there is a possibility that the reduction in the total number of moles may result in a pressure reduction which obscures the effect of any heat release. The tendency of soot to absorb hydrogen may also reduce the pressure rise, but this effect probably occurs at a relatively slower rate.

The results obtained indicate that ignition of the fuel occurs in the vapour phase. Thus the thickness and area of the liquid film may have little effect on the results, provided the quantity of liquid is more than sufficient to saturate the air in the compression cylinder.

To summarise the mechanism of compression ignition of a liquid film of fuel in air, the scheme diagrammatically illustrated in Fig.9.5. is proposed to account for the various types of results obtained.

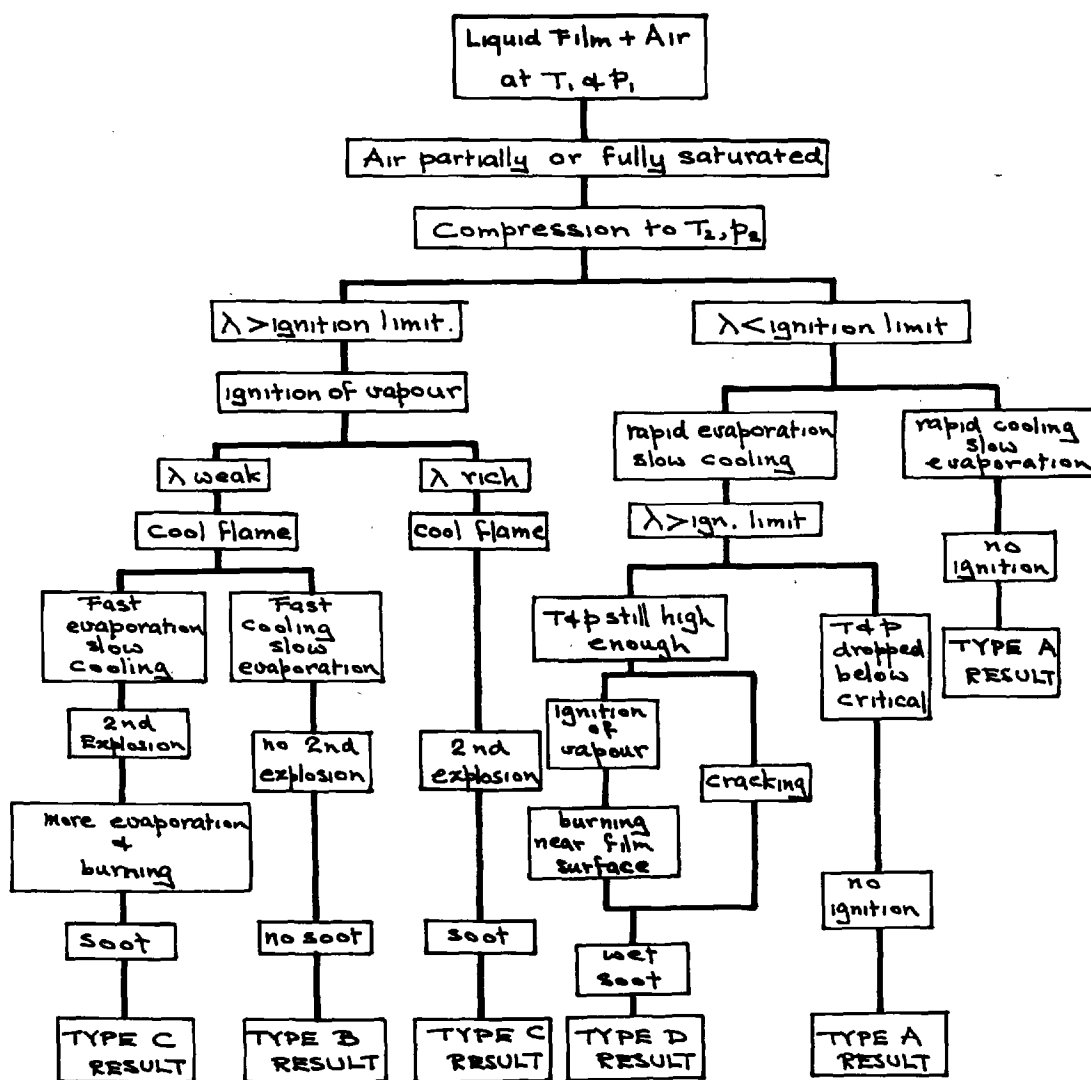


Fig. 95. THE MECHANISM OF FILM IGNITION.

In general, whether ignition occurs after compression depends on a number of factors. These are :

1. The degree of saturation of fuel in the air at T_2 and p_2 .
2. The critical temperature, pressure and mixture concentration of the vapour for ignition.
3. The temperature T_2 and pressure p_2 after compression.
4. The rate of evaporation of the liquid fuel.
5. The rate of cooling of the compressed mixture.

In this project, it was possible to determine the ignition parameters under various temperature and pressure conditions, such as those shown for n-decane film in Fig.9.3. Since factors 4 and 5 are governed by heat and mass transfer conditions, they are affected by the physical configuration of the machine. This is one of the reasons why the ignition parameters determined with this machine cannot be used directly to predict whether ignition will occur in other systems with different physical dimensions.

In any future work, other methods of investigation should be used to confirm the suggested mechanism for the compression ignition of liquid films. For example, an analysis of the vapour and carbon deposit after compression will reveal to what extent cracking of the fuel and polymerisation has occurred in the process of soot formation. Observation of the process through a transparent window by high speed photography will enable the incidence of light emission to be used to indicate the exact time at

which ignition occurs, and it will also serve to locate the positions of the flames in relation to the liquid film.

Furthermore, to be carried out in parallel with the above, suggested future investigations should be an extensive range of compression ignition tests with the relevant fuels and air in the form of vapour mixtures to determine their critical temperature, pressure and concentration limits of inflammability. Since the pressure rises resulting from ignition of very weak mixtures are very small, it is expected that flame detection with photoelectric means would be useful, especially for cases in which only cool flames occur.

A mathematical relation based on the heat and mass transfer between the liquid films and vapour should be developed and tested to correlate experimental data observed in the present compression machine, under known temperature and pressure histories, with the different conditions in practical pipe systems. The results of such correlation could best be expressed in terms of time under equivalent temperature and pressure conditions at which ignition of vapour will occur. If it is known that the temperature and pressure history in a practical pipe system has a time duration in excess of the critical time limit for ignition to occur under equivalent temperature and pressure conditions, as determined by the compression machine experiments, then it could be predicted that ignition will occur in the pipe line.

As oils may sometimes exist in the form of foams in pipe lines, it is also appropriate to investigate the ignition phenomena of foams of fuels under sudden compression. But the treatment of the problem regarding prediction of ignition in practical systems is expected to be similar to that suggested for oil films, because the case of foams merely differs from the case of films in two points, which are :

- (i) an increase in the surface area to liquid volume ratio; and
- (ii) the foam surfaces will be double sided with symmetrical transport phenomena.

CHAPTER 10.Conclusion.

The performance of the constructed compression machine was found to be generally satisfactory. It was operated with drive pressures up to 200 psi to compress within 12 msec., but when the drive gas pressure is increased to 500 psi, which is the designed working pressure, the compression time is expected to decrease to 6 msec. For compression times between 11 and 12 msec., the adiabatic efficiency in compressing air alone was found to range from 96% to 85% when the compression ratio was increased from 8.6:1 to 30:1 respectively.

The apparent values of γ for the compression of n-heptane/air mixtures were found to decrease with increasing mixture strength as expected. For the same mixture strength, reducing the initial pressure p_1 before compression appeared to lower γ slightly. This effect probably occurred because the reduction of the total mass of gas was greater than the reduction of heat loss through the walls during compression.

Ignition delay periods τ_1 and τ_2 were measured for n-heptane/air mixtures at various mixture strengths ranging between $\lambda = 0.2$ and $\lambda = 1.2$. The effect of increasing λ was found to increase τ_1 and decrease τ_2 , furthermore τ_2 tended to zero when λ became 1.0. Consequently, the total delay time was found to be a minimum at approximately $\lambda = 0.4$. The results indicate, however, that the mixture strength for minimum total delay time is temperature as well as pressure dependent.

In general, an increase in the pressure of temperature at the end and the compression tended to shorten τ_1 and τ_2 . The experiments also indicated that, for constant final temperature T_2 after compression, the effect of a lower compression pressure p_2 was to increase both τ_1 and τ_2 . This is in accord with Rogener's expression⁽³³⁾ that :

$$\tau = a p^{-n} e^{\frac{E}{RT}} .$$

Near the lower mixture concentration limit for ignition, only cool flames were propagated.

The effect of combustion products deposited on the cylinder walls was to shorten the delay periods.

Published ignition delay results for stoichiometric n-heptane/air mixtures in other compression machines were collected for the purpose of comparison. Although the results previously attained were very scattered, those obtained with this machine appeared to lie very near to the average values, over a wide range of conditions.

Experiments on the compression of n-decane films in contact with air, indicate that the consequent ignition is strongly dependent on the temperature before compression. If the initial temperature T_1 is below 20°C., no ignition occurs after compression even when the compression ratio is as high as 30:1. For T_1 ranging between 24°C. and 38°C., a moderate pressure rise, probably caused by a cool flame, occurs after compression. For temperature T_1 higher than 38°C., a moderate to severe pressure rise occurs, and it is accompanied by soot formation. The saturated vapour for T_1 between 24°C. and 38°C. coincides with mixture

strengths having fuel concentration values very near the limit of inflammability. Ignition in this range very much depends on the evaporation rate and cooling rate of the compressed material. If the conditions after compression are such that the temperature and pressure are higher than 800°K and 900 psia respectively, the tendency to form soot, and also the tendency for the liquid fuel to crack, increases.

As a result of the experiments, an idea of the mechanism of liquid film ignition has been formed. The combustion reaction is thought to occur mainly in the vapour phase. The question whether ignition occurs under certain conditions is dependent on the following factors:

1. The degree of saturation of the fuel in the air above.
2. Critical temperature, pressure and mixture concentration of the vapour for ignition.
3. Temperature and pressure condition after compression.
4. Rate of evaporation of the fuel.
5. Rate of cooling of the compressed mixture in its particular surrounding.

Further investigation by means of optical observation, using high speed photography, as well as analysis of the products, are suggested to verify the mechanism so far conceived.

List of Plates.

- I. View of compression machine inside protective cabinet.(p.148).
 - II. View of compression cylinder with displacement measuring unit.(p.149)
 - III. Electronic equipment with the pendulum time switch in the left hand back ground. (p.150).
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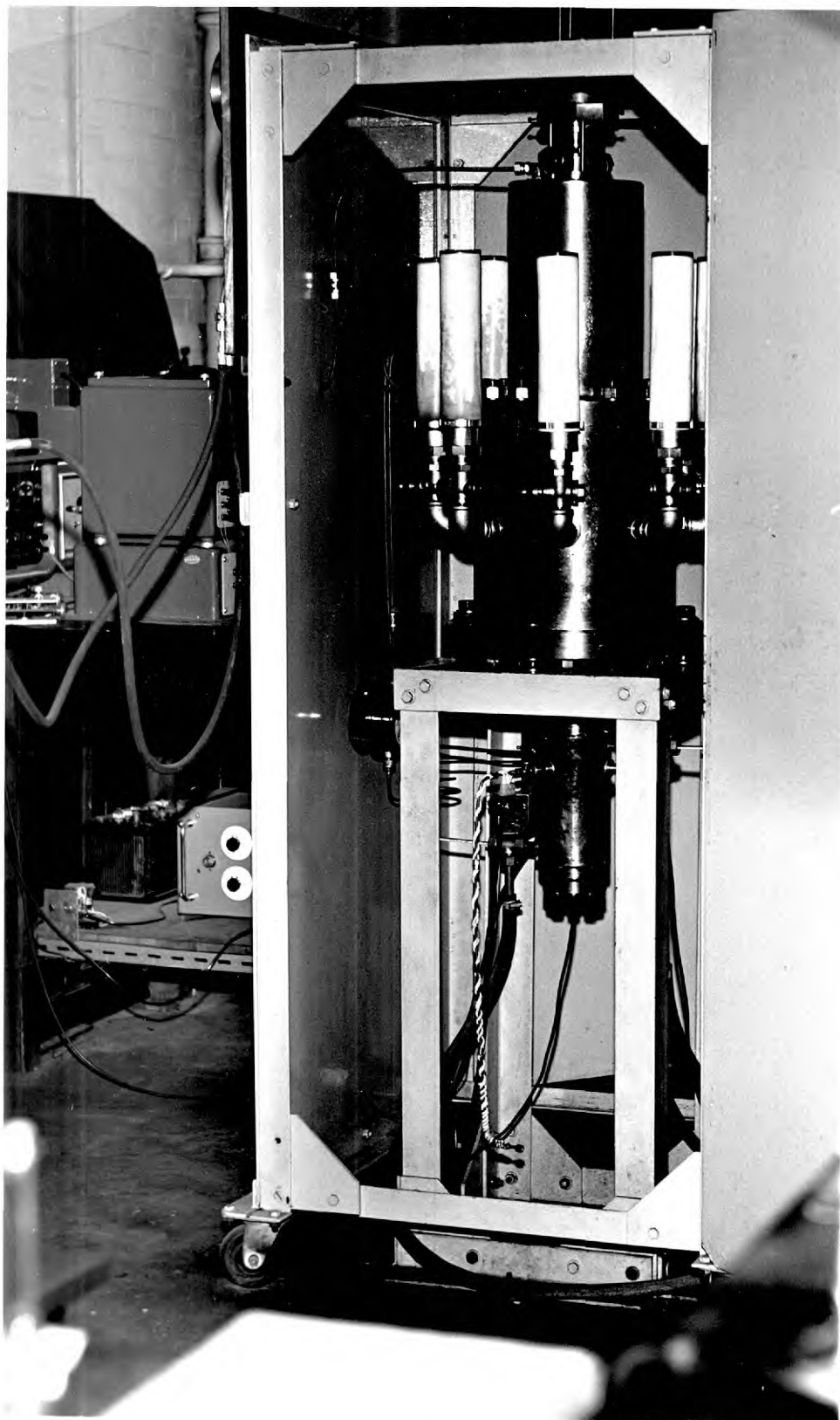


PLATE I

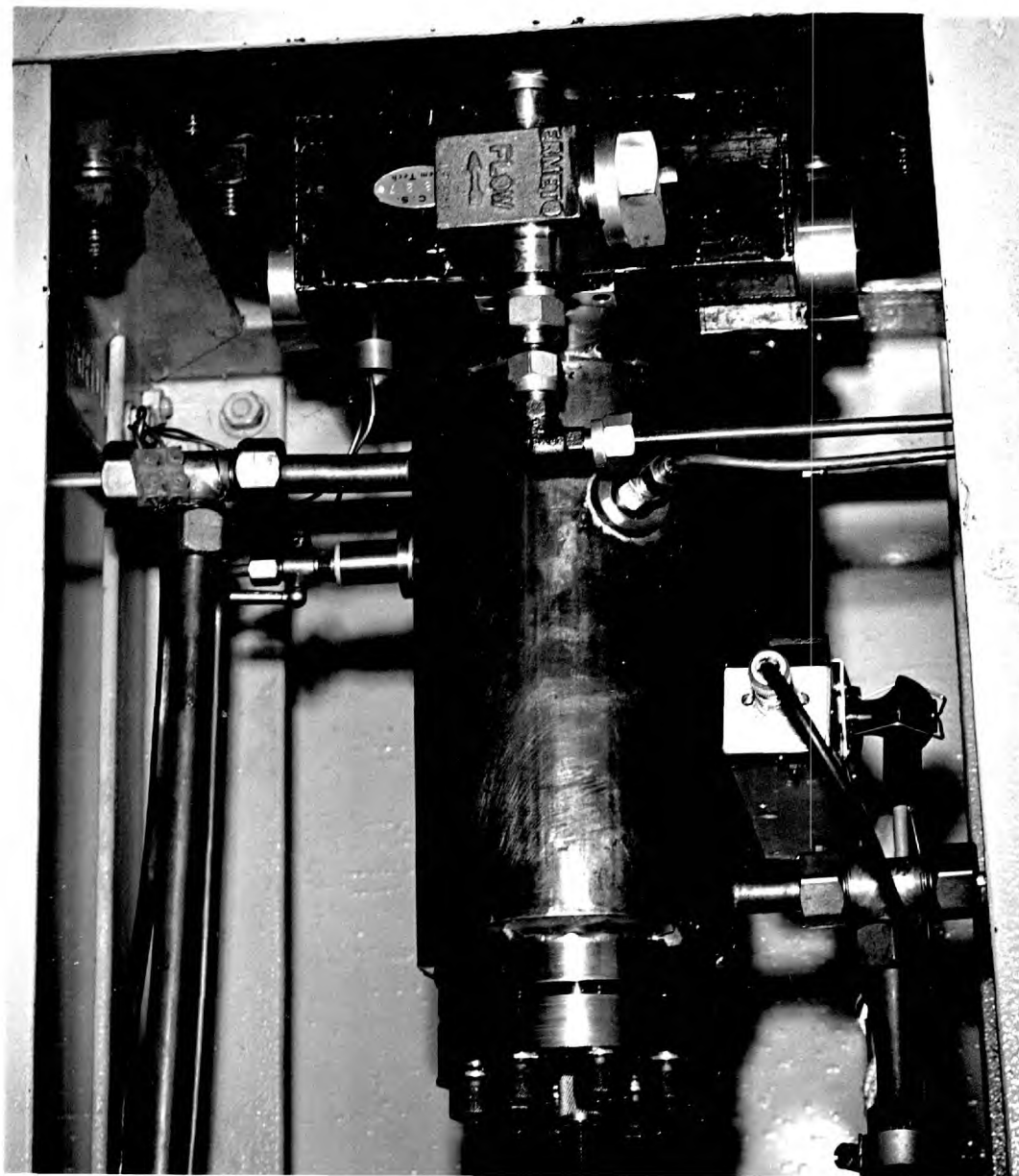


PLATE II

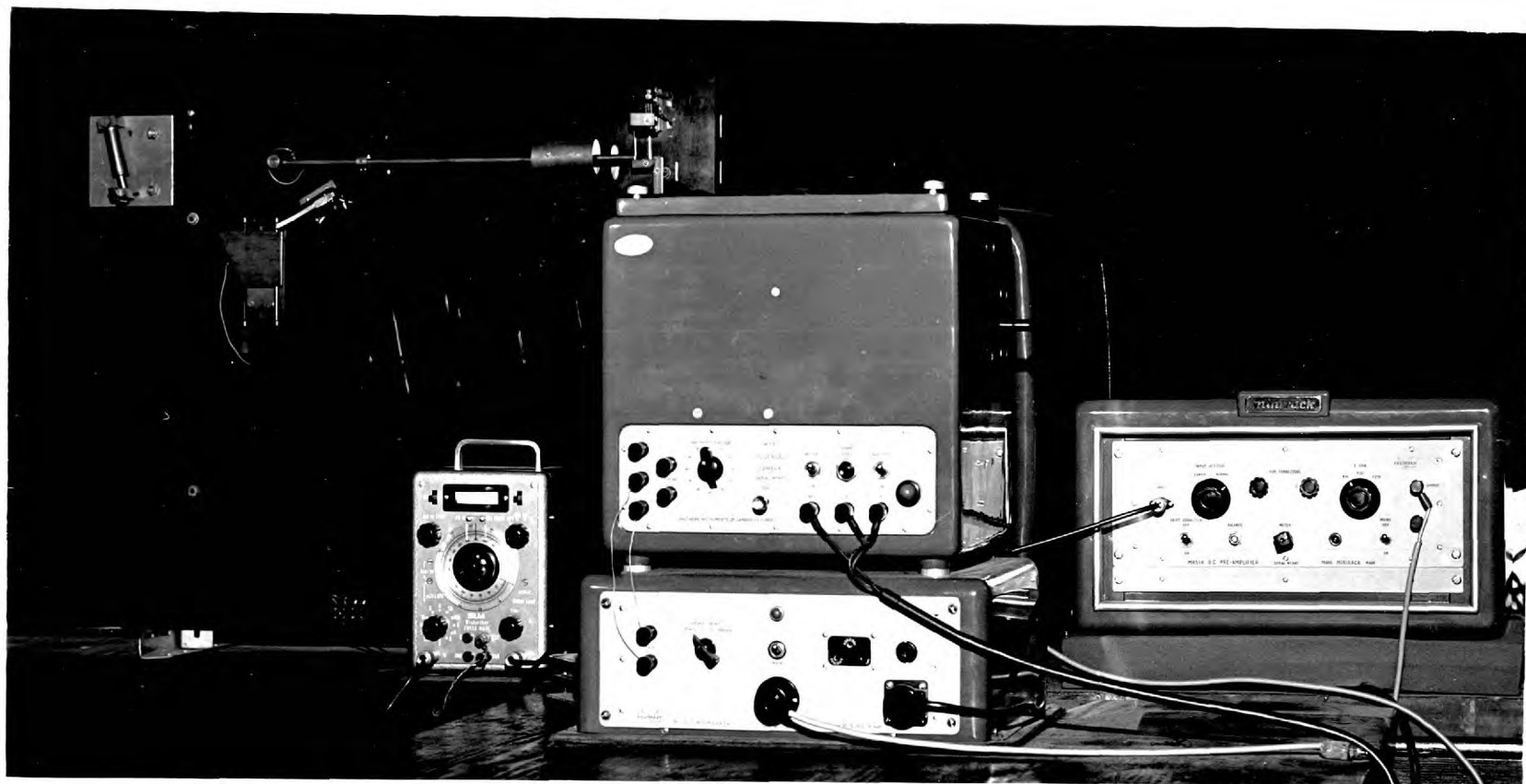


PLATE III

Appendix A.Calculation of the compression ratio necessary
to ignite various fuel-air mixtures.

Case I: For vapour fuel-air with minimum ignition temperature of
250°C. (e.g. n-heptane/air).

If the vapour is compressed at an initial temperature of 20°C.,
and if it is assumed that $\gamma = 1.3$, then by the relationship

$$T_2 = T_1 \left(\frac{V_1}{V_2} \right)^{\gamma-1}$$

it may be shown that the minimum compression ratio needed to attain a
temperature of 250°C. is 7:1.

Case II: For a fuel initially in the liquid phase.

It is assumed that the initial temperature is 50°C., the ignition
temperature is 600°C., the latent heat of vaporization of the fuel is
100 Btu/lb; and that after compression, the liquid is to be vaporized
into the air to form a stoichiometric mixture (say, $\frac{\text{Air}}{\text{fuel}} = 14.5$ by weight).

To obtain a final air temperature of 600°C. (873°K) without vapor-
izing the fuel, the necessary compression ratio is given by

$$T_2 = 873 = 323 (\text{C.R.})^{0.3}$$

$$\text{i.e. } \text{C.R.} = 27.5$$

The resulting air pressure is

$$p_2 = p_1 (\text{C.R.})^{1.3} = 14.7 \times 27.5^{1.3} = 1090 \text{ psia.}$$

One cubic foot of initial cylinder volume contains 0.0625 lbs. of
air, thus 0.0043 lbs. of fuel has to be evaporated per cubic foot to

produce a stoichiometric mixture. The evaporation requires a heat input of 0.43 Btu, which can be provided by the work done in further compression from p_2 to p_3 . The work done is given by :

$$W.D. = \frac{\gamma}{\gamma-1} p_2 v_2 \left[\left(\frac{p_2}{p_3} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]$$

From which it may be calculated that

$$\frac{p_2}{p_3} = 1.059.$$

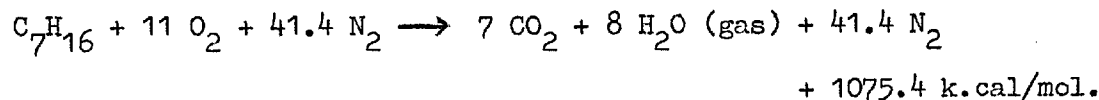
Therefore the overall compression ratio required is

$$27.5 \times 1.059 = 29.1$$

As the assumed value for $\gamma = 1.3$ is less than the theoretical value, the estimated compression ratios are thought to be sufficiently high to allow for heat losses during compression.

Appendix B.Explosion pressure of n-heptane/air mixture.

It is assumed that the mixture strength, at atmospheric pressure, and 300°K is stoichiometric and that the combustion process takes place at constant volume. If combustion is complete and dissociation is neglected, then the reaction may be represented by the equation



Let E = energy content of the gases

T_i = initial temperature

T_E = maximum temperature after explosion

p_E = " pressure " "

n = number of moles.

The maximum temperature after explosion T_E is calculated by equating the energy contents of the gases before and after the combustion.

The energy of the 'products' before combustion at 300°K is as follows:-

	n	E_{300}	$n \times E_{300}$
CO_2	7	1.66	11.6
$\text{H}_2\text{O (gas)}$	8	1.79	14.3
N_2	41.4	1.49	61.6
	$\sum n = 56.4$		$\sum n E_{300} = 87.5 \text{ k.cal.}$

After combustion, the total energy at T_E is $\sum n E_{T_E} = 87.5 + 1075.4 = 1162.9 \text{ k.cal.}$ From data for the energy content of gases ⁽²⁵⁾

At 2800°K $\sum n E = 1104 \text{ k.cal.}$
 and At 3000°K $\sum n E = 1197 \text{ k.cal.}$
 by interpolation $T_E = 2927^\circ\text{K.}$

The resulting pressure rise ratio is given by

$$\frac{P_E}{P.} = \frac{T_E \times n E}{T. \times n E} = \frac{2927 \times 56.4}{300 \times 53.4} = \underline{\underline{10.03.}}$$

Appendix C.Explosion Pressure of Compressed Gases.

To determine the pressure change as a result of ignition, after compression is completed in a compression machine, the first law of thermodynamics may be used.

If V_1, V_2 = volume of chamber before and after compression resp.

T_1, T_2, T_E = temperature before compression, after compression and after explosion.

P_1, P_2, P_E = pressure before compression, after compression and after explosion.

r = compression ratio.

γ = ratio of specific heats.

C_v = specific heat at constant volume.

R = gas constant.

W = mass of gas.

and J = mechanical equivalent of heat.

The first law of thermodynamics may be written as

$$\begin{aligned}
 Q &= \left(\cancel{P_E - P_2} \right) \frac{V_2}{\cancel{J}} + (T_E - T_2) C_v \\
 &= \left(\cancel{P_E - P_2} \right) \frac{V_2}{\cancel{J}} + (P_E - P_2) \frac{V_2 C_v}{W R} \\
 &= \left(\frac{1}{J} + \frac{C_v}{WR} \right) (P_E - 1) P_2 V_2 \\
 &= \left(\frac{1}{J} + \frac{C_v}{WR} \right) \left(\frac{P_E}{P_2} - 1 \right) P_1 r^\gamma V_1 \\
 &= K \left(\frac{P_E}{P_2} - 1 \right) r^{\gamma-1} \dots (I)
 \end{aligned}$$

where K = constant.

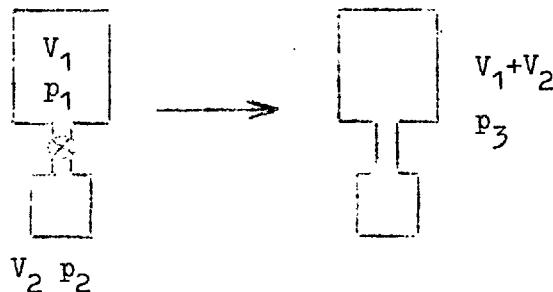
Equation (I) shows that, for a fixed heat of combustion Q , the pressure rise ratio $\frac{p_E}{p_2}$ decreases as compression ratio r is increased.

For example, the calculations in Appendix B show that, when a stoichiometric n-heptane-air mixture is ignited at an initial temperature of 300°K and pressure of 1 atmosphere, the resultant constant volume explosion pressure p_E is 10 atmospheres. If the value $p_E/p_2 = 10$ is substituted into Equation (I), then it may be calculated that the value of $\frac{p_E}{p_2}$ is reduced from 10 to 3.6 as the compression ratio r is increased from 1:1 to 30:1. This is illustrated graphically in Fig.11.1.

Appendix D.

Correction in initial drive gas pressure setting to allow for dead space below release valve.

Below the release valve (part 8 in Fig.3.1.) and the drive piston, there is a volume of 28 in^3 which is not initially filled by the driving gas because of the closed valve. Let the volume be V_2 at pressure p_2 while the initial drive gas at pressure p_1 occupies a volume V_1 above the valve. At the instant the valve is opened, the initial drive gas pressure expands adiabatically to p_3 and occupies volume $V_1 + V_2$.



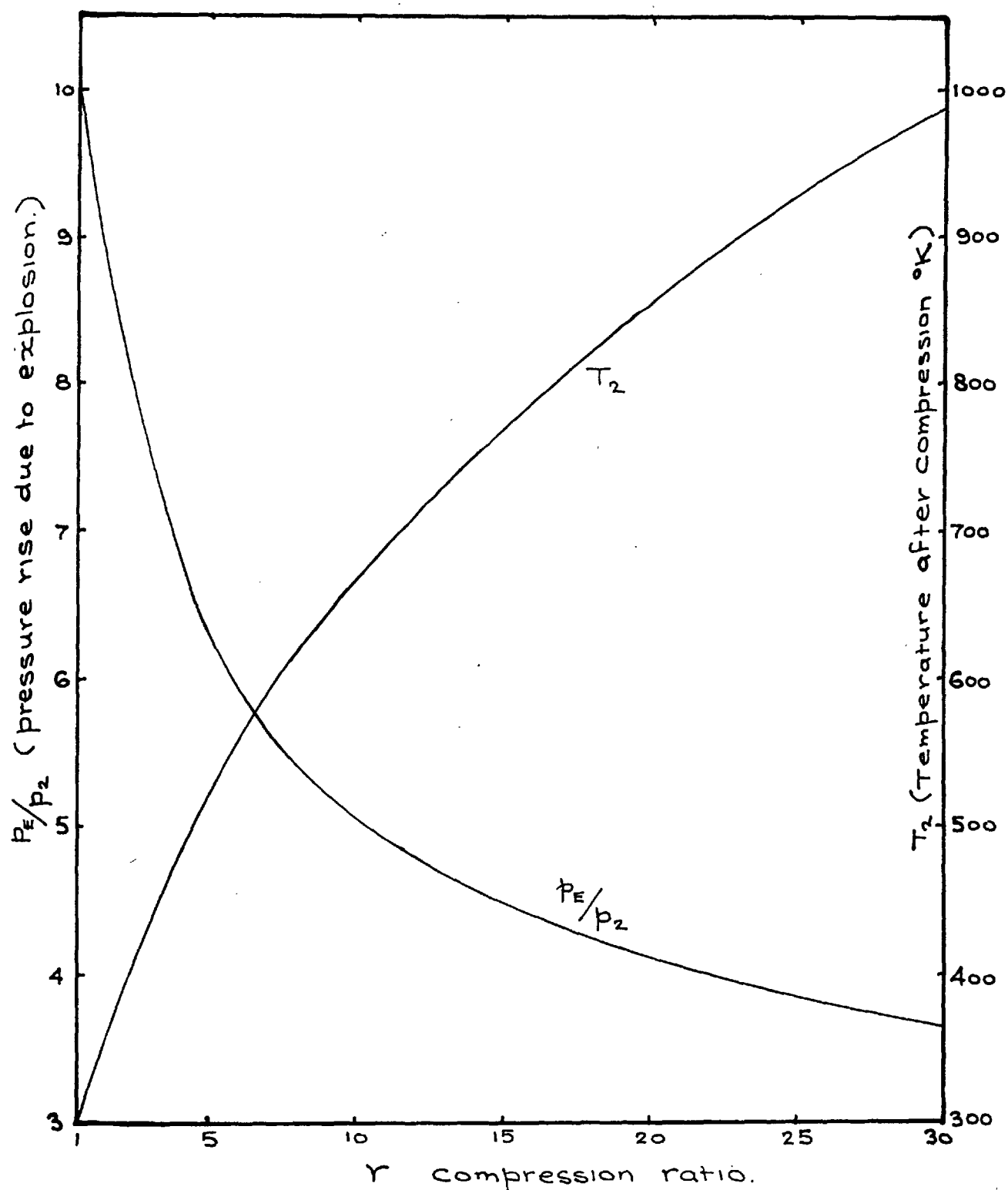


FIG. 11.1. CALCULATED EXPLOSION PRESSURES OF
COMPRESSED STOICHIOMETRIC n HEPTANE-AIR MIXTURE.

$T_1 = 300^{\circ}\text{K}$ & $\gamma = 1.35$

The correction to the initial drive gas pressure setting to allow for the dead volume is such that, for an initial pressure p_{K1} , calculated from Equation (3) in Section 3.3.1., the necessary corrected pressure setting should be $p_{K1} \times \frac{p_1}{p_3}$.

After the valve is opened,

$$p_3 = p_1 \left(\frac{V_1}{V_1 + V_2} \right)^{1.4} + p_2 \left(\frac{V_2}{V_1 + V_2} \right)^{1.4}$$

Now $V_1 = 450 \text{ in.}^3$, $V_2 = 28 \text{ in.}^3$ and p_2 may be neglected in comparison with p_1 .

Thus, $\frac{p_1}{p_3} \approx 1.088$

Therefore, the initial pressure setting should be $1.088 p_{K1}$.

Appendix E.

Stress calculation and mechanical details of the design.

In the following, the part numbers in parenthesis refer to the drawing of the machine in Fig.3.1.

Drive gas container. (6).

The vessel was of welded construction so as to avoid pressure sealing problems and to facilitate the alignment of the pressure release valve. Since the drive gas was stored in gas cylinders at a maximum pressure of 2000 psi, this pressure was taken as the maximum working pressure of the drive gas container.

If it is assumed that elastic failure of a thick walled cylinder made of mild steel is determined by the maximum shearing stress, then the internal pressure needed to cause the cylinder to yield at the bore is given by $p_y = \frac{\delta_y}{2} \left(\frac{k^2 - 1}{k^2} \right)$ where δ_y is the yield stress of the steel in tension and $k = 1.18$ the ratio of external to internal diameters of the gas container. Taking $\delta_y = 13 \text{ tons/in.}^2$ for mild steel, then

$$p_y = \frac{13 \times 2240}{2} \left(\frac{1.39 - 1.00}{1.39} \right) = 4110 \text{ psi.}$$

Thus, the minimum factor of safety of this component based on the ratio of the yield pressure to the maximum working pressure is 2.05. In view of the large volume of gas contained in the vessel under pressure, this factor of safety is not considered excessive.

The end plate thickness for both the upper and lower ends and internal fillet radii were designed according to B.S. 1500 : 1958 using a design pressure of 2000 psi and a design stress of 13,000 psi. The details of the joint between the cylindrical shell and the top plate were taken from the work of Lane, who showed that the joint had a satisfactory fatigue life. To compensate for the weakening of the plate caused by the necessary opening for the valve stem, the practice of increasing the end plate thickness by a volume equivalent to the volume of the metal removed was adopted. Compensation was not provided for the lower plate, because rigid clamping by the $12 - \frac{3}{4}$ in. studs on the outer periphery of the plate connecting the container to the cylinder casing (12) rendered this unnecessary.

In manufacture, the parts were first rough machined, then welded together in accordance to B.S. 1500 : 1958. The vessel was tested ultrasonically for the weld quality and for any possible flaws, followed by stress relief at a temperature between 600°C. and 650°C. for two hours. Next, the vessel was hydraulically tested to a pressure of 2800 psi; and finally it was finish-machined.

Release valve. (8).

The valve was designed to hold a maximum gas pressure of 2000 psi, at which the stress in the smallest cross-section of the valve stem is 31.8 tons/in.² This component was made of En.25 nickel chromium steel having a yield strength of 52 tons/in.²

The valve seating was sealed by a neoprene 'O' ring housed in a dove tail shaped groove.

Shear disc. (2).

On the upper end of the valve stem is a brass shear disc retained by a hardened steel punch and a retaining nut. The disc, being of a larger diameter than the valve stem, is supported outside by a hardened steel die which is press fitted into the nut (3). When this nut is tightened, the valve is raised and eventually closed. The valve may be opened by dropping a seven pound weight on to the valve stem to shear the brass disc between the punch and die. These were both made of cast steel, heat-treated to a hardness between 52 and 54 Rockwell 'C'. It was decided that the diametric clearance between the punch and die should be 0.004 in., approximately 10% of the average thickness of the shear discs which were expected to range between 0.020 in. and 0.060 in. thick.

Drive cylinder.

This consists of a cylinder casing (12) and a cylinder liner (10). The former was made from a forging of Siemens Martin Basic .20/.33% carbon steel and the internal bore was hone finished. The outside of the liner, which was made of centrifugally cast iron of Brinell hardness between 179 and 240, was ground so that there was an interference of 0.003 in. on the diameter between the casing and the liner. A maximum force of 16 tons was needed to press fit the liner in place, after which the bore of the liner was honed to a diameter of 8.000 ± 0.0005 in. and a surface finish of approximately 15 microinches.

As a result of the press fit, the liner is well supported over most of its length, except at that area where there is a recessed annulus in the casing surrounding the exhaust ports of the liner. Thus, the largest tensile stress in the liner is located in the $\frac{1}{2}$ in. wide portion of the unsupported area when the piston is in the bottom position and the unsupported area is subject to the final driving gas pressure. If it is assumed that the initial driving gas pressure is 500 psig., the final pressure will be 265 psi. As a result of the pressure, the liner is subject to a tangential tensile stress f_t and in addition it is subject to a tangential compressive stress f_c arising from the press fit. Assuming that Young's modulus for cast iron is 18×10^6 psi, and that the liner may be treated as a 'thin cylinder', then it may be shown that the compressive stress f_c is -6200 psi and the tangential stress 8900 psi. Thus, the resultant tangential stress is 2700 psi, which is acceptable as cast iron may be assumed to have an ultimate tensile strength of 15 tons/in.²

In order that the exhaust port positions in the liner may be adjusted, two spacers one of $\frac{3}{32}$ in. and another $\frac{5}{32}$ in. thick were provided. One of these was placed above and the other below the liner. By interchanging them or having both either above or below the liner, it is possible to locate the exhaust ports in four different positions within a range of $\frac{1}{4}$ in. If the liner is taken out of the casing and refitted upside down, the range of adjustment can then be extended ^{over} ~~in~~ $\frac{7}{16}$ in.

Radial arrangement of exhaust ports.

The maximum width of an exhaust port in the tangential direction is governed by the need to prevent the ends of the piston rings from protruding inside the ^{ports} ~~liner~~ and causing damage. In internal combustion engine design, it is generally considered that a port 18° wide is the maximum safe width. So as not to weaken the liner too much by removing an excessive width of metal between ports, it was decided to use 16 ports each 15° wide equally spaced round the cylinder.

It was impracticable to provide means to prevent the piston from rotating relative to the cylinder, and the possibility of rotation during the piston stroke had to be borne in mind when the exhaust ports were designed. For a fixed axial position of the piston, the effect of piston rotation on the total port area opened is dependent on the number and sizes of ports in the piston. This is illustrated in Table I, in which the effective port areas are expressed in terms of the angles subtended by the width of the ports. As the piston is rotated in relation to the

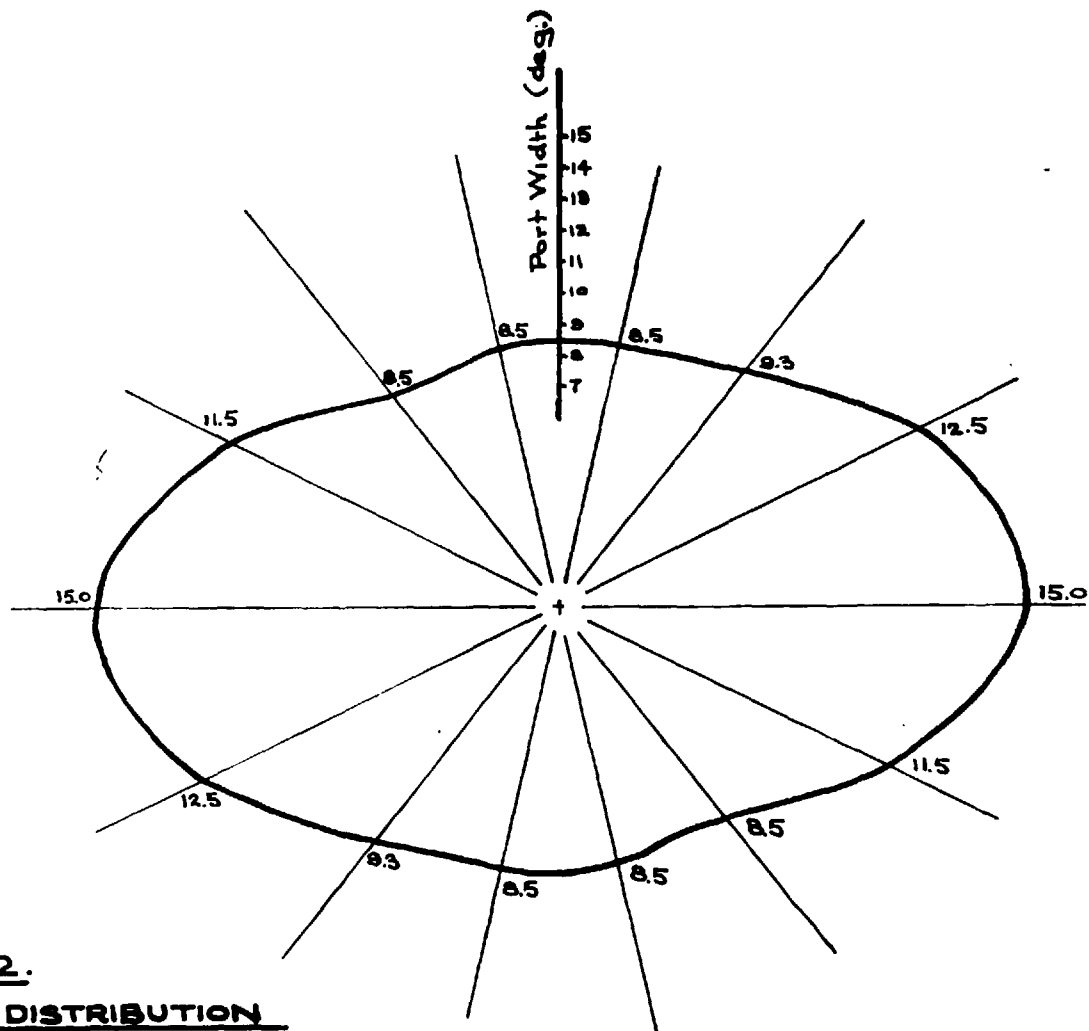


FIG. 11.2.

**POLAR DISTRIBUTION
OF PORT OPENING SIZES.**

14-16° Ports in piston.

16-15° Ports in liner.

liner, the total effective port width varies about a mean value; but it was found that the percentage variation could be greatly reduced when using 15 instead of 16 ports in the piston, although a small sacrifice in the mean value of the total effect port width had to be made.

Table I : Total effective port widths for various exhaust port combinations.

No. of Ports and radial port width.		Total effective port width.			
In liner.	In Piston.	Mean	Variation due to rotation.		
			Maximum	Minimum	% Variation.
16 x 15°	16 x 15°	188°	240°	136°	± 27.7
16 x 15°	15 x 15°	150°	151°	149°	± 0.66
16 x 15°	14 x 16°	147°	148°	146°	± 0.68

A study of the polar distribution of the widths of opened ports showed that the combination of 15 ports in the piston and 16 ports in the liner caused an unsymmetrical distribution. Had this combination been employed, a side force would have been generated whilst the machine was discharging exhaust gas. The assymetry, however, does not exist when 14 ports are used in the piston. The polar diagram given in Fig.11.2. shows the distribution of opened port width for the combination of 14 equally spaced ports of 16° width in the piston and 16 ports of 15° width in the liner.

Drive Piston. (9).

In order to avoid excessive weight of the moving parts, it was decided to construct the drive piston from an alluminium alloy. Hiduminium RR.58 alloy (see Appendix F) having an ultimate tensile strength of 28 tons/in.² and an elongation of 8% was chosen for this purpose. This material has good shock resistance and, although its maximum service temperature (300°C.) is below the adiabatic temperature of the damping gas, the duration of the peak temperature is so short that the piston will not sustain any damage.

A conical shaped piston top was desirable for two reasons. Firstly, because it saved height, i.e. it allowed room for the release valve (8) while the exhaust ports in the piston were still as near the top of the piston as possible; and secondly, from the point of view of stress distribution, the bending stresses developed when a load is applied to the connecting rod are smaller than those which would have existed with a flat top.

It was calculated in Section 3.4. that when a driving pressure of 500 psia is used, the maximum deceleration and acceleration will be 3700 g. and 960 g. respectively. The greatest load on the piston will therefore occur during the period of maximum deceleration when the mass of the combustion piston and connecting rod (7 lb.wt.) result in an axial inertial force of 26,000 lbs. in the drive piston. In order to reduce the concentration of this loading, large steel spacers were used above and below the piston faces at the point where the connecting rod was attached to

the piston, such that the maximum bearing stress on the piston did not exceed 9 tons/in.² under the above conditions. This is less than a third of the 0.1% proof bearing stress of the alloy with which the piston is made.

The conical head of the piston was shaped so that a fairly uniform stress distribution was achieved. It is calculated that the highest shear stress concentration of 1.8 tons/in.² occurs at a diameter of 1¹/₂ in. near the periphery of the upper spacer where the metal thickness is 1³/₈ in. thick, this stress is one tenth of the ultimate shear stress of the alloy. Furthermore, it is assumed that the inertial axial load of 26,000 lb. in the downward direction is opposed by tensile forces acting along the conical surface towards the upper edge of the piston. Since the half angle of the cone is 40° and the smallest cross-section area through which this force acts is 10.4 in.², the maximum tensile stress f_t is given by

$$f_c = \frac{26,000}{10.4 \cos 40^\circ} = 3,280 \text{ psi.}$$

In the manufacturing process, the hand forged billet was rough machined to within 1/2 in. of the final dimension, ultrasonically tested, then heat-treated and stress-relieved in accordance with aircraft specification DTD.731. The component was then machined to the final dimensions.

Drive Piston Rings.

The diametric clearance between the piston and the cylinder sleeve bore was 0.003 in. To seal the gap, three piston rings were inserted in the drive piston. The top and middle rings were arranged with the pressure side facing upwards, and the bottom ring with pressure side facing downwards.

Since it is essential to hold pressure with the piston in a stationary condition, leakage past the ring gaps had to be kept to a minimum. Therefore, the gaps were sealed with scarf-stepped joints which allow circumferential expansion of the rings and at the same time prevent leakage from the pressure side of the rings towards the back. Rings of cast iron, made by Wellworthy Ltd. were used. The $\frac{3}{16}$ in. wide rings had a radial depth of 0.26 in. and their fitted closed gaps were 0.008 in./0.015 in. In order to locate the rings in their correct position and also to provide an initial sealing pressure, crimped side springs were placed on the pressure side of the rings. The ring grooves in the piston were 0.251 in. wide and 0.267 in. deep. The external faces of the rings were of fine turned finish to facilitate quick bedding in on the honed bore of the liner. Furthermore, to obtain equal rates of wear of the piston rings and the cylinder liner, and to produce a smooth surface finish on both components, relatively harder piston rings (Brinell hardness 235 min.) are used with a softer cylinder liner (Brinell hardness 179-240).

Connecting rod. (13).

The chief stresses in the connecting rod arise as a result of the maximum acceleration of 960 g. and deceleration of 3700 g. to which the rod may be subjected.

When accelerating, the maximum bearing load on the rod occurs at that point where the thrust is transmitted to the steel spacer and drive piston.

The bearing stress $f_b = \frac{\text{accelerating force due to the rod}}{\text{bearing area}}$

$$= \frac{\frac{7}{8} \times 960 \text{ g.}}{\frac{\pi}{4} (1.0^2 - 0.643^2)} = 14,600 \text{ psi} = \underline{\underline{6.5 \text{ tons/in.}^2}}$$

When decelerating, the maximum stress occurs in the core of the threaded portion near the top end of the rod.

$$f_t = \frac{\frac{7}{8} \times 3700 \text{ g.}}{\frac{\pi}{4} \times 0.643^2} = 79,700 \text{ psi} = \underline{\underline{35.6 \text{ tons/in.}^2}}$$

The maximum shear stress in the base of the threads as a result of the deceleration is calculated to be 8.2 tons/in.² En. Spec. 25 steel was used for this part.

A $\frac{1}{4}$ in. diameter hole was drilled axially for $3\frac{1}{2}$ in. from the lower end of the rod to provide a passage for the leads of a pressure transducer, should the transducer be mounted inside the piston.

Midgland. (14).

A special housing was provided in the mid-flange (17) to accommodate a Chevron packing seal through which the connecting rod passes. The housing is inserted from below the mid-flange, so that the packing may be replaced without dismantling any part of the machine above the mid-flange.

Mid-flange. (17).

The mid-flange of mild steel was designed according to B.S.1500 : 1958 for a maximum pressure of 1,770 psi, which corresponds to the condition for a driving gas pressure of 500 psi.

On the upper face of the flange, a groove $\frac{1}{8}$ in. deep (15) was machined and filled with a 50/50 tin lead solder. The purpose of the solder is to adsorb the impact energy by deformation in the case of a 'bad landing', because the hardness of the solder (Brinell 15) is very much less than that of the drive piston.

Combustion Cylinder. (24).

The combustion cylinder was made of nickel chromium molybdenum steel En.Spec.25, having an ultimate tensile strength of 50 tons/in.²

The cylinder is capable of working pressures in excess of 5,000 psi, because its external dimensions were originally designed for a working pressure of 5,000 psi should it become necessary to enlarge the bore from 1 1/2 in. to 2 in. in future.

The bore of the cylinder was honed to a surface finish of approximately 4 microinches.

Combustion Piston. (22).

Two pistons were designed : One with two seals so that both pressure and vacuum could be contained in the cylinder; the other with one seal only was designed to accommodate a pressure transducer inside the piston crown.

Both pistons illustrated in Fig.5.2. may be fixed to the connecting rod in the same manner. By maintaining a minimum clearance between the spigot at the end of the connecting rod and the corresponding recess in which it fits in the piston, satisfactory alignment between the piston and connecting rod was achieved. Both pistons were made of Ni-Cr-Mo steel En. Spec.25.

The double seal version of the pistons has a stainless steel (Appendix F) crown, the outside periphery of which has a width of $\frac{3}{16}$ in.; this is the only section of the piston which is in contact with the cylinder bore. To avoid direct metal to metal contact, the outside periphery of

the piston crown was sprayed with a 0.003 in. thick layer of P.T.F.E. so that the net clearance between the piston crown and cylinder bore was 0.0005 in. The 'U' seals made of P.T.F.E. were supplied by Dowty Seals Ltd. On the pressure side of each 'U' seal, a neoprene 'O' ring was placed inside the 'U' section, so as to occupy the dead space. Since all contact between the piston and cylinder bore is via P.T.F.E., it is possible to operate the piston without lubricating oil.

In the version in which a pressure transducer is accommodated, the 'U' seal is retained by a retaining ring press fitted on a spigot which is tapered with a 1.5° half angle. A tapered spigot was used in preference to a straight one because it is easy to dismantle.

Combustion Cylinder Head. (26).

The compression ratio is determined by the compressed volume; two stainless steel (Appendix F) cylinder heads containing wells of different volume, and five cylinder head spacers of different thicknesses, enabled the range of compression ratios from 8.85:1 to 30:1 to be covered in 10 steps.

The cylinder head was sealed by an 'O' ring, as is illustrated in Fig.6.2., and retained in position by a flange and eight $\frac{1}{2}$ in. bolts.

Water Jacket. (25).

The weight of the steel tube water jacket is supported by three brass pipe connectors which are screwed radially into the compression cylinder and to which they provide access. Rubber 'O' rings are used for sealing all the joints, so that the thermal expansion of the jacket

can be accommodated. All surfaces of the jacket and cylinder which are in contact with the heating fluid are protected by 'Metagalv', a cold galvanising paint.

Machine alignment.

Both the release valve in the drive gas container and the moving assembly, consisting of the drive piston, connecting rod and the combustion piston, had to be accurately aligned.

The alignment of the release valve was accomplished by making the drive gas container as a welded unit and machine turning the valve guide and seating in the completed unit. Good alignment of other parts was maintained, firstly, by arranging for those contact faces which are perpendicular to the axis of the machine to be of large area; and secondly, by using the smallest clearances practicable between all concentric spigots and corresponding counter bores.

The design tolerances and clearances of the important dimensions are summarised in Table II.

Table II : Clearances between machine parts.

[illegible]

Table II - contd.

Part			A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P
Name.	Dimension & Tolerance.																	
E	Cylinder Casing	8.750" - 0				-2	1			3	1							
	I.D.	- 1				-4	5			8	4							
F	Liner spacer O.D.	8.750"- 2					1											
		- 6					5											
G	" " I.D.	8.000" + 5	7															
		+10	13															
H	Top Vessel spigot	8.750"- 4					3											
		- 8					8											
I	Midflange upper spigot.	8.750"- 2					1											
		- 4					4											
J	" bottom bore	4.250"+1.5											0					
		- 0											22					
K	Combustion Cyl. spigot.	4.250"+ 0											0					
		-0.7											22					
L	" Cyl.bore.	1.500"+ 1															0	
		- 0															1.5	
M	Con.Rod upper end	0.750"+ 0	0															
		-0.5	1.5															
N	" " bottom "	0.625"+ 0																0
		- 1																3
O	Combustion piston O.D.	1.500"+ 0											0					
		-0.5											1.5					
P	" piston I.D.	0.625"+ 2														0		
		- 0														3		

Units of tolerances and clearances are in 0.001".

Negative values indicate interference.

Suspension of the machine.

The machine is supported at four points under the mid-flange by disc springs (18), which have a combined stiffness of 31,500 lb/in. and a maximum deflection of 0.22 in.

Appendix F.Specification of Metals used.

1) Nickel Chromium Molybdenum steel En. Spec. 25V.

2) Stainless steel (E.M.S.). En. Spec. 58M.

3) Aluminium Alloy

Hyduminium RR.58.

Aircraft spec. DTD.731.

4) Rolled Brass sheets.

(for shear discs, part 2 in Fig.3.1.).

Composition: 63/37 Cu.Zn. to B.S.265

Condition: half hard.

The ultimate shear strength and the safe working shear strength of the sheets were determined by loading the material in the punch and die assembly taken from the machine under an hydraulic press. The results are shown in Fig.11.3.

A 7 lb. weight, falling through a distance of 18 in., was found capable of shearing a 22 S.W.G. disc loaded at any gas pressure between 80 and 310 psi.

Appendix G.

Time lag in measuring damping gas pressure
through a long passage.

If pressure is measured through a long passage the time lag between the application of pressure at one end of the passage and the pressure rise at the other end depends on the velocity of propagation of the pressure wave through the medium⁽⁴⁴⁾.

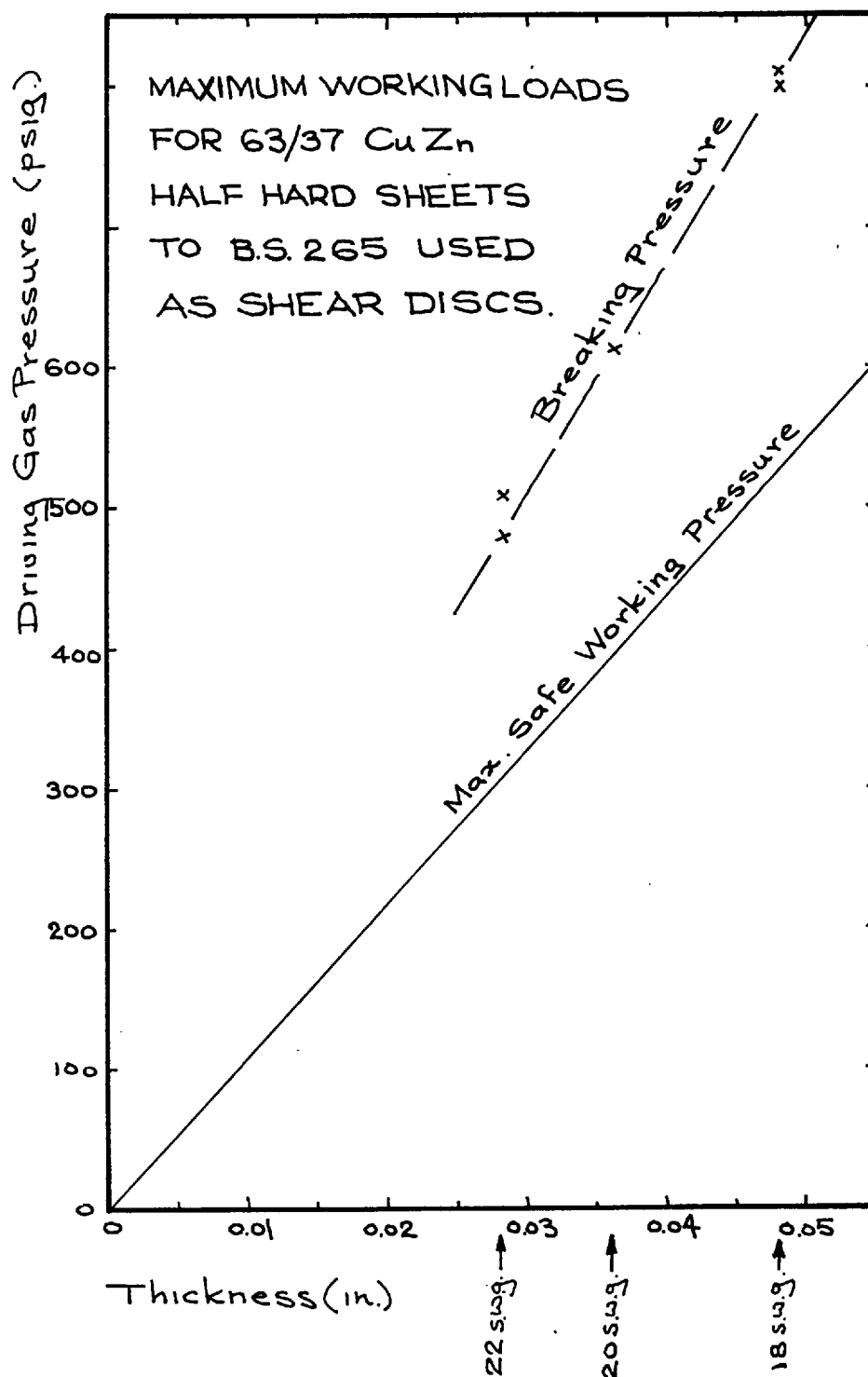


FIGURE 11.3.

The Sonic velocity is given by

$$v_s = \sqrt{\gamma R g T_2} = \sqrt{\gamma R g T_1 \left[\frac{p_2}{p_1} \right]^{\frac{\gamma-1}{\gamma}}}$$

where T_1 and p_1 are the initial temperature and pressure conditions, and T_2 , p_2 the conditions in which the pressure wave is propagating. Taking $T_1 = 60^\circ\text{F.}$, $R = 55 \text{ ft.lb/lb.}/^\circ\text{R}$ for nitrogen and the length of the passage as 10 in., the values of time lag at various pressures are tabulated as follows:

$\frac{p_2}{p_1}$	Sonic velocity ft./sec.	Time lag msec.
1	1137	0.73
5	1430	0.58
10	1580	0.53
15	1672	0.50
20	1743	0.48
35	1890	0.44

Appendix H.

Operating procedure for the machine.

A typical operating procedure for the machine is as follows:

- 1) Switch on combustion cylinder jacket temperature control system and all electronic equipment, except for the optical displacement unit light source, at least one hour before the experiment so that the equipment will reach a steady working temperature.
- 2) Load film into drum camera.

- 3) Fit shear disc. This is carried out with the pistons in their lower position and a gas pressure of 25 to 50 psig in the drive gas container. This gas pressure serves to force the valve stem upwards and prevents it from dropping down whilst the shear disc is fitted.
- 4) Adjust electronic equipment for zero levels and correct amplification.
- 5) Set pendulum switch unit to its starting position.
- 6) Raise piston - by releasing gas pressure from drive gas container, close all valves on the damping gas exhaust passages and admit gas to the damping gas space at 8 psig pressure until the pistons reach top position.
- 7) Open valves to the exhaust silencers.
- 8) Set damping gas pressure by adjusting the pressure controller.
- 9) Close drive gas release valve.
- 10) Set non-return catch on top of release valve stem.
- 11) Fill compression cylinder chamber with the desired experimental mixture.
- 12) Fill drive gas container to a pressure which is 6.80 ± 0.05 times the absolute pressure of the damping gas pressure.
- 13) Attach falling weight to electro-magnet.
- 14) Switch on displacement unit light source.
- 15) Start drum camera and open shutter.
- 16) Close valve in the line to the damping gas pressure measuring gauge.
- 17) Press trigger button on time marker unit to record camera drum speed.
- 18) Release pendulum switch to operate the machine.
- 19) Stop gas supply to the damping gas space.
- 20) Switch off displacement unit light source.

Once the apparatus is at a steady working temperature, the total time for a complete run varies between 30 to 45 minutes, depending on the experimental mixture and consequently the method of filling the combustion cylinder.

Appendix I.Calculation of the adiabatic efficiency
of the machine when compressing air.

The ratio of the specific heat γ for air varies continuously with temperature and pressure, thus, in order to calculate the adiabatic efficiency when compressing air from temperature T_1 and pressure p_1 to a final theoretical condition of T'_2 and p'_2 , a value γ'_m which is the arithmetical mean of the ratios of specific heats at T_1, p_1 and T'_2, p'_2 was used. This involved several reiterative calculating processes before the correct γ'_m was found. The values of γ were taken from tables compiled by Hinselrath⁽¹³⁾.

The value of the apparent γ_m was derived from the measured pressures p_1, p_2 and the known compression ratio $\frac{V_1}{V_2}$, i.e.

$$p_2 = p_1 \left(\frac{V_1}{V_2} \right)^{\gamma_m}$$

and the final temperature was calculated from

$$T_2 = T_1 \left(\frac{V_1}{V_2} \right)^{\gamma'_m - 1}$$

Therefore, the adiabatic efficiency is

$$\frac{T_2}{T'_2} = \left(\frac{V_1}{V_2} \right)^{\gamma_m - \gamma'_m} \times 100\%.$$

Appendix J.Estimation of heat lost during compression.

It is assumed that, during the compression stroke, the rate of heat loss through the cylinder walls is proportional to the temperature difference between the gas and the walls and also to the surface/volume ratio

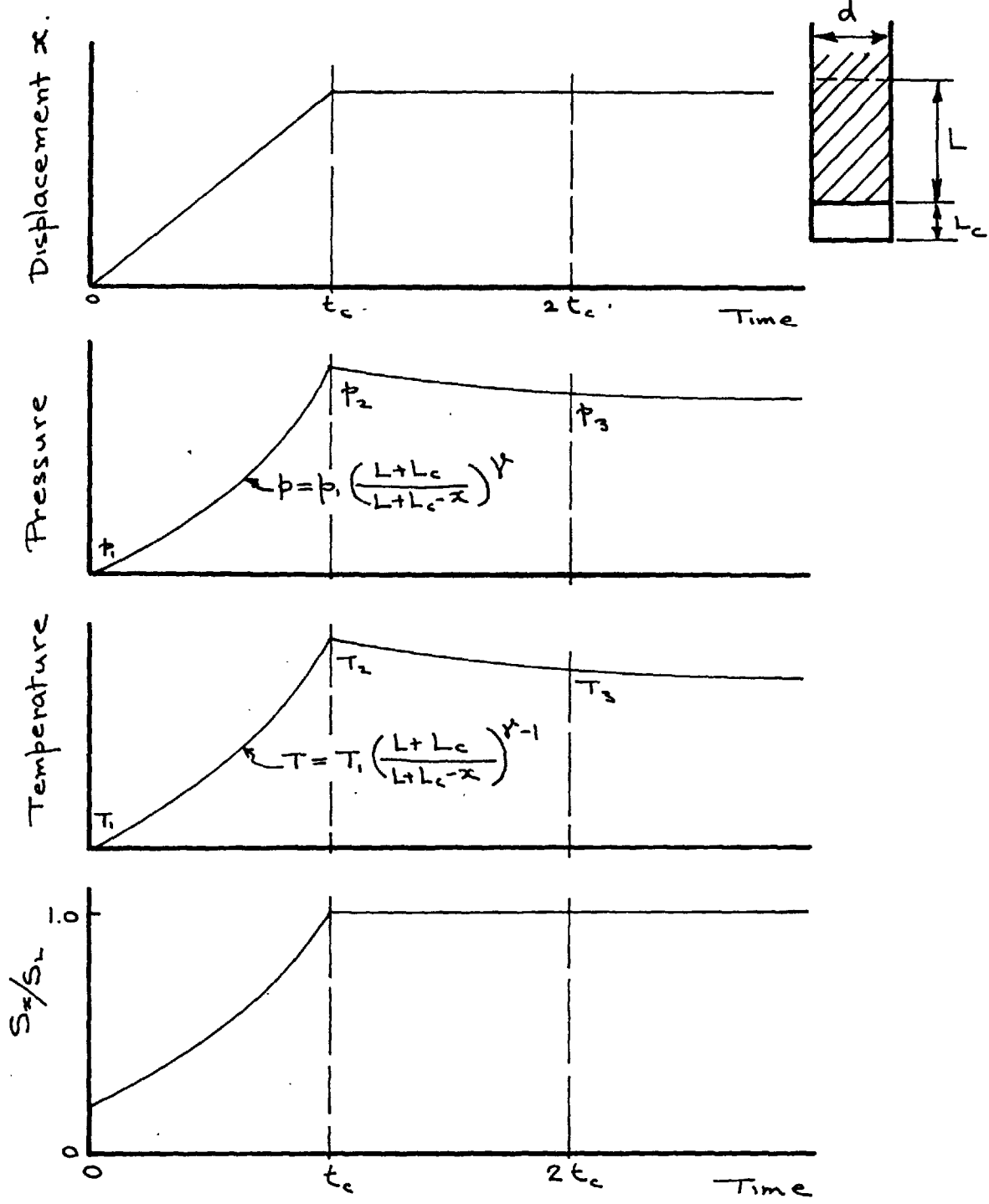


Fig. 11.4. Changes of conditions during and after compression.

of the contained gas. For convenience of calculation a uniform piston velocity was assumed.

Let L = length of piston stroke

L_c = " " clearance volume.

x = piston displacement.

t_c = compression time.

S_x, S_L = surface to volume ratio when the piston is at x and at the end of stroke respectively.

p = gas pressure.

T = temperature of the gas.

T_1 = initial temperature of the gas = wall temperature.

Provided there is no leakage for a fixed volume of gas,

$$\frac{dT}{T} = \frac{dp}{p}.$$

Therefore, the value $\frac{p_2 - p_3}{p_2}$ measured between a period from $t = t_e$ and $t = 2 t_c$, can be taken as the proportion of heat lost immediately after compression during the period t_c . It can be shown that, by allowing for the changes in temperature and surface to volume ratios during compression, the proportion of heat lost during compression is given by :

$$\frac{p_2 - p_3}{p_2} \int_{\frac{x}{L}=0}^{\frac{x}{L}=1} \frac{S_x}{S_L} \cdot \frac{T - T_1}{T_2 - T_1} d\left(\frac{x}{L}\right) \quad \dots (1).$$

From definition, it can also be derived that

$$\frac{S_x}{S_L} = \frac{4(L + L_c - x) + 2d}{L + L_c - x} \times \frac{L_c}{4L_c + 2d} \quad \dots (2).$$

$$\frac{T-T_1}{T_2-T_1} = \frac{\left(\frac{L+L_c}{L+L_c-x}\right)^\gamma - 1}{\left(\frac{L+L_c}{L_c}\right)^\gamma - 1} \quad \dots (3).$$

$$\text{and the compression ratio} = \frac{L + L_c}{L_c} \quad \dots (4).$$

The changes of conditions during compression are as shown in Fig.11.4. Equation (1) was integrated graphically; and the values of $\frac{p_2 - p_3}{p_2}$ were measured from actual recording of compressions of air in which the compression times were 12 msec. With an initial condition of 40°C., atmospheric pressure, and taking $\gamma = 1.35$, the calculated heat losses during compressions when the compression ratios were 11.5:1 and 31:1 are tabulated in Table III.

Table III : Heat loss during compression at various compression ratios.

C.R.	$\frac{p_2 - p_3}{p_2}$	Heat loss during compression. %
11.5:1	0.073	0.66
31:1	0.194	0.58

Appendix K.

Heat transfer across the cylinder walls.

For gas contained in a cylinder of fixed volume, the rate of heat loss through the walls is given by $\dot{q} = \frac{dT}{dt} \cdot C_v \cdot m = h \cdot \Delta T \cdot A$

where $\frac{dT}{dt}$ = rate of temperature drop in the gas.

C_v = spec. heat at constant volume of the gas.

m = mass of gas.

h = coefficient of heat transfer between the gas and the cylinder walls.

ΔT = the temperature difference between the gas and the cylinder walls.

$$\text{therefore } h = \frac{C_v \cdot m}{\Delta T \cdot A} \cdot \frac{dT}{dt}$$

The values of ΔT and $\frac{dT}{dt}$ were estimated from the pressure records of Runs No. 246, 247 and 251 in which air was compressed from initial conditions of $T_1 = 40^\circ\text{C}$. and $p_1 = 1 \text{ atm.}$. Assuming that for air $C_v = 0.19 \text{ CHU/}^\circ\text{C.lb.}$, the values of h given in Fig. 7.3. were calculated.

It is of interest to substitute the known conditions in the experiments into the equation for convective heat transfer across cylinder walls derived by Annand⁽¹⁾, which is

$$Nu = a (R_e)^{0.7}.$$

i.e.

$$h = a (R_e)^{0.7} \frac{k}{D}$$

where R_e is the Reynolds No. referring to the mean piston velocity V_p and piston diameter D ,

k is the thermal conductivity of the gas, and

a is a constant depending on the degree of turbulence.

The information obtained from the experiments mainly concerns the conditions after compression is completed, i.e. when the piston is at a standstill. However, if the conditions shortly after compression is completed, say 3 msec., are considered, it is to be expected that the gas motion in the cylinder will still bear a relationship to the piston velocity during compression. Hence, for the purposes of this calculation,

half the mean piston velocity during compression is taken as the piston velocity in the calculation of R_e .

$$\text{Assuming that } C_p = .224 + .471 \times 10^{-4} \times T \text{ CHU/lb.}^\circ\text{C.}$$

$$\mu = .316 \times 10^{-7} T^{0.645} \text{ lb/ft. sec.}$$

$$\text{and that } P_r = \frac{C_p \mu}{k} = 0.7$$

Then for air, it is found that for the conditions in Run No.247,

$$h = a \times 0.127 \frac{\text{CHU}}{^\circ\text{C.ft.}^2 \text{ sec.}}$$

On substituting the value previously found for h into the above expression, the value of a is found to be 0.34. Similarly, the values of a in other runs were calculated and shown in Table IV.

Table IV : Coefficients of heat transfer across cylinder walls
at various compression ratios, 3 msec. after completion
of compression stroke.

Run	C.R.	h		a
		$\left(\frac{\text{CHU}}{\text{sec.ft.}^2 \text{ } ^\circ\text{C.}} \right) \left(\frac{\text{Btu}}{\text{hr.ft.}^2 \text{ } ^\circ\text{F.}} \right)$		
246	9.6:1	0.009	32	0.07
247	17.2:1	0.043	155	0.34
251	29.8:1	0.105	378	0.83

From experimental observations in ^{motored} engines, Annand⁽¹⁾ found that the value of 'a' varies with the degree of turbulence, and it is interesting to note that the values of 0.34 and 0.83 found for Runs No.247 and 251 correspond to values given by Annand for conditions of 'low swirl' and 'high swirl' respectively. This implies that the degree of turbulence

varies with the compression ratios used in the machine, and that a higher degree of turbulence is found to occur with high compression ratios. Thus, the values of h are comparatively high when high compression ratios are used, as is the case with the adiabatic compression machine.

Appendix L.

Analysis of the hydrocarbons.

The n-heptane and n-decane fuels were supplied by Shell Research Ltd.; according to ^{tests by} infrared mass spectography and gas liquid chromatography, both liquids are 99% pure. The refractive indices of the liquids are 1.385 for the n-heptane and 1.410 for the n-decane measured at 25°C.

Appendix M.

Calculation of vapour mixture strength.

The calculation was based on a knowledge of partial pressures of the fuel and air.

Let p_f = partial pressure of fuel

p_a = " " " air

p = total pressure

V = vessel volume

V_{inj} = volume of injected fuel liquid.

T = absolute temperature.

R = universal gas constant.

m_f = molecular weight of fuel.

m_a = " " " air.

w_f = weight of fuel.

w_a = " " air.

$$\text{then } p_f = \frac{w_f}{m_f} \frac{RT}{V} \quad \dots (1)$$

$$\text{and } p_a = \frac{w_a}{m_a} \frac{RT}{V}$$

$$\text{giving } \frac{w_f}{w_a} = \frac{p_f}{p_a} \times \frac{m_f}{m_a} \quad \dots (2)$$

$$\text{but } p = p_a + p_f \quad \dots (3)$$

Substituting (3) into (1) :

$$\frac{w_f}{w_a} = \frac{p_f}{(p - p_f)} \frac{m_f}{m_a} \quad \dots (4)$$

The vessel volume $V = 483.8$ ml.

and for n-heptane $m_f = 100$

$$\text{S.G.} = 0.683$$

Substituting these values in (1) and (4) the final expressions for mixture strength of n-heptane/air mixtures are :

$$\frac{w_f}{w_a} = 3.45 \frac{p_f}{p - p_f}$$

$$\text{and } p_f = 0.880 V_{inj} T.$$

in which the pressures are measured in mm.Hg., V_{inj} in ml. and T in $^{\circ}\text{K}$.

Appendix N.Prediction of ignition delay .

Livingood and Wu⁽²⁸⁾ predicted ignition delay periods by assuming that the reaction rate is dependent on parameters having the following general form:

$$\frac{d(x)}{dt} = \phi_1 (p, T, t), \phi_2 (F, \text{Chemical Composition})$$

where (x) = concentration of pertinent reaction products

t = time

p = absolute pressure

T = absolute temperature

ϕ_1, ϕ_2 = empirical functions

F = fuel/air ratio.

This implies that, for a given fuel/air ratio, there exists a fixed functional relationship between the rate of the aggregate reaction, the instantaneous physical state and time. Under certain physical conditions, a low speed reaction takes place to build up the concentration of chain carriers. When a critical concentration x_c is reached, the reaction changes to a high speed explosion. Before this state is reached, the relation between reaction rate and delay time τ is expressed as

$$\frac{d}{dt} \left(\frac{x}{x_c} \right) = \phi \left(\frac{t}{\tau} \right) .$$

Although function ϕ is not known, it is assumed that the reaction rate does not change with time during a fixed state process, i.e.

$$\phi \left(\frac{t}{\tau} \right) = \frac{1}{\tau}$$

$$\text{therefore } \frac{x}{x_c} = \int_{t=0}^{t=t_e} \frac{1}{\tau} dt = 1.0$$

where t_e is the overall reaction time.

If, during compression of a fuel/air mixture, the temperature and pressure histories are known then, by using ignition delay data τ determined by a rapid compression machine, the value t_e and the particular delay period after compression can be determined by graphical integration.

In applying this theory to demonstrate the effect of different rates of compression on delay periods, it was assumed that the piston speed was constant; the ignition delay data for stoichiometric n-heptane/air mixture was taken from the results of the M.I.T. rapid compression machine⁽²⁸⁾. The accuracy of this data is not critical for this application, since it is merely used as a datum line for the calculation; the object of the calculation being to find the differences in the predicted delay times for different compression times rather than the actual delay periods.

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