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# PUMPLESS DIESEL INJECTION SYSTEMS

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ABSTRACT

After reviewing earlier pumpless injection systems for diesel engines, the report presents an account of the investigations on such systems at the Internal Combustion Engineering Department, Indian Institute of Science, Bangalore. From the results obtained it is apparent that these systems fail to meet simultaneously the demands of proper injection timing and fuel atomisation to ensure satisfactory performance of a diesel engine. Some exploratory work for applying this principle to petrol engines is also reported.

## INTRODUCTION

Compression ignition engines of all sizes and speeds have their fuel system problems on account of the exacting requirements and the technical difficulties of manufacture of high pressure injection systems. These difficulties are accentuated in small high speed engines and the relative cost of the injection equipment becomes disproportionately high. A possible solution to this seems to lie in devising a simpler injection system. The successful development of such a system would place compression ignition engines in a more competitive position *vis a vis* the spark ignition type in the field of small displacement internal combustion engines.

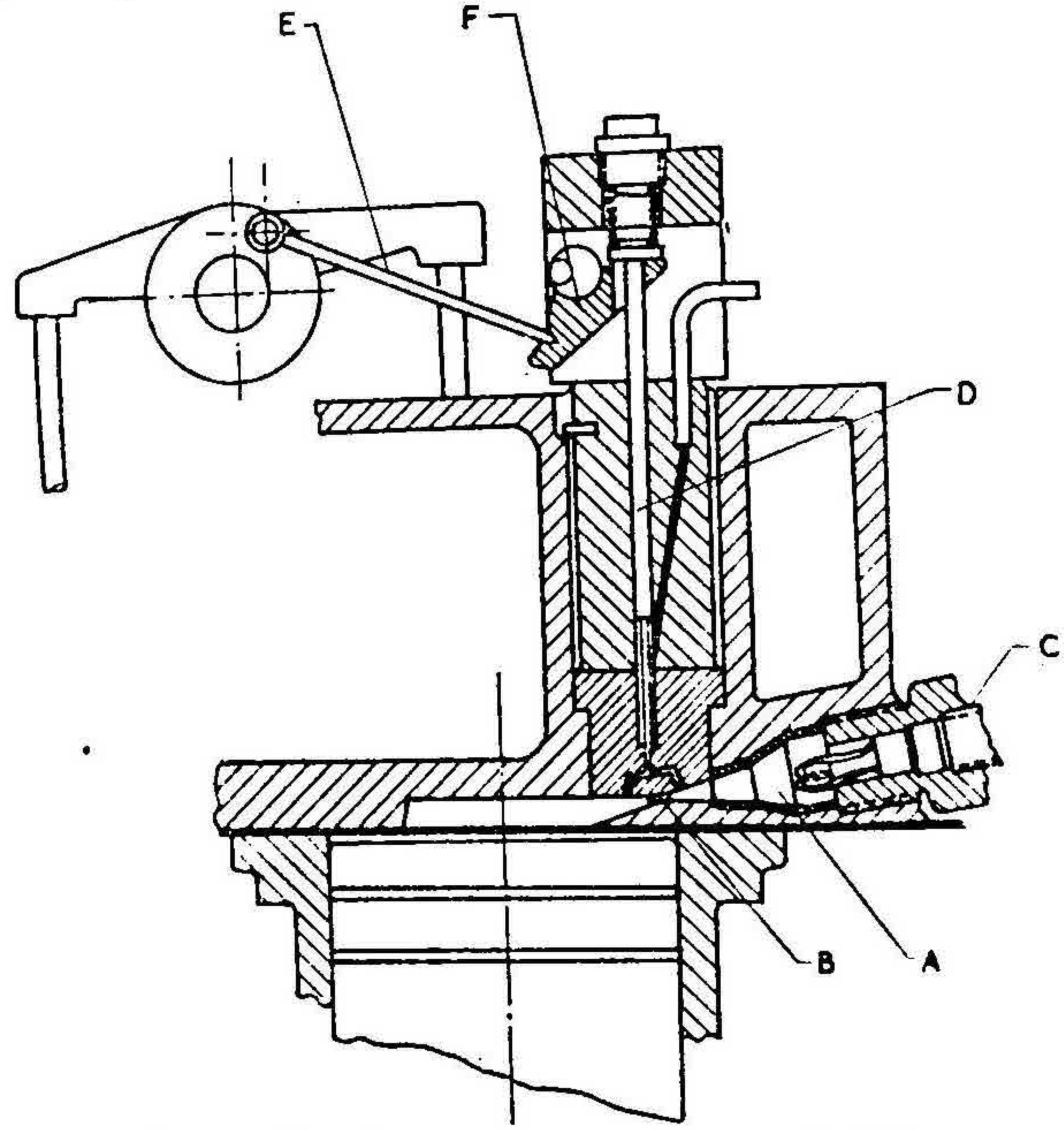
Several systems, generally referred to as 'Pumpless Injection Systems' have been tried by various investigators towards this end. The principle employed is the introduction of the fuel into an injector suitably located and the creation at the proper moment of a pressure difference between the engine cylinder and an auxilliary chamber to effect injection of fuel. The features distinguishing one system from another are the general configuration of the combustion chambers and the methods used to provide for fuel metering and the timing of injection.

## PUMPLESS INJECTION SYSTEM

Pumpless injection systems tried out by L'orange, Chapman and Witzky are shown in Figs. 1, 2, 3 & 4.

L'orange carried out his experiments on a single cylinder four stroke engine of 300 cc (Fig. 1) and a two stroke engine of 100 cc displacement (Fig. 2). The two stroke version used a lightly loaded non-return valve for admitting fuel. The 300 cc engine had a compression ratio of 19 : 1, a normal output of 4 b.h.p. at 2400 r.p.m. corresponding to a b.m.e.p. of 73 psi., and a maximum output of 5 b.h.p. at 3000 r.p.m. A specific fuel consumption of 0.5 to 0.55 lb/b.h.p-hr was claimed for the unit. The 100 cc unit could be run up to 5000 r.p.m.

In tests on a four stroke engine, Chapman obtained the following results: Specific fuel consumption of 0.6 lb/b.h.p-hr at 1500 r.p.m. with b.m.e.p. ranging from 40 – 80 psi. Metering and combustion were reported to be good at speeds up to 3000 r.p.m.




A : Pre-Chamber
B : Venturi Throat
C : Glow Plug
D : Needle Valve
E : Lever
F : Rocker Arm

FIG. 1

L'orange Injection System, Four-Stroke Version

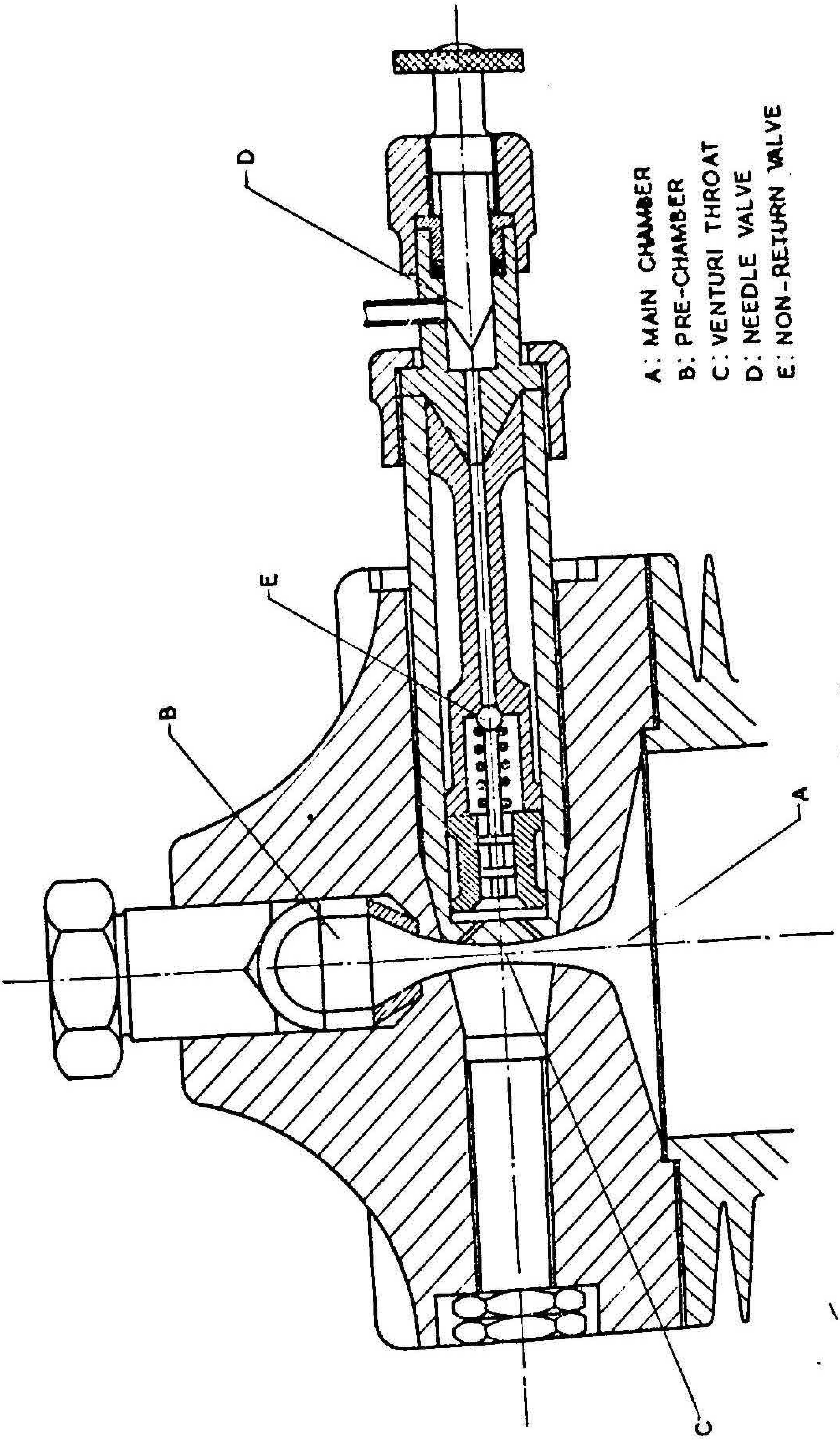
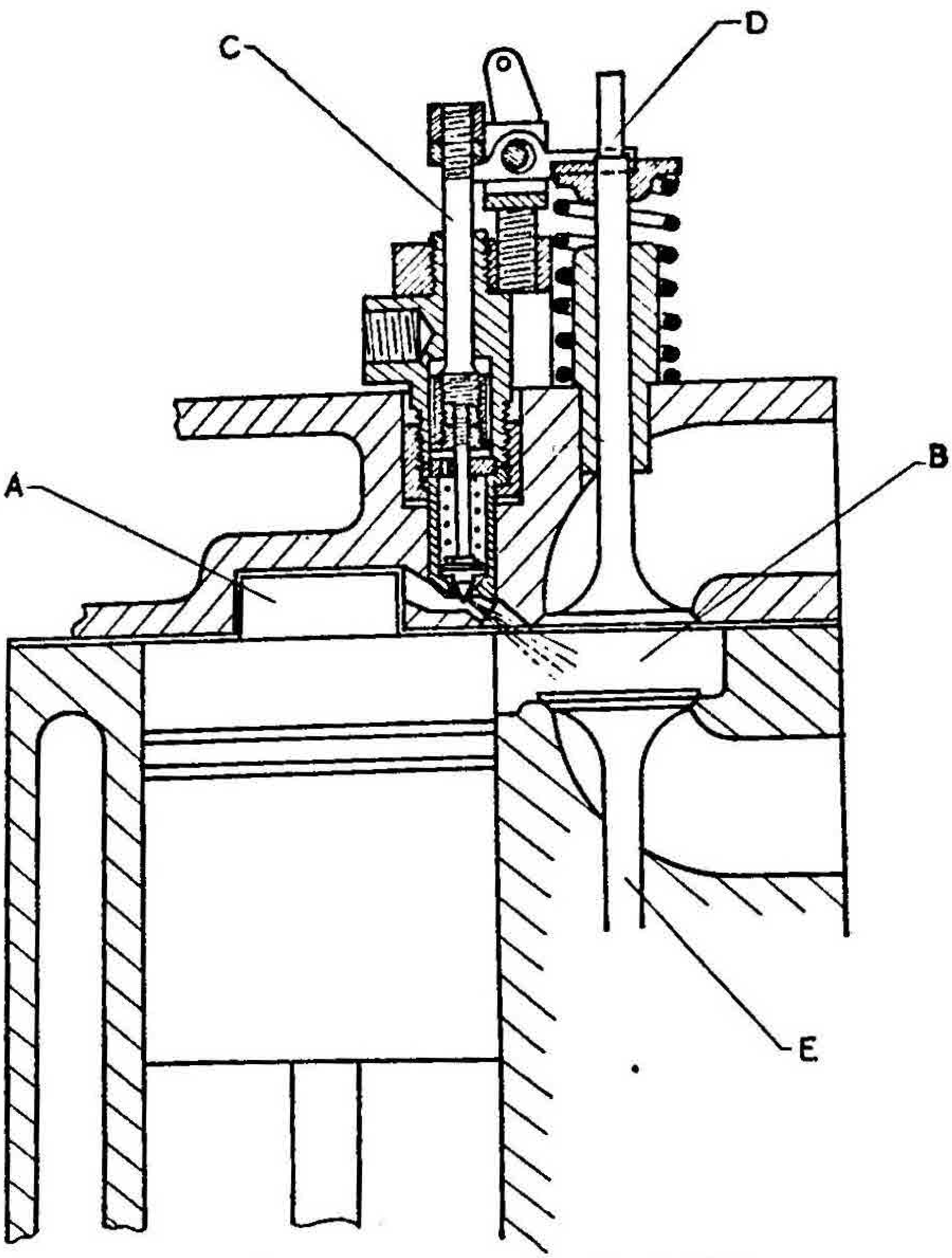



FIG. 2

L'orange Injection System, Two Stroke Version




A: Piston Plug
B: Side Combustion
C: Fuel Valve
D: Intake Valve
E: Exhaust Valve

FIG. 3
Chapman Injection System

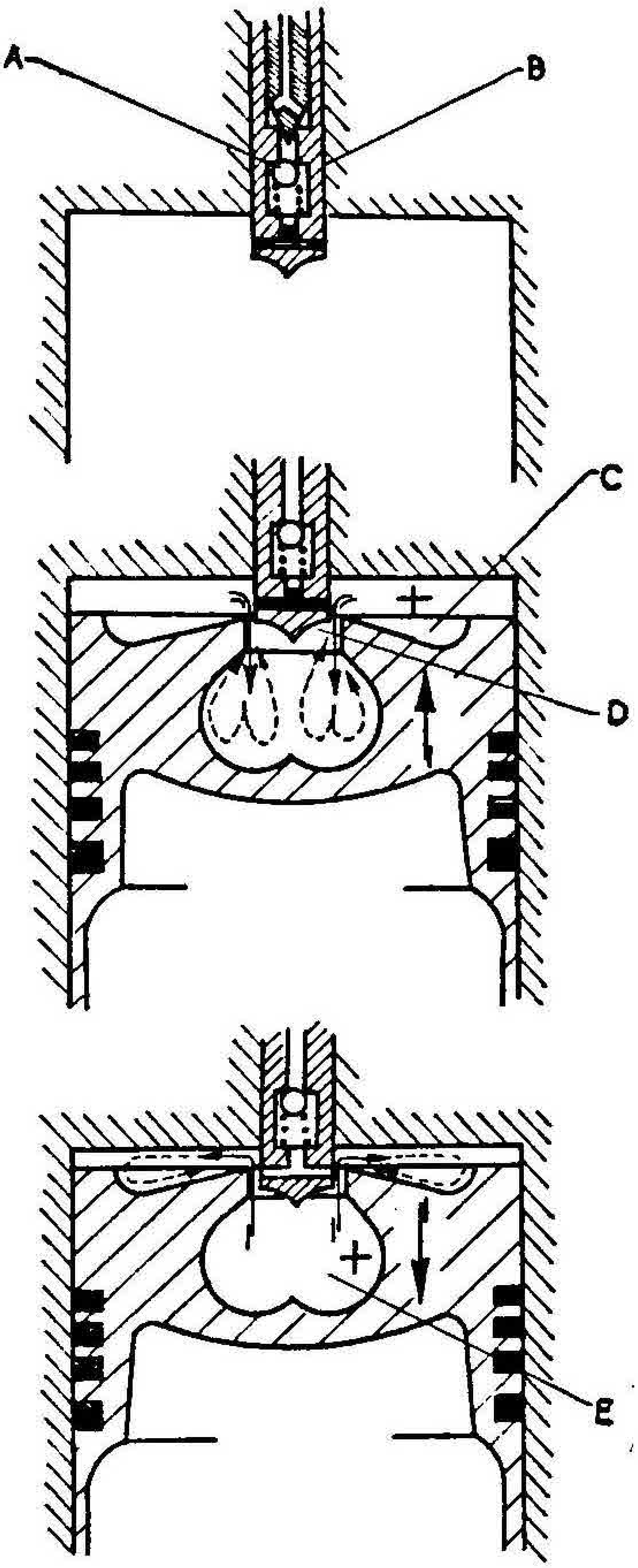
Witzky tried his system on a modified C.F.R. diesel test engine of 3.25 in. bore and 4.5 in. stroke and the results obtained were as follows : Low rates of pressure rise and low peak pressure, indicated specific fuel consumption of 0.38 lb/i.h.p-hr and a maximum i.m.e.p. of 120 psi.

The feasibility of the Witzky system was further investigated by Hussman *et al*[1]. A two stroke cycle air cooled gasoline engine of 2.5 in. bore and 2 in. stroke was converted to operate on the Witzky system. The functioning of

After Intake Stroke or Scavenging

Compression Stroke: Injection into Piston Chamber Starts. Ignition in Piston Chamber

Expansion Stroke: Injection Continues into Main Combustion Chamber




A: Check Valve
B: Nozzle
C: Main Chamber
D: Throat in Piston Chamber
E: Piston Chamber

FIG. 4

Witzky Injection System

the nozzle was studied under simulated conditions and a number of nozzle designs were tried out on the engine. The maximum power output of the engine was found to be seriously limited by early ignition with increasing load. Controlling the angle of ignition by modifying the nozzle design and piston configuration was reported to be unsuccessful.

In recent years experiments have been carried out on pumpless injection systems by Narayanaswamy[2] at the I.C.E. Department of the Indian Institute of Science, Bangalore. The systems which were investigated are variations of two basic types. The first one, similar to that of L'orange, was tried on a two stroke diesel engine of 4.0 in. bore and 4.0 in. stroke with a rated speed of 600 r.p.m. The cylinder head of the test engine was modified to incorporate the L'orange type pressure differential device (Figs. 5 & 6). The design of the cylinder head was such that it was possible to alter the geometry of the system by fitting interchangeable inserts for the venturi and prechamber. The effect of varying the venturi throat diameter and the volume of the prechamber relative to the total clearance volume could thus be studied while maintaining the overall compression ratio more or less constant.

In addition, designs in which the air and fuel passages were variously disposed in relation to each other were also tried out.

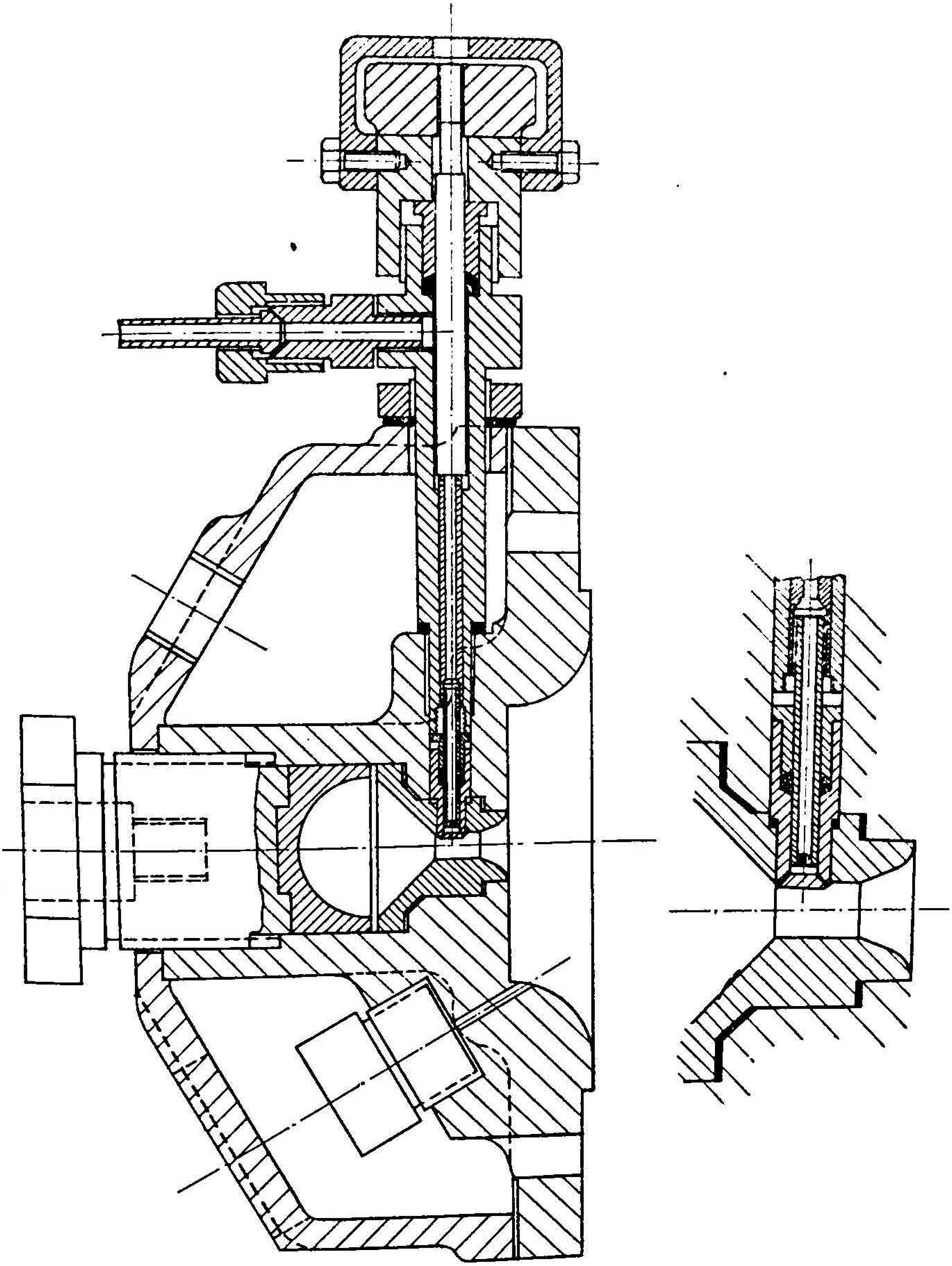
It is apparent from the geometry of the system that the pressure difference between the cylinder and prechamber would commence building up almost as soon as effective compression starts in the cylinder. This would lead to early injection of fuel and a coarse spray. Consequently the test engine showed rather poor performance.

To gain some control over the injection timing the second type of system incorporating a displacer type piston which would delay the build up of differential pressure till the desired moment was investigated on a test engine of bore 4.00 in. and stroke 5.625 in. running at 980 r.p.m. Fig. 7 shows the cross section of the cylinder head which was used for these experiments. This system was found to be superior to the first one and the engine performance was relatively better.

## CONCLUSIONS FROM REVIEW OF PUMPLESS INJECTION SYSTEMS

In the L'orange system there is no control over the point of pressure build up and hence on fuel injeetion timing. The systems tried out by Chapman, Witzky and Narayanaswamy appear to be better in this respect.

The basic principle of injection of all the above systems is substantially similar to the conventional airblast system. There are however certain differences in the actual working. In the conventional airblast system the quantity and pressure of air used for injection can be controlled independently of the engine. Usually the amount of air used for injecting the fuel is about 5 to 8% by weight of cylinder charge and the pressure of the air is about 800 to 1000 p.s.i. In a pumpless injection system, the geometry of the pressure differential device plays the major role in controlling these factors.



DETAILS OF RETRACTABLE TIP

FIG. 5

Modified fuel System, QBM Engine

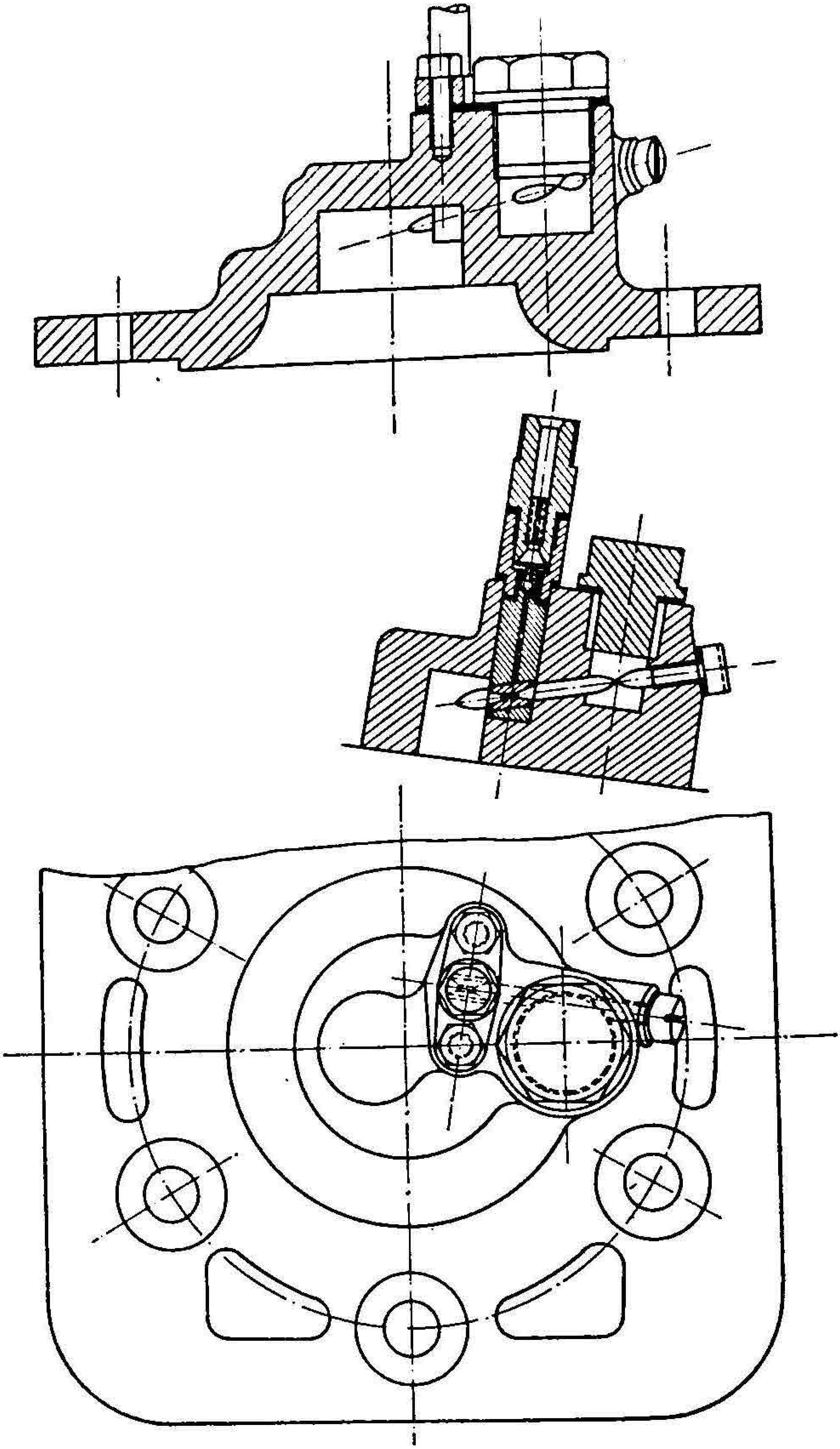
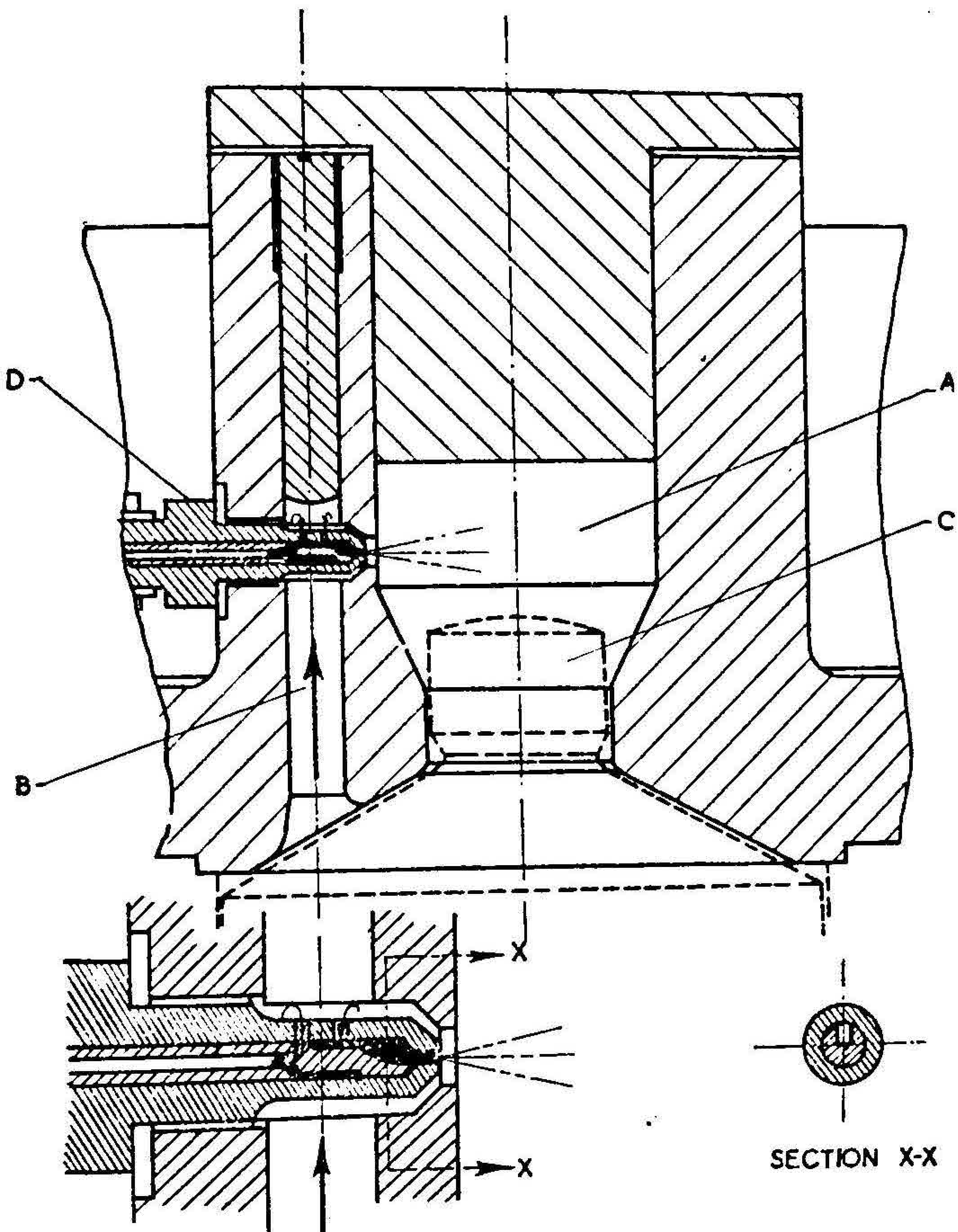


FIG. 6
QBM Cylinder Head No. 2




A : Main Chamber
B : Air Passage
C : Knob
D : Nozzle

FIG. 7
Displacer Type System

It is therefore necessary first, to establish the minimum airblast requirements for achieving satisfactory atomisation of the fuel and secondly, to estimate the pressure and quantity of air that can actually be made available by such a device.

In order to establish the minimum airblast requirements the relative effects of the various factors on spray characteristics were investigated. As already reported[3], this investigation showed that for satisfactory atomisation of the fuel the air velocity should be around 700 f.p.s. and the flow ratio (the ratio of volume of air used for atomisation to volume of fuel) 2000 – 3000 corresponding a mass ratio of 2.4 to 3.6. For effective utilisation of the air delivered for atomising the fuel it was necessary to design the nozzle in such a way as to ensure good intermingling between fuel and air at the point where air to fuel relative velocity is maximum.

Further experiments were carried out on a test engine to determine the air flow conditions (velocity and the quantity) at the nozzle and the efforts were made to improve the performance of the system by modifying the design. The following is an account of these tests.

## TEST ENGINE AND INSTRUMENTATION

The test engine was a Textool two-stroke diesel engine rated at 5 HP at 700 r.p.m. and 7 HP at 980 r.p.m.

The cylinder head and piston of the engine were modified for operation on pumpless fuel injection. Fig. 8 is a cross section of the modified cylinder head and piston. From the instant the knob fixed on top of the piston crown enters the throat, the compression in the cylinder proceeds at a more rapid rate than in the combustion chamber. A pressure difference is thereby generated between the two and air flows from the cylinder into the combustion chamber through the nozzle and the clearance between the knob and throat. In addition to conventional instrumentation for measuring engine variables such as power output, speed and operating temperatures, a standard Sunbury engine indicator with an electromagnetic pressure pick up was used to study the pressure-time phenomena inside the combustion chamber and cylinder. Fig. 9 is a photograph of the complete test-rig.

## EXPERIMENTAL PROGRAMME

This consisted mainly of two parts:—(*a*) Establishing the flow conditions through the nozzle in the test engine and examining if the optimum airblast requirements for satisfactory atomisation of the fuel are met and (*b*) In the light of the results obtained from (*a*) to conduct further investigations on the engine to improve the performance of the system.

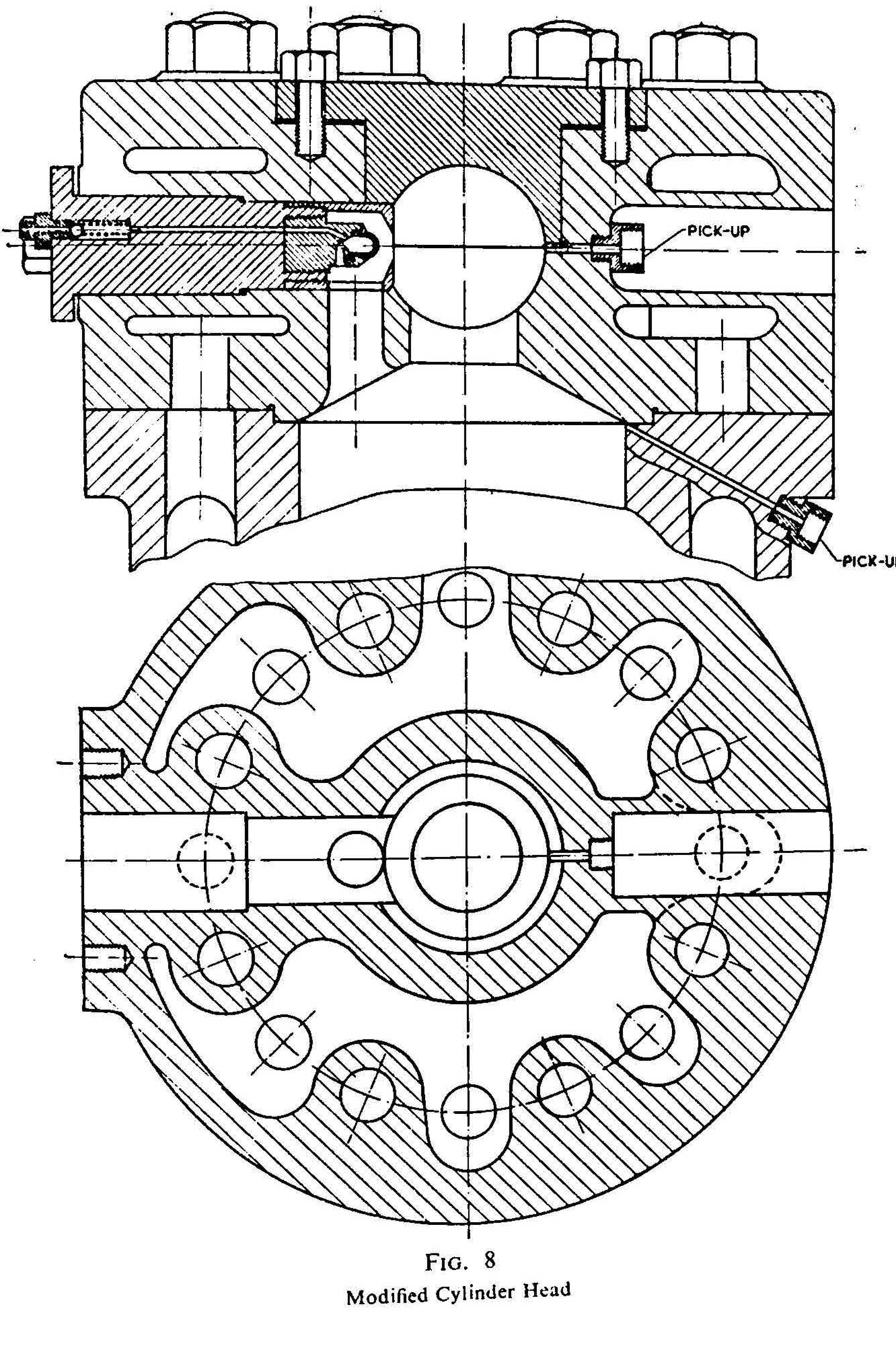



FIG. 8
Modified Cylinder Head

## DETERMINATION OF VELOCITY AND MASS FLOW OF AIR THROUGH THE NOZZLE IN THE TEST ENGINE

On account of the close interdependence of the conditions which govern the transfer of air from the cylinder to the combustion chamber, the determination of the velocity and mass flow of air through the nozzle by theoretical means is laborious and requires many simplifying assumptions. The following experimental method was therefore adopted. The engine was motored at 675, 775 and 1175 r.p.m. and at each speed the nozzle diameter was varied from 0.125 in. to 0.25 in. and pressure time diagrams were taken for the combustion chamber and cylinder. The velocity and mass flow of air through the nozzle were calculated from the pressure-time diagrams as follows:

Assuming isentropic flow, the flow velocity at any instant is given by

$$U = C_v \sqrt{\left[2\,\frac{k}{k-1}\cdot R\,T_c\,g\left\{1-\left(\frac{P_{cc}}{P_c}\right)^{k-1/k}\right\}\right]} \qquad [1]$$

where $C_v$ is the velocity coefficient. Subscripts $c$ and $cc$ refer to conditions in the cylender and combustion chamber respectively.

The equations governing the pressure time relations for divided chambers have been theoretically derived by Hussman *et al*[1] by applying the laws of mass and energy transfer between the two chambers. The relation between the pressure in the two chambers is given by the following two simultaneous differential equations:

$$k\,P_c\,d(V_c) + V_c\,d(P_c) + AC_1\left(\frac{P_{cc}}{P_c}\right)^{\frac{3k-1}{2k}}\sqrt{\left[\left(\frac{P_{cc}}{P_c}\right)^{2/k} - \left(\frac{P_{cc}}{P_c}\right)^{\frac{k+1}{k}}\right]}\cdot dt = 0 \qquad [2]$$

and

$$k\,P_{cc}\,d(V_{cc}) + V_{cc}\,d(P_{cc}) = -k\,P_c\,d(V_c) - V_c\,d(P_c) \qquad [3]$$

where

$$C_1 = \sqrt{\left[\frac{2gk^3\,(P_{c_1})^3}{k-1}\,v_{c_1}\right]}$$

$A$ is the area of the nozzle and subscript 1 refers to initial conditions. One of the assumptions made by the authors in arriving at these equations was to assume that the cylinder gas is compressed isentropically. However, if the compression in the cylinder is assumed to follow a polytropic law equations [2] and [3] become

$$n\,P_c\,d(V_c) + V_c\,d(P_c) + nA\sqrt{\left[\frac{2gv_{c_1}\,k\,(P_c)^{[3(n-1)]/n}\,P_{cc}^{2/k}}{(k-1)(P_{c_1})^{2/k-3/n}}\left\{1-\left(\frac{P_{cc}}{P_c}\right)^{(k-1)/k}\right\}\right]}\,dt = 0 \qquad [4]$$

and

$$k\,P_{cc}\,d(V_{cc}) + V_{cc}\,d(P_{cc}) = -k\,P_c\,d(V_c) - (k/n)\,V_c\,d(V_c) \qquad [5]$$

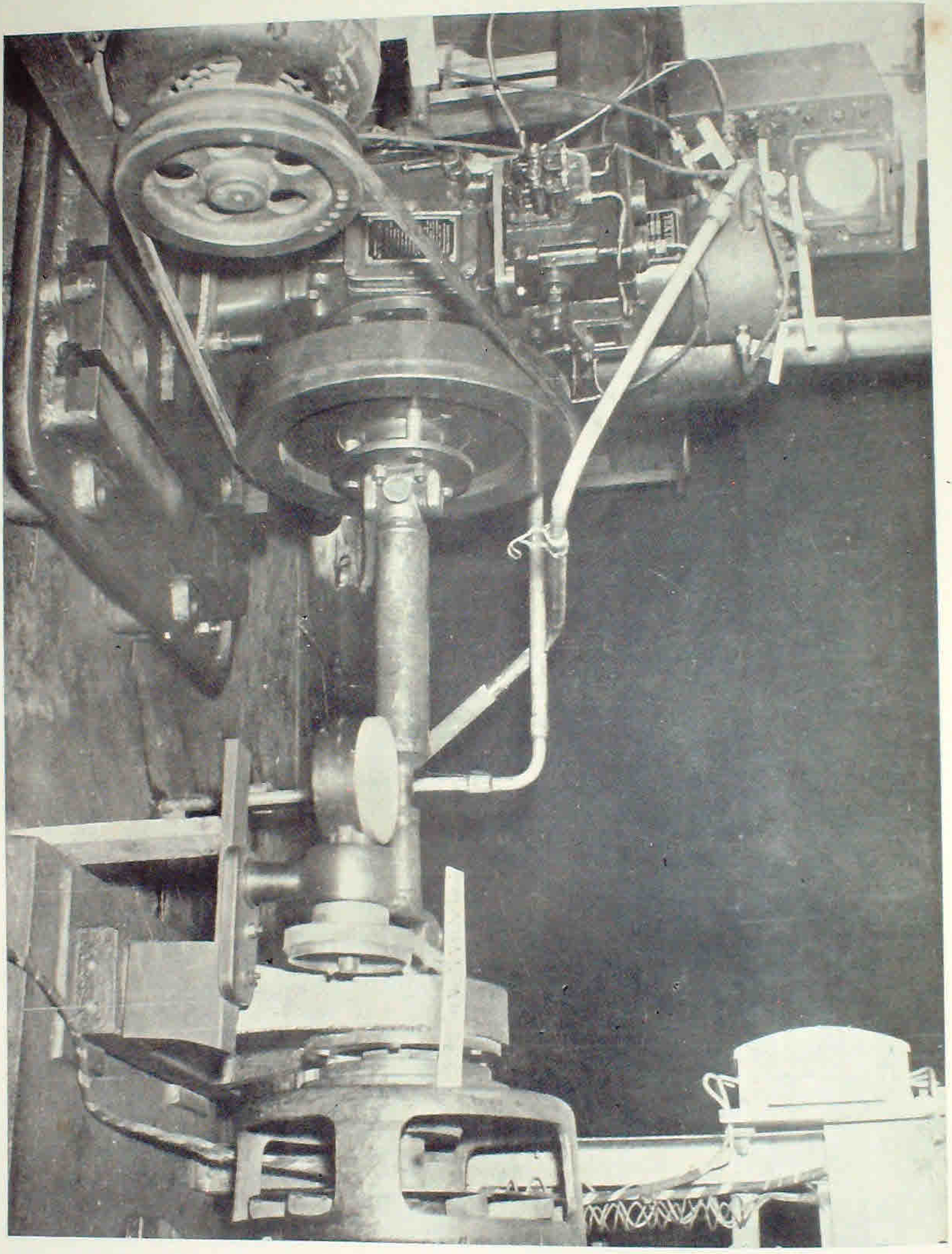


FIG. 9

The pressure and volume in the two chambers at any instant during compression being known, the index of compression '$n$' for the cylinder may be determined from equation 5.

Now assuming the initial charge temperature as 50°C and an average value of 0.99 for $C_v$ the air velocity at any instant may be calculated from Equation (1).

The mass flow through the nozzle in tione $\Delta t$ is $M = C_d A U \rho . \Delta t$

where $C_d$ is the coefficient of discharge for the nozzle and $\zeta$ is the density of the charge.

The velocity through the nozzle under engine conditions is both varying and intermittent. Further the pressure and temperature at the nozzle are also varying. The coefficient of discharge under such conditions is not likely to remain constant. So in order to obtain the probable value of the coefficient of discharge at any instant the nozzle was calibrated over a range of flows under atmospheric conditions. The coefficient of discharge of the nozzle was evaluated and plotted as a function of Reynolds number (Fig. 10).

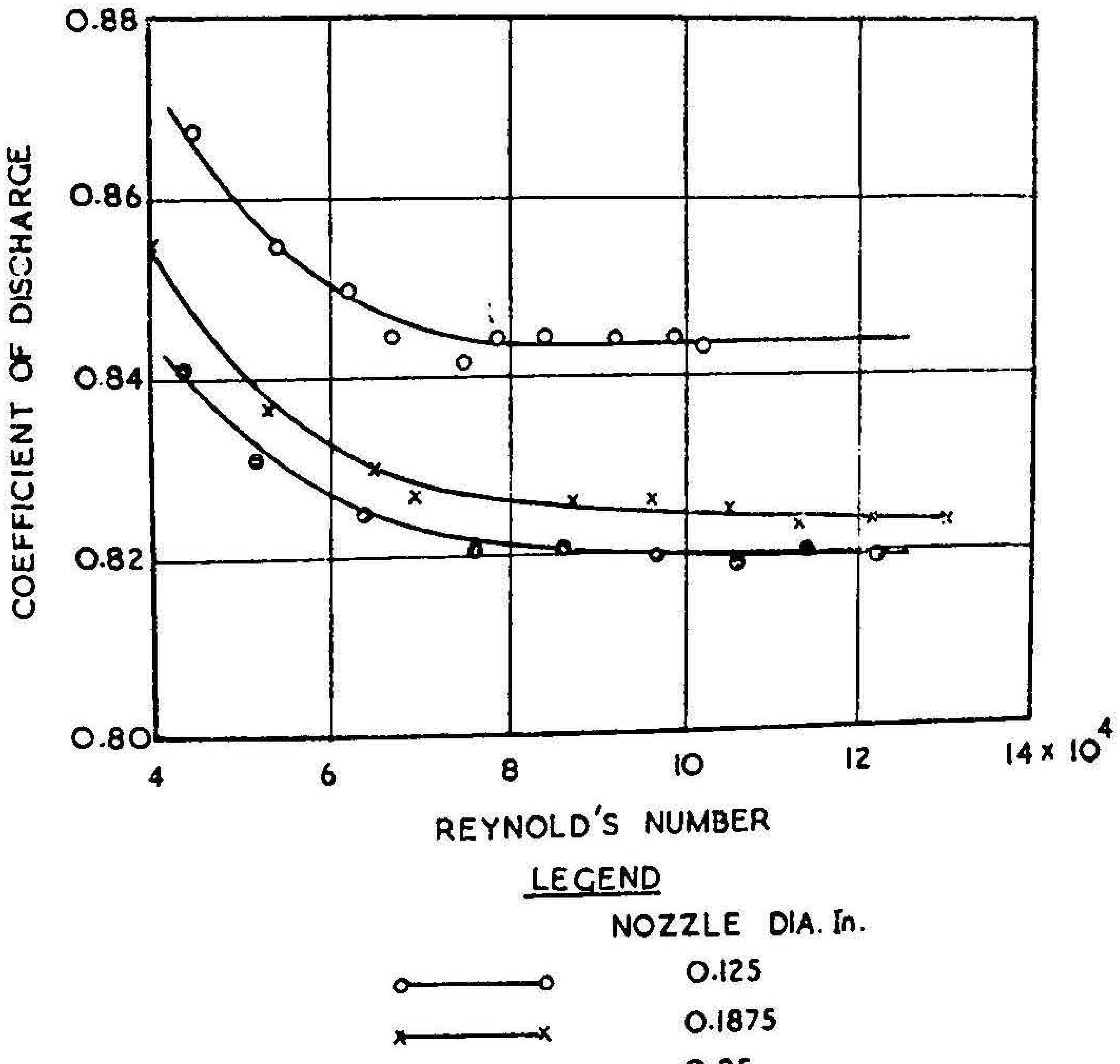


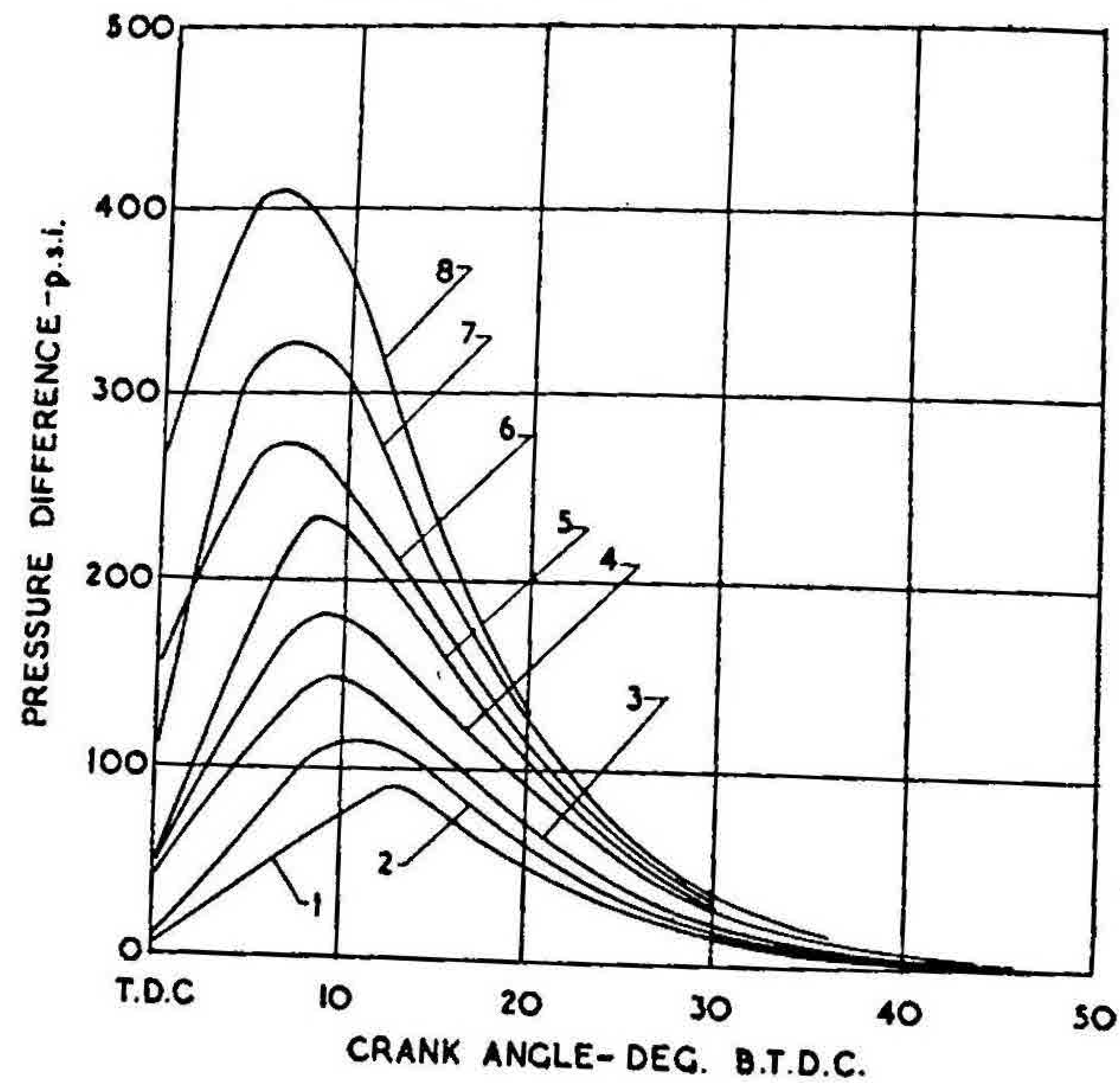

FIG. 10

Coefficient of Discharge. Vs Reynold's Number for the Test Nozzles

The probable coefficient of discharge under actual engine conditions was obtained from Fig. 10 by first calculating the Reynolds number for flow through the nozzle during any particular interval. The mass flow through the nozzle was then calculated from equation [6], by a step by step method considering small intervals of time and taking averagevalues of U, $\rho$ and C during each interval.

The results of the analysis of the pressure-time diagrams for pressure difierence, air velocity and air mass flow through the nozzle are given in Figs. 11, 12 and 13.

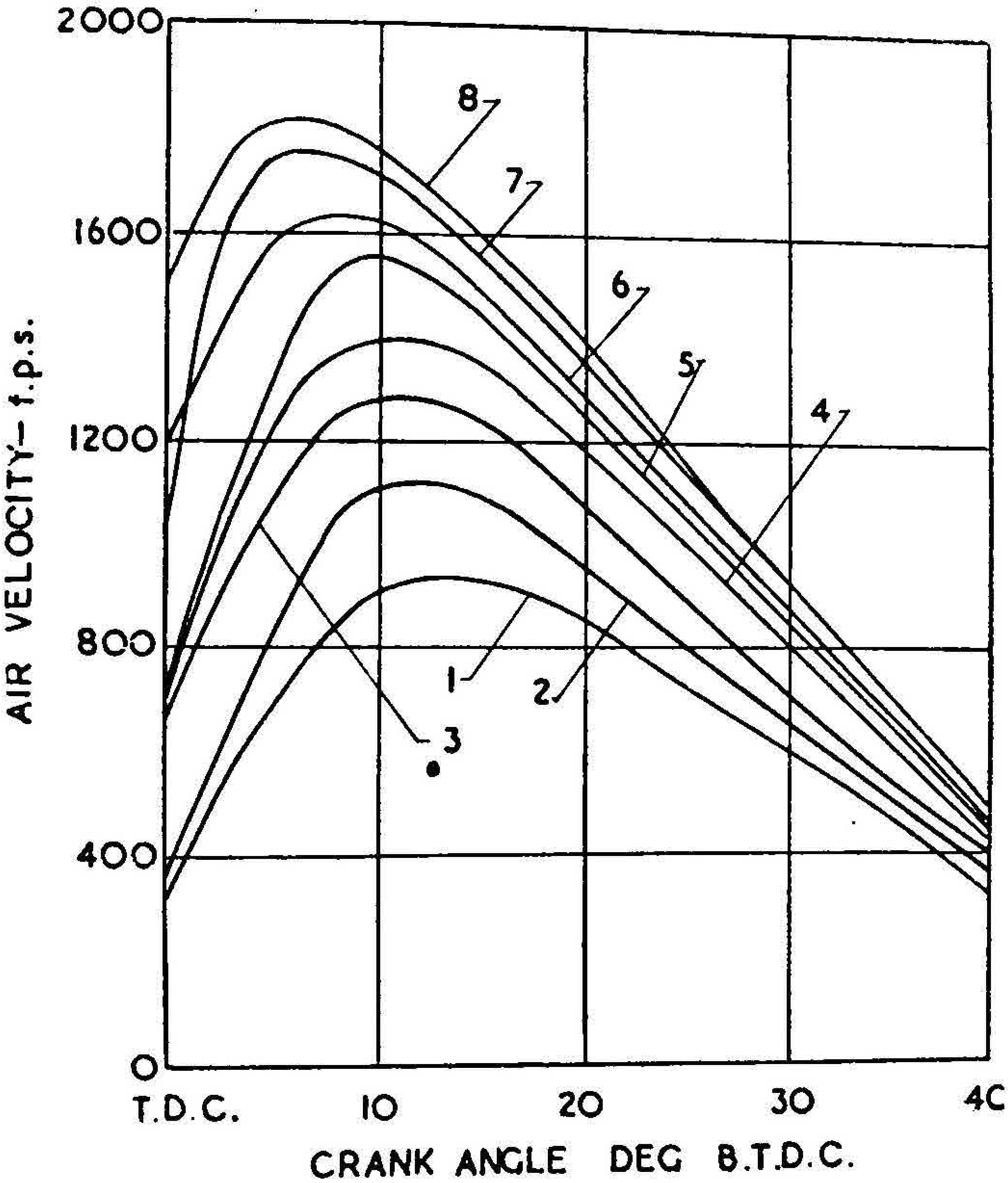
Angle of knob entry—46° B.T.D.C.
Compression Ratio—16.5 : 1




| No. | Nozzle Dia | Engine Speed |
|---|---|---|
| 1 | 0.25" | 675 R.P.M. |
| 2 | 0.25" | 775 |
| 3 | 0.25" | 1175 |
| 4 | 0.1875" | 675 |
| 5 | 0.1875" | 775 |
| 6 | 0.125" | 675 |
| 7 | 0.1875" | 1175 |
| 8 | 0.125" | 775 |

FIG. 11
Pressure Difference Between Cylinder and Combustion Chamber

Angle of knob entry: 46° B.T.D.C.
Compression Ratio: 16.4 : 1

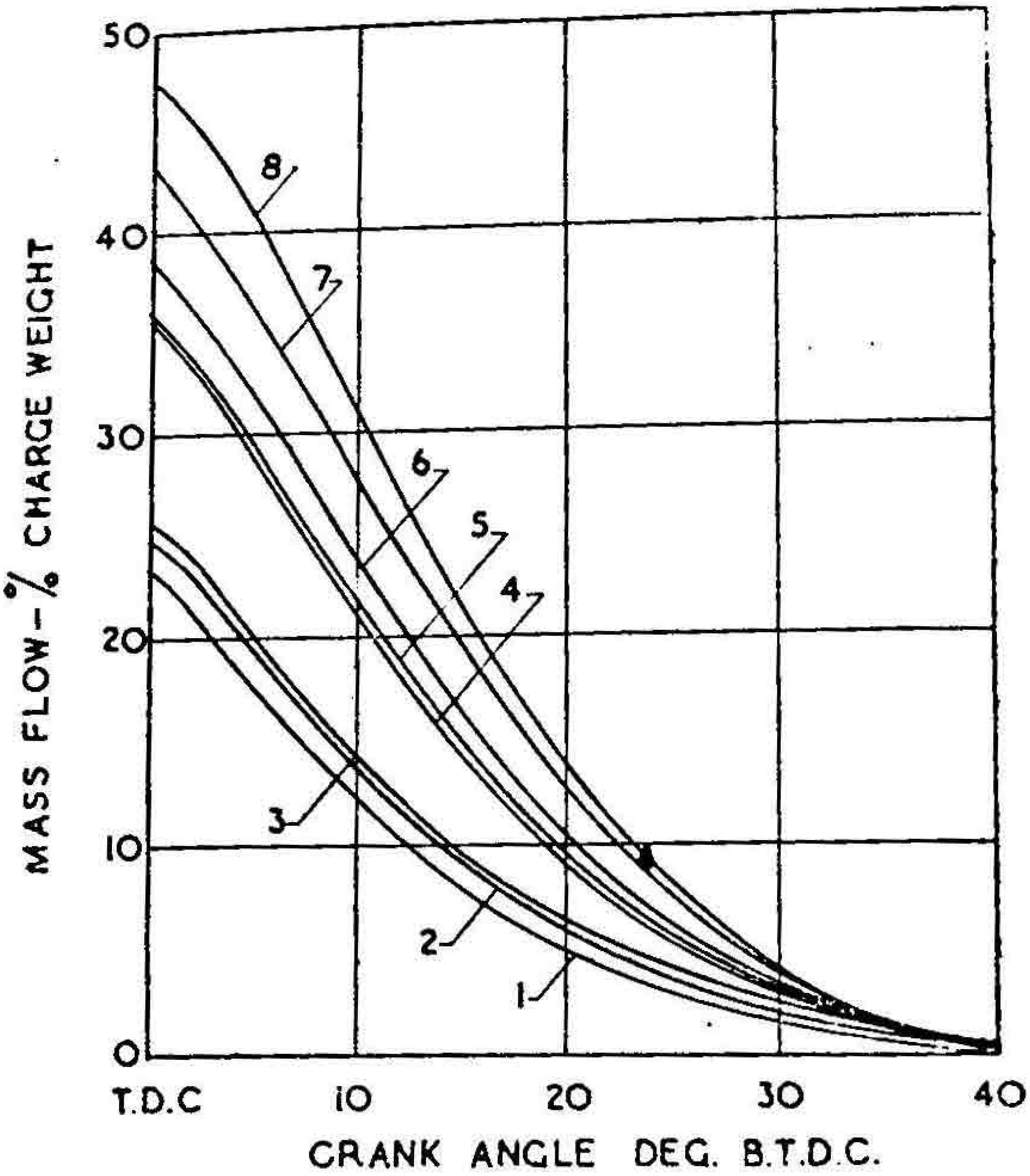



| No. | Nozzle dia. | Engine speed |
|---|---|---|
| 1 | 0.25" | 675 RPM |
| 2 | 0.25 | 775 |
| 3 | 0.25 | 1175 |
| 4 | 0.1875 | 675 |
| 5 | 0.1875 | 775 |
| 6 | 0.1875 | 1175 |
| 7 | 0.125 | 875 |
| 8 | 0.125 | 775 |

FIG. 12

Velocity of air through nozzle

Angle of knob entry: 46° B.T.D.C.
Compression Ratio: 16.4 : 1




| No. | Nozzle dia. | Engine speed |
|---|---|---|
| 1 | 0.125″ | 775 R.P.M. |
| 2 | 0.125 | 675 |
| 3 | 0.1875 | 1175 |
| 4 | 0.25 | 1175 |
| 5 | 0.1875 | 775 |
| 6 | 0.1875 | 675 |
| 7 | 0.25 | 775 |
| 8 | 0.25 | 675 |

FIG. 13

Mass Flow of Air through nozzle

It is observed that the velocity and mass flow of air through the nozzle increase rapidly from 30° BTDC. Under actual engine operating conditions the flow would practically cease soon after combustion starts in the prechamber. Assuming then, the period during which the air flow is effective in atomising the fuel is from 30° to 5° BTDC, the mass of air made available for injection with a nozzle of 0.25 in. dia. at an engine speed of 775 r.p.m. is 31% of the cylinder charge. This amounts to about 11 times the weight of the fuel under full load conditions for the test engine. The air velocity during the period varies from 700 to 1100 f.p.s. Thus the mass and velocity of air made available appear to be sufficient for satisfactory atomisation of the fuel within the engine cylinder.

## Engine Tests

Three main types of nozzle designs shown in Fig. 14 were tried out on the engine. The nozzle diameter for all the three types was fixed at

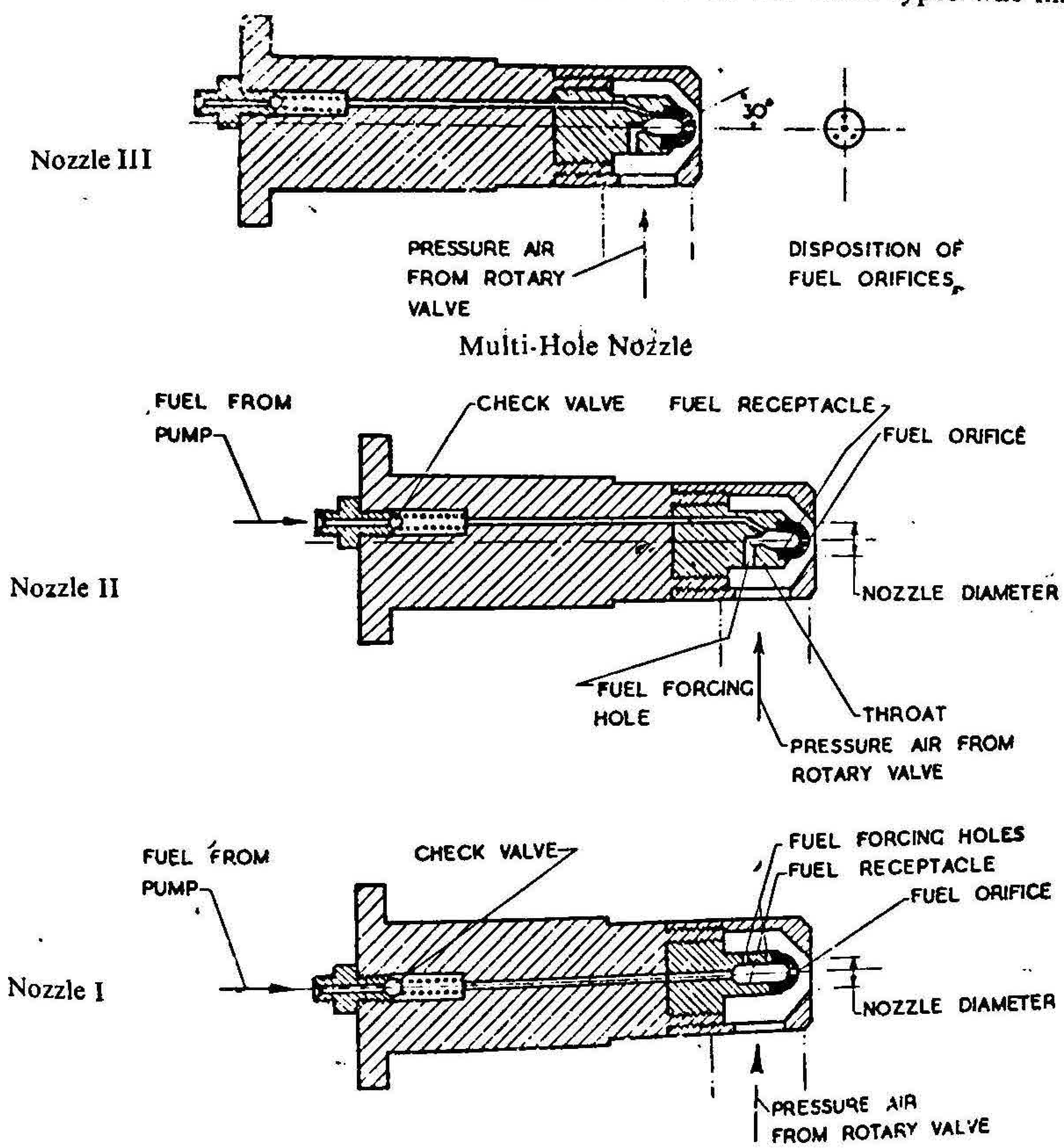



FIG. 14
Nozzle Design

0.25 in., since the flow conditions would then be sufficient for proper atomisation of the fuel.

Preliminary tests indicated that with all the three nozzels the engine could be operated although the maximum power output was very low. The engine would stall beyond this maximum limit. It was observed on visual observation of the pressure time phenomena on the oscilloscope, that with nozzles I and II the ignition angle was around 6° BTDC under light load conditions. With the multihole nozzle it was even earlier.

With increasing load the igrition angle would advance rapidly resulting in engine stalling.

Nozzle I was apparently ineffective when pressure reversal took place. Any fuel left over after injection in the first phase of flow (i.e., from the cylinder into the combustion chamber) would not be injected into the main cylinder and would carbonise within the nozzle. Thus it was prone to accumulate carbon rapidly.

Nozzles II and III were designed to be effective in injecting fuel in both phases. Nozzle III was in addition of the multihole type. In the initial stages of the investigation carbon formation was observed with nozzles II and III as well. This was however overcome by modifying the nozzle designs as shown in Fig. 15. The modified nozzle design was used in all subsequent experiments.

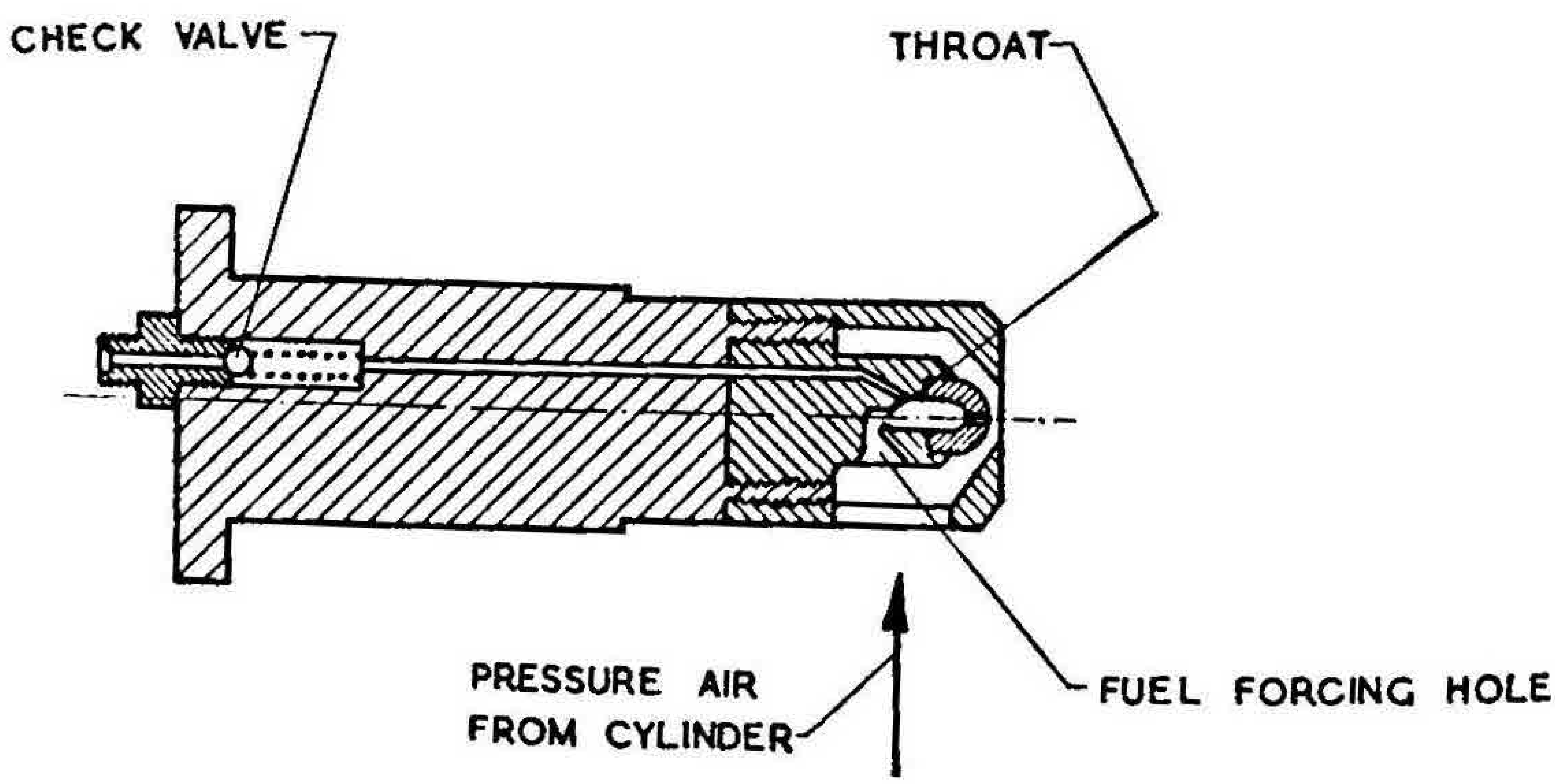



FIG. 15

Modified Nozzle Design

Further experiments were conducted to study the effect of nozzle diameter. fuel orifice diameter, fuel receptacle volume and angle of pressure build up on the performance of the engine.

The effect of the fuel orifice on engine performance was determined by varying it from 0.02 to 0.05 in. diameter. The test results are given in Table 1. It was observed that the angle of ignition was closely related to the fuel orifice diameter. As the fuel orifice diameter was increased the angle of ignition would advance rapidly and the engine would stall.. The power output and fuel consumption of the engine improved as the fuel orifice diameter was made smaller. The smallest possible diameter of fuel orifice that could be used was however limited by blocking of the fuel orifice with soot. Increasing the number of fuel orifices had little effect either on the performance of the engine or the ignition angle. Increasing the length of the fuel orifice would retard the angle of ignition. But prolonged runs with long orifices was not possible due to early blocking up of the fuel orifice.

TABLE 1

Effect of fuel orifice diameter

Fuel used: High speed diesel oil;
Nozzle diameter, 0.201 in.; Dynamometer constant: W N/3000

| Sl. No. | Fuel orifice diameter in. | Speed r.p.m. | Load on brake arm lb. | Exhaust temp. °F | B,H.P. | Specific fuel consumption lb/b.h. p-hr. |
|---|---|---|---|---|---|---|
| | | 700 | ...... | 340 | | |
| | | 700 | 2 | 365 | 0.46 | 7.45 |
| 1 | 0.050 | 690 | 4 | 375 | 0 92 | 3.80 |
| | | Engine stalls beyond 4 lb. load | | | | |
| | | 700 | — | 3 0 | | |
| | | 700 | 2 | 340 | 0.45 | 3.74 |
| 2 | 0.032 | 700 | 4 | 380 | 0.935 | 2.15 |
| | | 690 | 6 | 440 | 1.385 | 1.93 |
| | | Engine stalls beyond 6 lb. load | | | | |
| | | 700 | ...... | 320 | | |
| | | 700 | 2 | 330 | 0.46 | 3.15 |
| 3 | 0.020 | 700 | 4 | 350 | 0.935 | 1.68 |
| | | 700 | 6 | 390 | 1.4 | 1.45 |
| | | 690 | 8 | 440 | 1.84 | 1.37 |
| | | Engine stalls beyond 8 lb. load | | | | |

The effect of nozzle diameter on engine performance was studied by varying it from 0.201 in. to 0.377 in. the fuel orifice diameter being 0.02 in. The results of the experiments are given in Table 2. It was observed that the ignition angle was hardly affected by the nozzle diameter. The specific fuel consumption and power output improved as the nozzle diameter was increased. However the maximum power output of the engine would decrease beyond 0.35 in. diameter.

TABLE II

Effect of nozzle diameter

Fuel used: High speed diesel oil;
Fuel orifice diameter: 0.02 in.; Dynamometer constant W N /3000

| Sl. No. | Nozzle diameter in. | Speed r.p.m. | Load on brake arm lb. | Exhaust temp. °F | B.H.P. | Specific fuel consumption lb./b.h.p -hr |
|---|---|---|---|---|---|---|
| | | 700 | ...... | 320 | | |
| 1 | 0.201 | 700 | 4 | 350 | 0.935 | 1.685 |
| | | 690 | 8 | 410 | 1.87 | 1.385 |
| | | Engine stalls beyond 8 lb. load | | | | |
| | | 700 | ...... | 320 | | |
| 2 | 0.246 | 700 | 4 | 345 | 0 935 | 1.65 |
| | | 700 | 8 | 400 | 1.87 | 1.22 |
| | | Engine stalls beyond 8.5 lb. load | | | | |
| | | 700 | ...... | 320 | | |
| | | 700 | 4 | 350 | 0.935 | 1.595 |
| 3 | 0.323 | 700 | 8 | 400 | 1 87 | 1.11 |
| | | 690 | 10 | 420 | 2.34 | 0.975 |
| | | Engine stalls beyond 10 lb. load | | | | |
| | | 700 | ...... | 315 | | |
| | | 700 | 4 | 340 | 0.935 | 1.595 |
| 4 | 0.350 | 700 | 8 | 400 | 1.87 | 1.1 |
| | | 690 | 11 | 425 | 2.56 | 0 902 |
| | | Engine stalls beyond 11 lb. load | | | | |
| | | 700 | ...... | 320 | | |
| | | 700 | 4 | 340 | 0.935 | 1.685 |
| 5 | 0.377 | 700 | 8 | 390 | 1.87 | 1.17 |
| | | 680 | 10 | 410 | 2.34 | 1.0 |
| | | Engine stalls beyond 10 lb. load | | | | |

Engine performance for a given fuel orifice and nozzle diameter was found to be practically independent of the receptacle volume. When the receptacle volume exceeded about 0.25 cc, however, there would be carbon formation in the receptacle leading to blocking up the fuel orifice.

The point of pressure build up was varied by varying the angle of entry of the knob from 46° to 35° BTDC. As the point of pressure build up was varied from 46° to 38° BTDC the ignition point as found from indicator diagrams was almost correspondingly retarded thereby. Further retarding the point of pressure build up had however very little effect on the angle of ignition. The nozzle configuration was less critical with the point of pressure build up retarded to 38° BTDC. The fuel orifice diameter and nozzle diameter could be varied from 0.02 to 0.025 in. and 0.3125 to 0.377 in. respectively with little change in engine performance. The results of the experiments are given in Table 3.

TABLE III

Effect of point of pressure build up on ignition angle

| Sl. No. | Angle of entry of knb deg. B.T.D.C. | Load on brake arm lb. | Ignition angle deg. B.T.D.C. |
|---|---|---|---|
| 1 | 46 | No load | 6 |
| | | 6 | 16 |
| 2 | 40 | No load | 3 |
| | | 6 | 9 |
| | | 10 | 13 |
| 3 | 38 | No load | 2 |
| | | 6 | 8 |
| | | 10 | 10 |
| | | 14 | 12 |
| 4 | 35 | No load | 2 |
| | | 10 | 10 |

The optimum nozzle design in relation to the geometry of the pressure differential device was arrived at from the test results. The optimum point of pressure build up is 38° BTDC and the corresponding nozzle dimensions are as follows: fuel orifice diameter 0.02 in. nozzle diameter 0.35 in. and receptacle volume 0.2 cc.

Performance test results of the engine with the optimum nozzle and pressure differential configuration are shown plotted in Fig. 16.

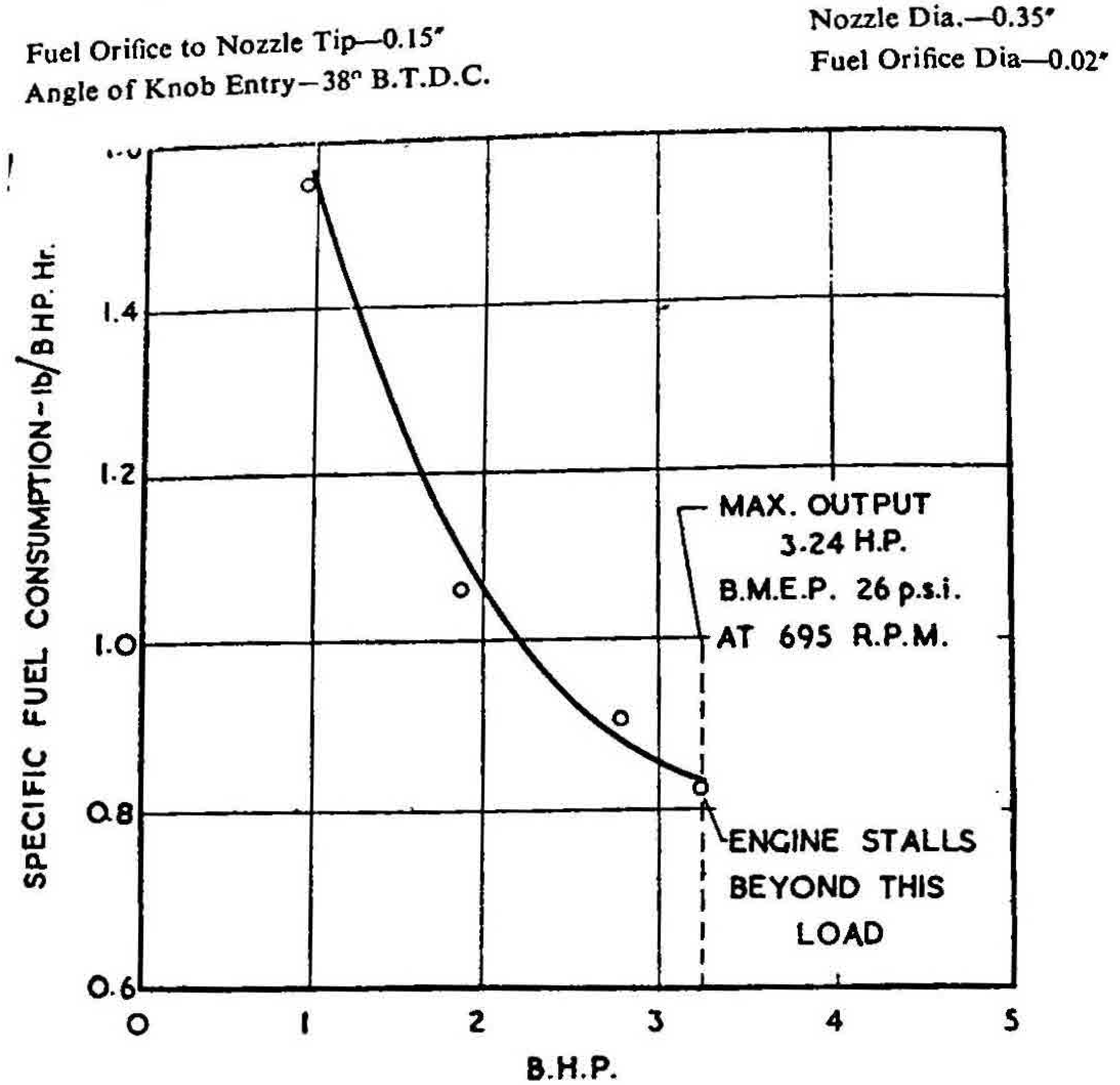
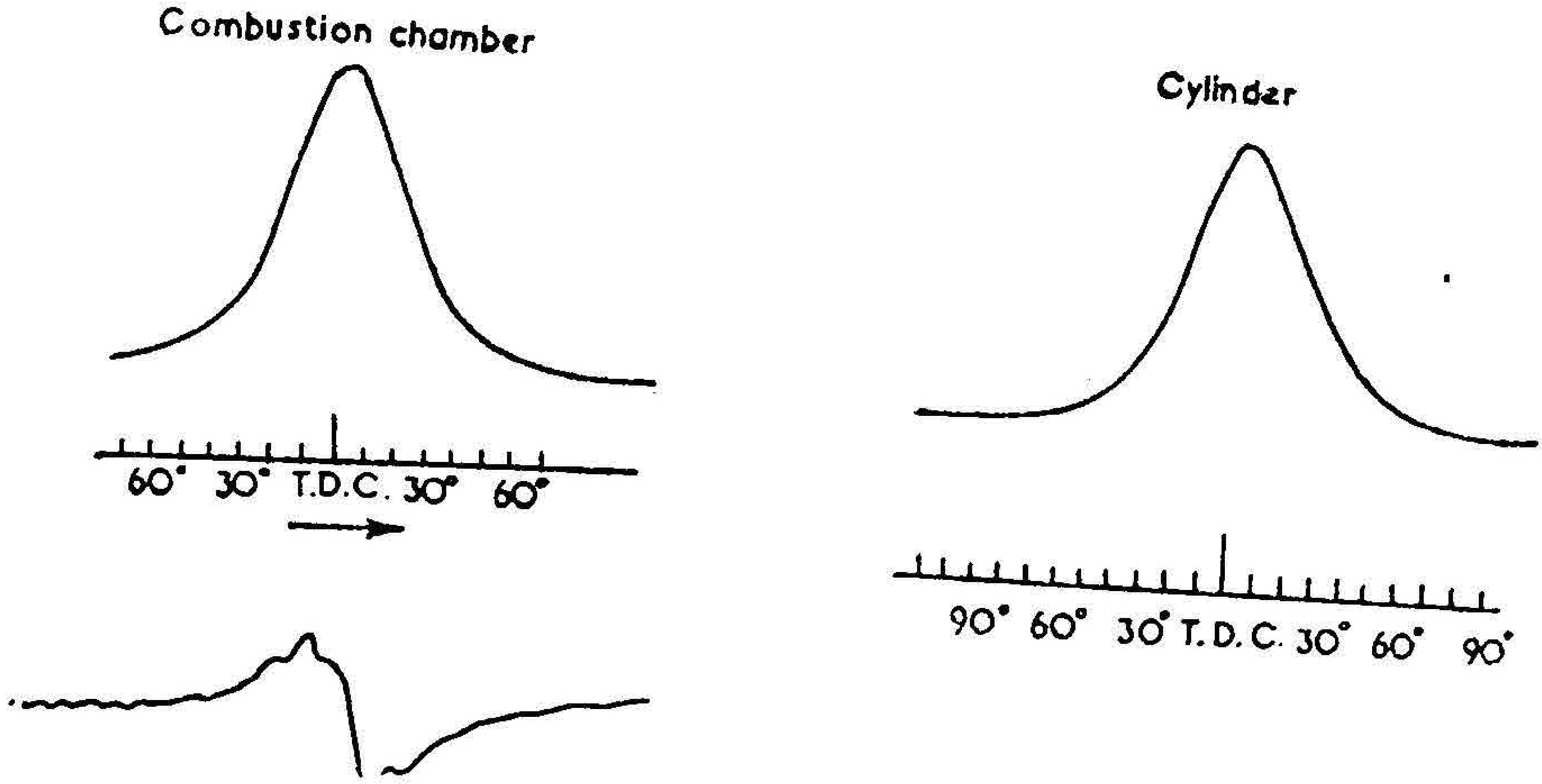



FIG. 16

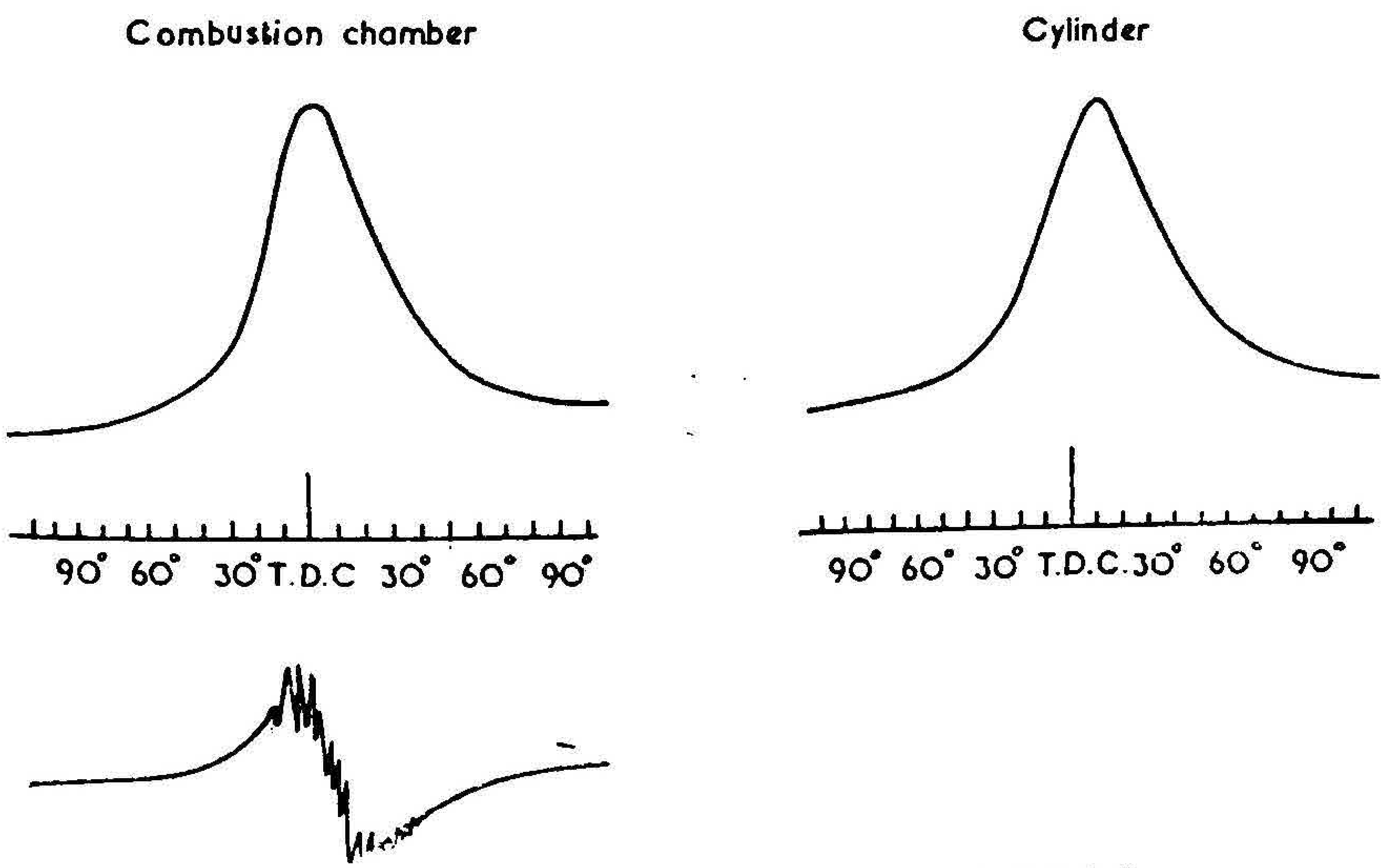
Specific fuel Consumption with Optimum Nozzle and Pressure Differential Configuration

The maximum power output of the engine under steady running conditions was 3.7 HP (corrected) at 695 r.p.m., corresponding to 74% of the rated power output of the engine. The specific fuel consumption corresponding to this load was 0.825 lb/b.h p-hr. It was quite clear from pressure time diagrams that early ignition was mainly responsible for limiting the maximum power output of the engine. Fig. 17 shows representative pressure-time diagrams for the cylinder and combustion chamber with the optimum nozzle.

Combustion conditions as indicated by the exhaust discolouration at light loads were satisfactory. As the load was increased the conditions steadily deteriorated, the exhaust being whitish suggesting presence of unburnt fuel. The general running of the engine was smooth and steady. In all the experiments the combustion chamber top half was separate and acted as a heat storage member. Further insulation of the combustion chamber by interposing gasket material between the top flange and combustion chamber as shown in Fig. 18 improved combustion slightly but had no effect on maximum power output.

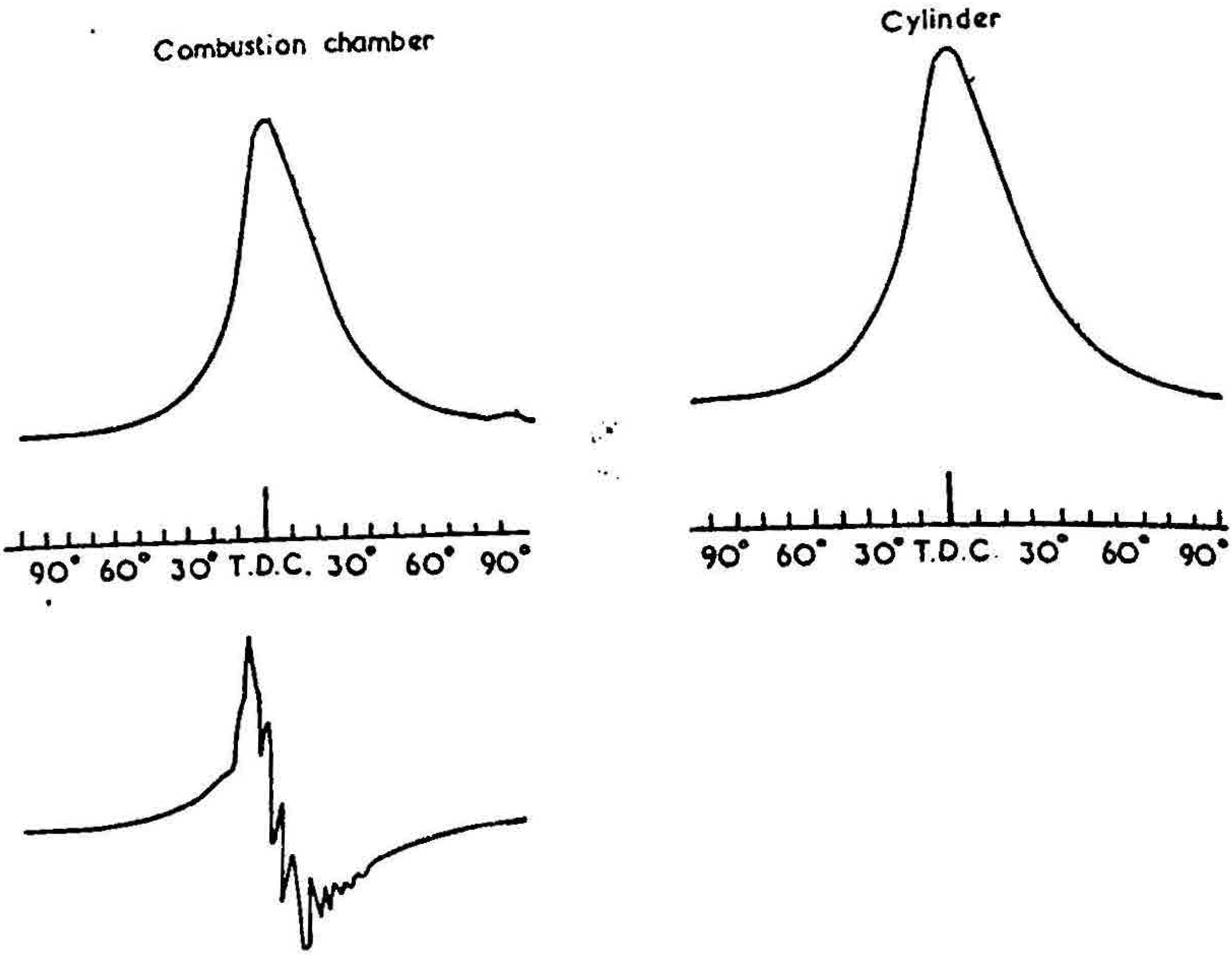



Load—3 lbs, Speed—700 r.p.m., Ignition angle—5° B.T.D.C.




Load—7 lbs, Speed—700 r.p.m., Ignition angle—8° B.T.D.C.

FIG. 17—*(continued in next page)*




Load—14 lbs., Speed—690 r.p.m., Ignition angle—12° B.T.D.C.

FIG. 17

Rate of pressure rise and pressure time diagrams for the cylinder and combustion chamber

## EXPERIMENTS TO RETARD IGNITION ANGLE

In an effort to gain control over the ignition timing the pump was used for both timing and metering of fuel.

It was expected that this would indicate how efficient a displacer type air blast system could be if the point of ignition could be controlled.

The results obtained with this arrangement however were disappointingly poor. The performance of the engine was even worse than with timing controlled by the pressure differial device. It was almost impossible to operate the engine with injection retarded beyond 35° BTDC as severe knocking and missing resulted. With injection very near TDC the engine would hardly run on its own power with combustion taking place late in the expansion cycle.

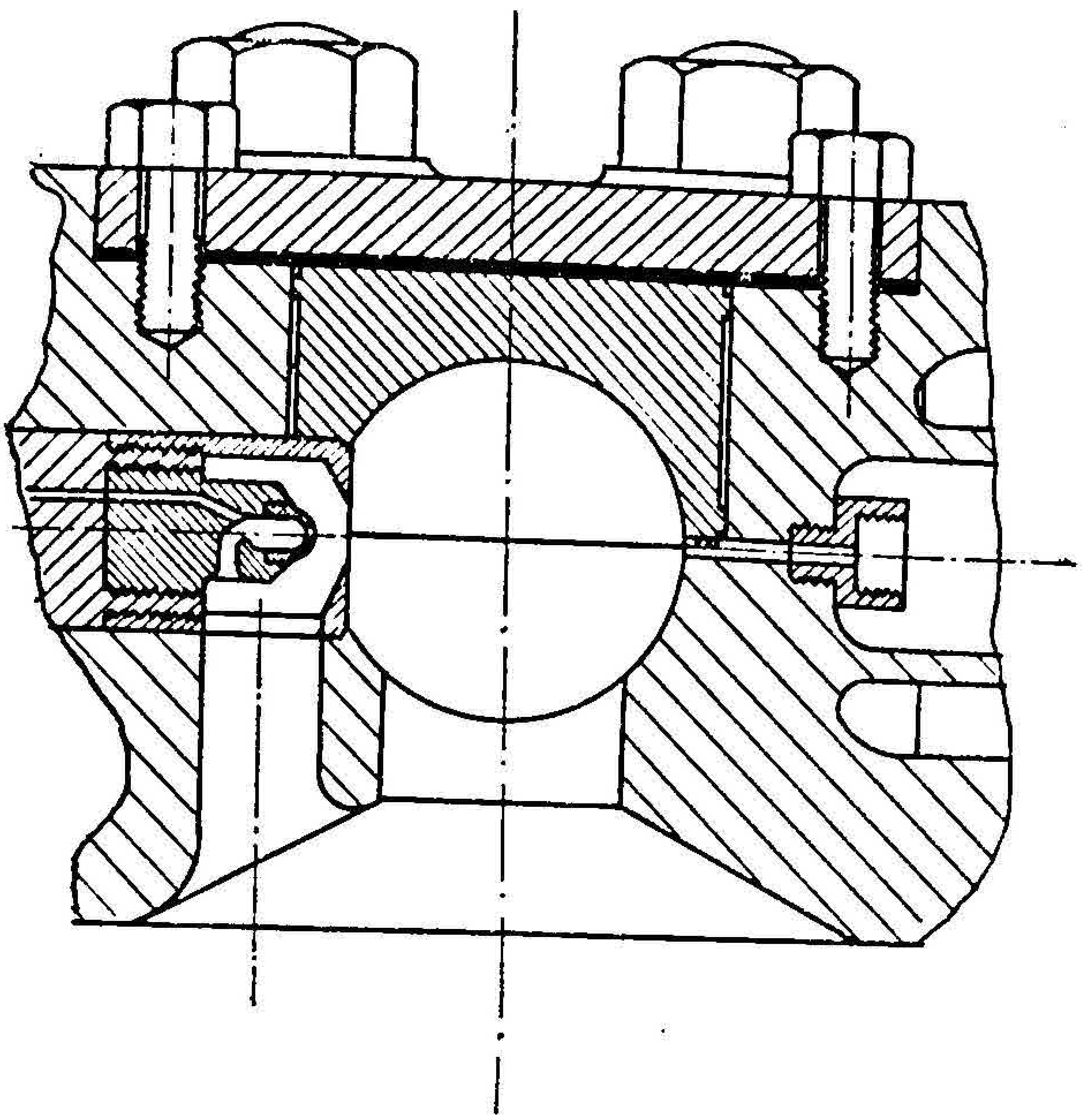


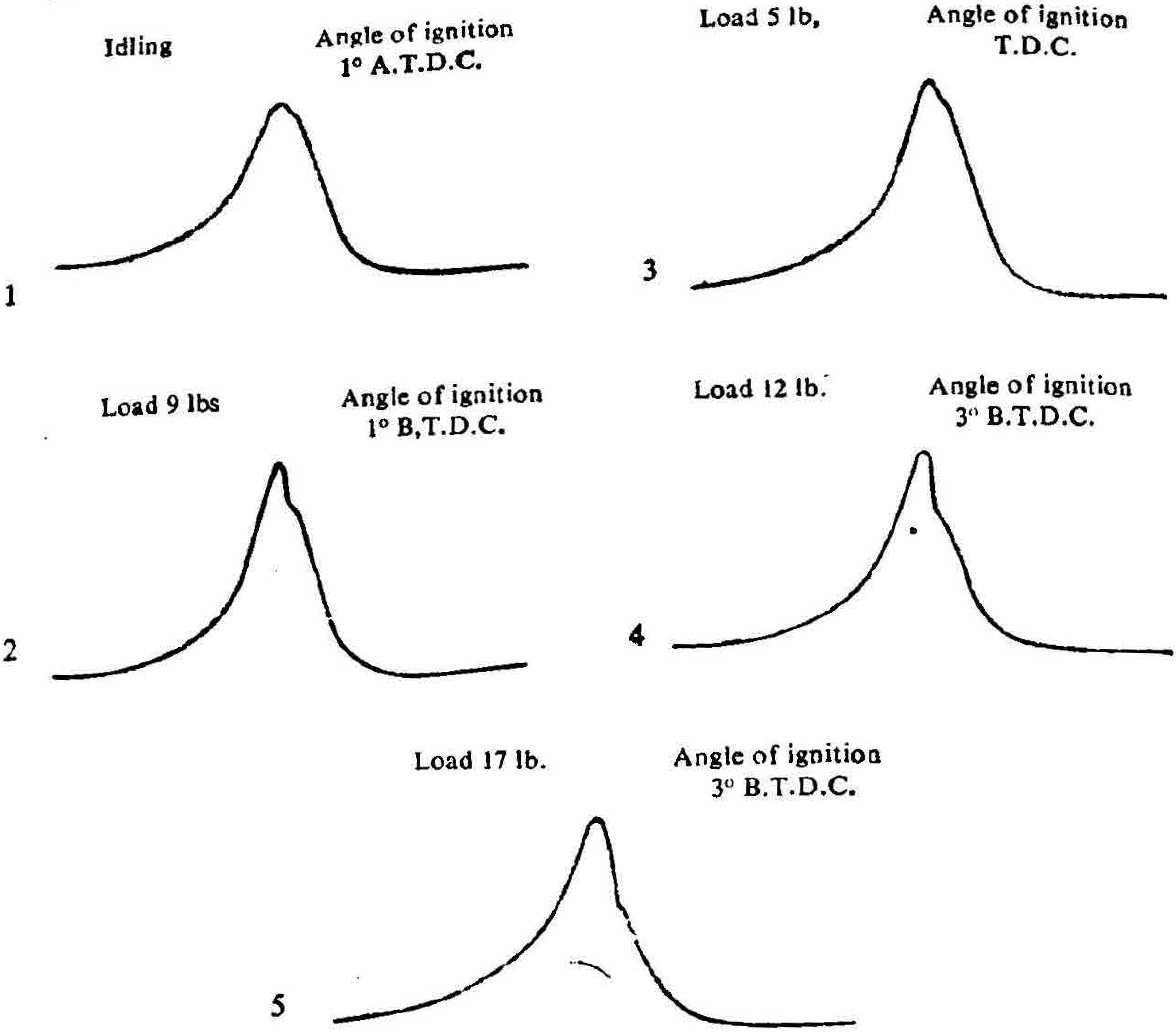
FIG. 18

## ALCOHOL INDUCTION

Induction of alcohol along with the air into the engine was tried out as a possible means of retarding the ignition angle, by fitting a carburettor on the air intake. As is well known the large latent heat of evaporation of alcohol depresses the charge temperature and increases the delay period[4]. This had the desired effect and the engine could also deliver a higher output. Fig. 19 shows pressure time diagrams with alcohol induction at various loads. The maximum power output was improved to 4.5 hp from 3.7 hp and the ignition angle retarded to 3 BTDC from 10 BTDC. In view of the fact that the test engine was of the crankcase scavenged two stroke type there was some uncertainty in regard to the quantity of alcohol actually reaching the cylinder. No serious attempt was therefore made to estimate the rate of alcohol consumption.

## DISCUSSION OF RESULTS AND CONCLUSIONS

It appears from the results obtained that the two requirements to be achieved by the pressure differential device, namely proper timing of injection for maximum output and satisfactory atomisation of the fuel for rapid and




Engine Speed 700 RPM, Dynometer Const $=\frac{WN}{3000}$

FIG. 19

Pressure Time Diagrams at Various Loads With Alcohol Induction

complete combustion are incompatible. As the point of pressure build up is varied from 46° to 38° BTDC, the ignition angle is retarded thereby improving power output and fuel economy. Beyond 38°, however, the angle of ignition remains constant and the maximum power output of the engine begins to decrease. The pressure differential system is ineffective in controlling the angle of ignition beyond this limit. Retarding the point of pressure build up reduces the velocity and mass flow of air through the nozzle. Beyond a particular limit the flow conditions through the nozzle would possibly decrease to such an extent as to affect the quality of atomisation of the fuel. Combustion would then deteriorate resulting in low power output and poor fuel economy.

The improvement noted in the engine output when inducting alcohol can be mainly attributed to the increased delay in ignition while the pattern of pressure build up and hence the degree of atomisation remain unaffected. The effect of alcohol induction on fuel economy can be properly assessed if the system is tried out on a four stroke engine.

Another means of overcoming the drawback of early ignition appears possible by resorting to deposition of the fuel on a surface with closely defined temperature limits to regulate the rate of fuel release, as in the M-combustion system. Earlier injection then becomes feasible without the danger of too early ignition. With such an arrangement there would be no need for fine atomisation either. Careful consideration would have to be given in such a system for proper control of the surface temperature and air movement in the combustion chamber.

Some preliminary work done to explore the possibilities of applying pumpless injection to petrol engines gave encouraging results. The test engine used for the diesel experiments was operated successfully as a petrol engine by changing the compression ratio to 10.5 to 1 and fitting a spark plug in place of the pressure pick up. No other modifications were made to the combustion chamber. The engine could develop 3.5 hp at 700 r.p.m. without any sign of knock in spite of the fact that the octane number of the petrol used was about 75. This line of development of the system appears to hold promise especially for stratified charge engines with divided combustion chambers.

## Acknowledgement

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