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EFFECT OF IGNITION POINT LOCATION AND CONNECTING PASSAGE GEOMETRY ON PERFORMANCE AND COMBUSTICN CHARACTERISTICS OF TORCH IGNITION ENGINE

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ABSTRACT

The present experimental analysis is carried out to investigate the effect of connecting passage shape and flame initiation point on combustion characteristics in a homogeneous charge torch ignition engine. The main part of the experimental program is concerned with the optimization of the combustion process in such engine.. This can extend the effective lean burn limit in spark ingition engine. Two different passage shape and three different spark gap projection are used in this study. Engine performance and combustion characteristics at different engine operating variables are studied. The burning rate for the investigated geometry at different equivalence ratio is determined from the measured cylinder pressure vs crankangle diagram. Engine cyclic variations are also considered.

The obtained results shows that engine performance and combustion characteristics are influenced by these design parameters. Convergent-divergent passage shape and extended spark gap projection appear beneficial. Improved fuel economy, enhancing the combustion rate and extended the effective lean burning limit are recorded.

INTRODUCTION

Based on theoretical view-point, operation of spark ignition engine at lean fuel-air ratios and higher compression ratios should improved engine thermal efficiency. This is due to favorable thermodynamic properties of the charge and reduced throttling losses. This is confirmed in current practice with compression ignition engine. However, conventional spark ignition engine actually reach the lowest specific fuel consumption at fuel-air ratios not very far from the stoichiometric. This because, with leaner mixtures combustion becomes too slow and erratic (1,2). Improved spark ignition engine fuel economy to day becomes imperative due to the increasing fuel costs and operating such engines most of time at part load where thermal efficiency is low. To burn lean effectively it is necessary to accelerate combustion rate in order to compensate for the lower burning velocity of weak mixtures. Fast burning is also required in order to avoide the onset knocking at higher compression ratios.

The logical means of acceleration the combustion rate is to enhance the degree of turbulence within the combustion chamber. It has been found that the most important turbulence generated is that present near the spark plug, at or shortly after, the time of ignition. This is because at this time the small flame kernel ; most susceptible to the turbulence in the mixture surrounding it (3). The turbulence motion of the charge are generated primarly during the intake and compression stroke. The methods have been proposed for increasing.the motion of the charge includes modified intake design (4), the use of shrouded valves (5) and special designs of engines combustion chambers (6).

It was found that much accelerated combustion rates can be achieved by adopting the torch chamber engine concept (7). In such system, the movement of the piston during compression stroke comprisses the mixture and forces part of it into the swirl pre-chamber creating a great deal of turbulence therein. The mixture is then ignited by the spark source located in it. Because of the resulting strong eddies within the pre-chamber, the mixture burnt very quickly causing a rapid rise in pressure. The burnt gases is then rapidly expanded through the connecting passage serves as an ignition source for the main chamber mixture. The flame then propagates through the main chamber mixture wherein the bulk of the energy release occurs.

The present work deals with the findings from continued development of the torch ignition combustion process. During the extensive optimization program two main design parameters are investigated, namely the connecting passage shape and spark gap projection. Engine performance, effective lean burn limit, combustion characteristics and cyclic variation are studied.

EXPERIMENTAL APPARATUS AND TECHNIQUES

The experimental set-up is shown schematically in Fig. 1. The engine used is a modified four stroke, four cylinder air cooled engine. The engine specifications are shown in Table 1. The combustion chamber cavity is completly accomudated in the cylinder head and of the swirl chamber type. The design features of the cylinder head are shown in Fig. 2a. The connecting passage inters the swirl chamber in a tangential direction (30° inclination) to creat strong motion of the mixture in it. Because of the numerious design restrictions of swirl chamber cavity, a limited design parameters are investigated. The spark plug is located in the position of the heating plug in the swirl chamber.

Table 1. Engine Specifications

Engine Bore (Cm)	11.0
Engine Stroke (Cm)	14.0
Displacement (liter)	5.32
Connecting Rod Length (Cm)	21.5
Compression Ratio	9:1
Preohamber to total Clearance Volume Ratio(%)	17
Connecting Passage Diameter (Cm)	.9

Two sets of cylinder heads equipped with two different connecting passage shape have been used. The first set is originally fitted with a straight passage having equivalent diameter of .9 Cm. The second set of cylinder head is equipped with a convergent divergent connecting nozzle having a .9 Cm hroat diameter. The divergent angle towards swirl chamber is taken to be of 19°. The connecting passage diameter has been determined according to the optimum turbulent motion using the method proposed by Adams (8). The minimum connecting orifice diameter is chosen also such that the flame can pase it without quenching.

Other engine modifications include a reduction of compression ratio from 17:1 to 9:1 by fitting spacers with predetermined thickness between liner and crankcase body. The intake manifold is modified to be fit with a conventional carburetor selected according to engine air flow requirements at stream conditions. The carburetor used is modified to be fit with an adjustable needle valve as shown in Fig. 2b. The engine is also fitted with a conventional ignition system. The task which necessitates a power take-off unit to be developed to transimit a rotary motion to the distributer is shown in Fig. 2c.

The rate of fuel consumed is determined using volumetric type flow meter, while the rate of air consumption is indicated using a tank and orifice technique. The ignition timing is indicated using a (CUSSON - P. 4605) ignition advance unit. A fluid friction dynamometer type HOFMAN-BRN-38 is used as engine brake and it also fitted with a speed meter to record the engine RPM. The instantaneous gas pressure and pressure rates are measured using a (CUSSON-P4546) comperhensive kit. It comprises a KISTLE-P4558 piezo-electric pressure transducer with a spark plug adopter, charge amplifire, crankangle degree marker system, a cathod ray oscilloscope and conventional camera. Engine cyclic variations are investigated using a system shown in Fig. 3. It comprises two groups, the first is to record a sample of consecative engine cycle which comprises a charge amplifier, a tape recorder and a magnetic tape. The second group is to reproduce the recorded cycle samples and display them on special cards. It comprises a frequency analysis oscilloscope, and X - Y plotter. The system is adjusted and calibrated such that one Cm displayed on the oscilloscope screen is equivalent to pressure of 10 Kg/Cm^2.

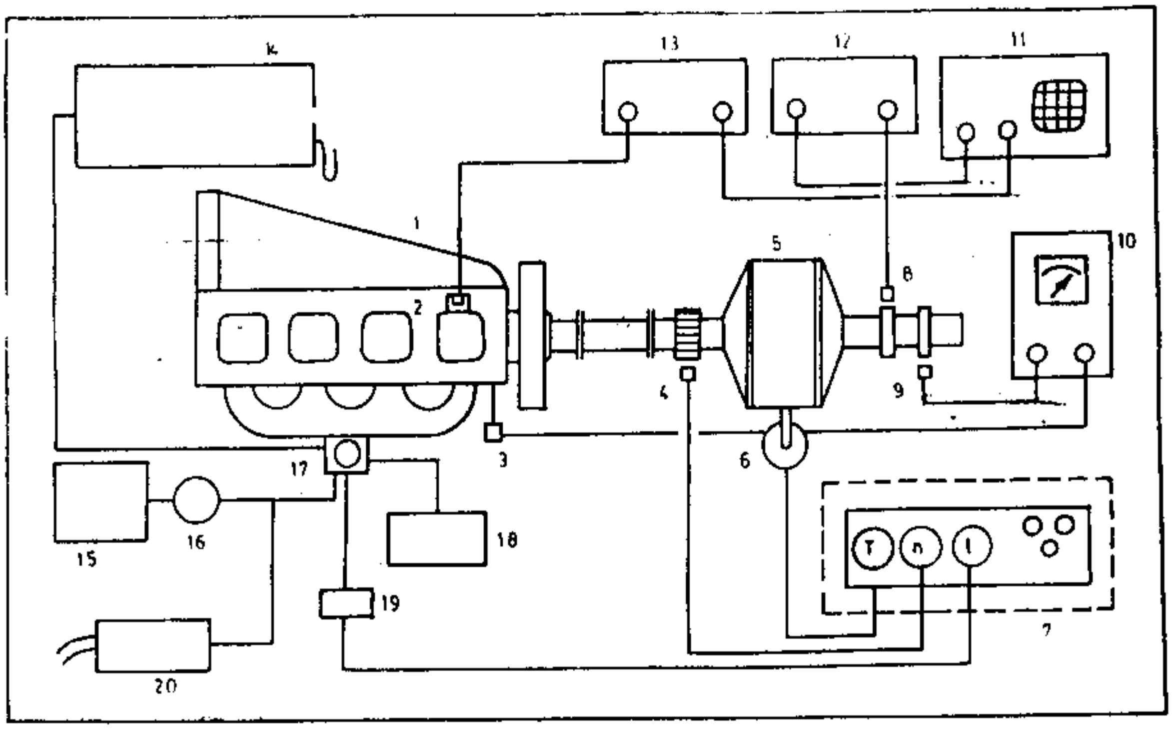
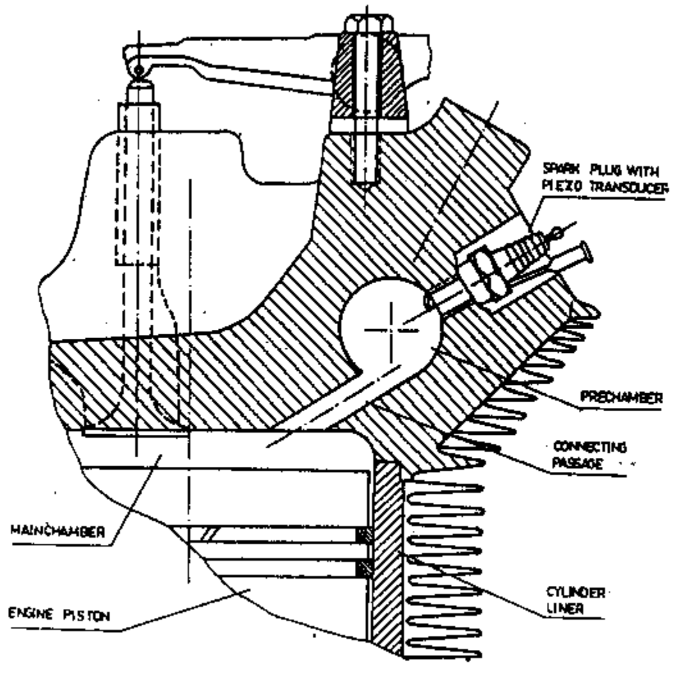


Fig. 1 SCHEMATIC DIAGRAM OF THE EXPERIMENTAL SET UP



Fig(2a) REVISED ENGINE CYLINDER HEAD

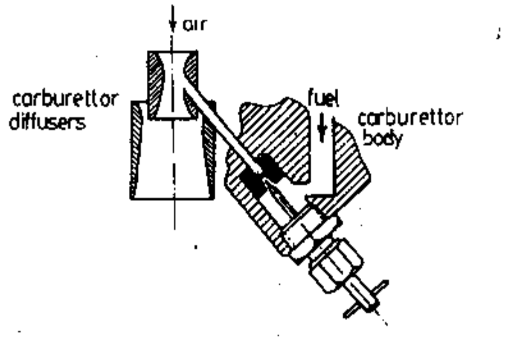
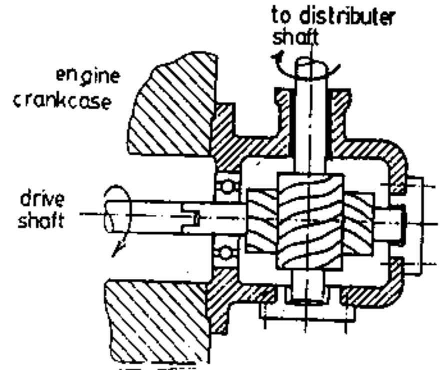
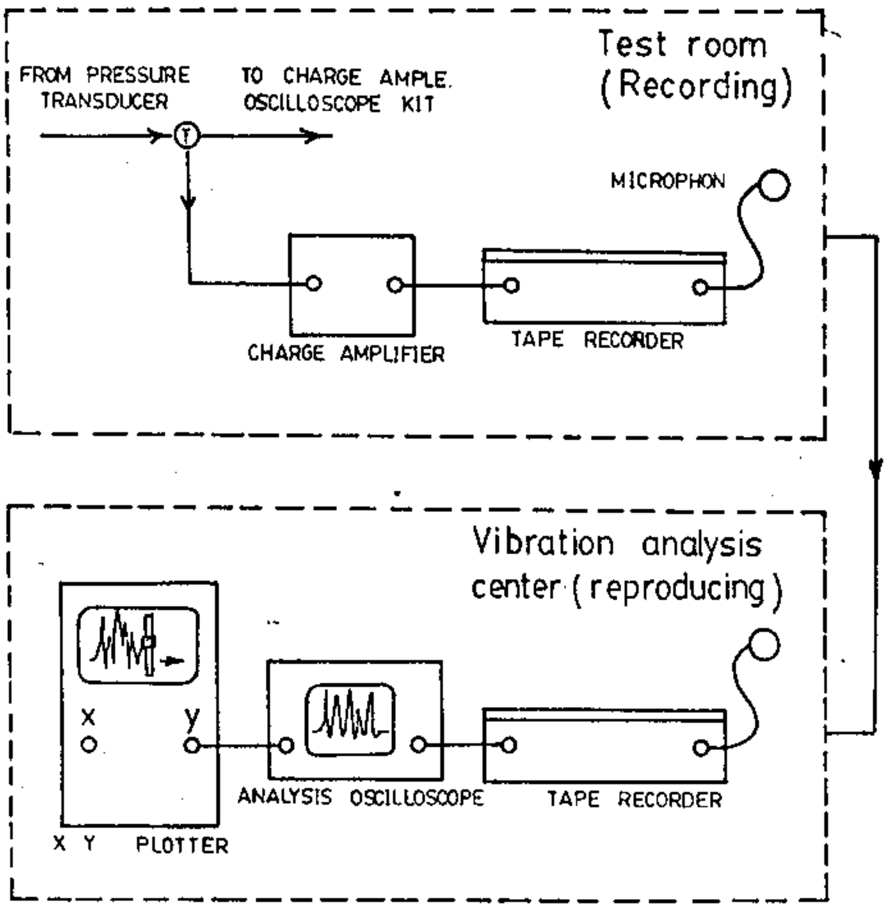


Fig (2b) MIXTURE STRENGTH CONTROLLING VALVE



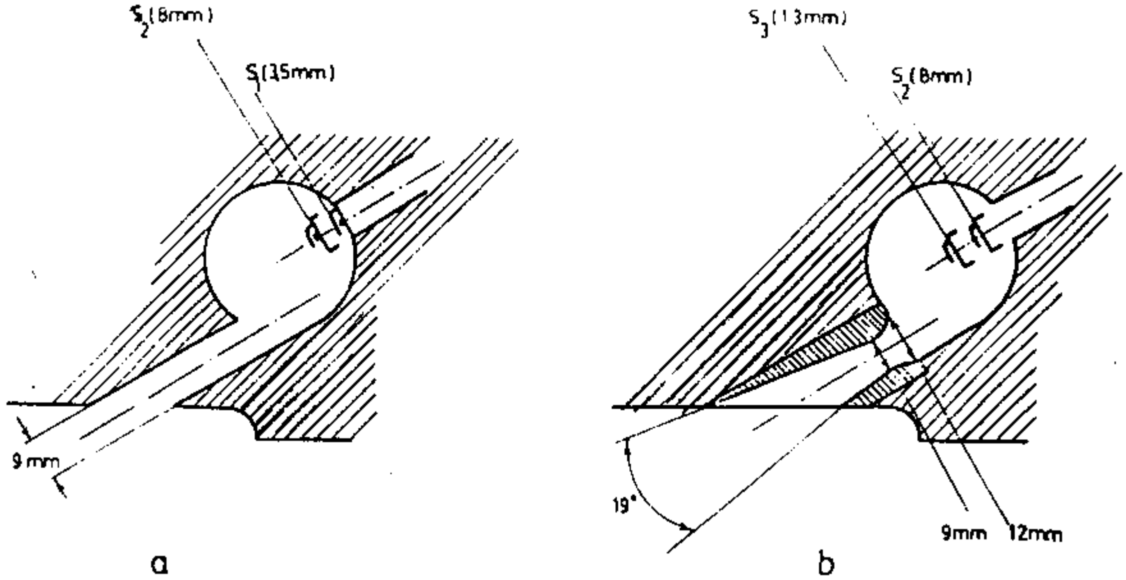
Fig(2c) DEVELOPED POWER TAKE OFF TO DRIVE IGNITION DISTRIBUTER SHAFT



Fig(3) CYCLIC VARIATION INVESTIGATING SYSTEM

TEST RESULTS AND DISCUSSIONS

The matrix of engine tests are summarized in Table 2. Two series of test results are obtained. In the first series, engine performance and lean running ability for all the investigated geometry A, B, C and D as shown in Fig. 4 are studied. The lean misfire limit (LML) of stable engine operation is determined by occasional audible misfire and loss of power. In the second series of test results, combustion characteristics and cyclic variations are studied from the measured cylinder pressure vs crankangle diagram. Allover the tests, the engine power is determined at MBT (minimum best torque) spark advance. The inlet mixture intake temperature is maintained constant at 30 ± 1 °C by preheating system and thermostate. A limited maximum engine speed of 2000 RPM because the engine was originally low speed Diesel engine.



Fig(4) PRECHAMBER ARRANGEMENTS TO BE TESTED

ARRANGEMENT A _	STRAIGHT CONNECTING PASSAGE WITH SPARK GAP LOCATION	S_1
ARRANGEMENT B _		S_2
ARRANGEMENT C _	CONVERGANT DIVERGENT	S_2
ARRANGEMENT D _		S_3

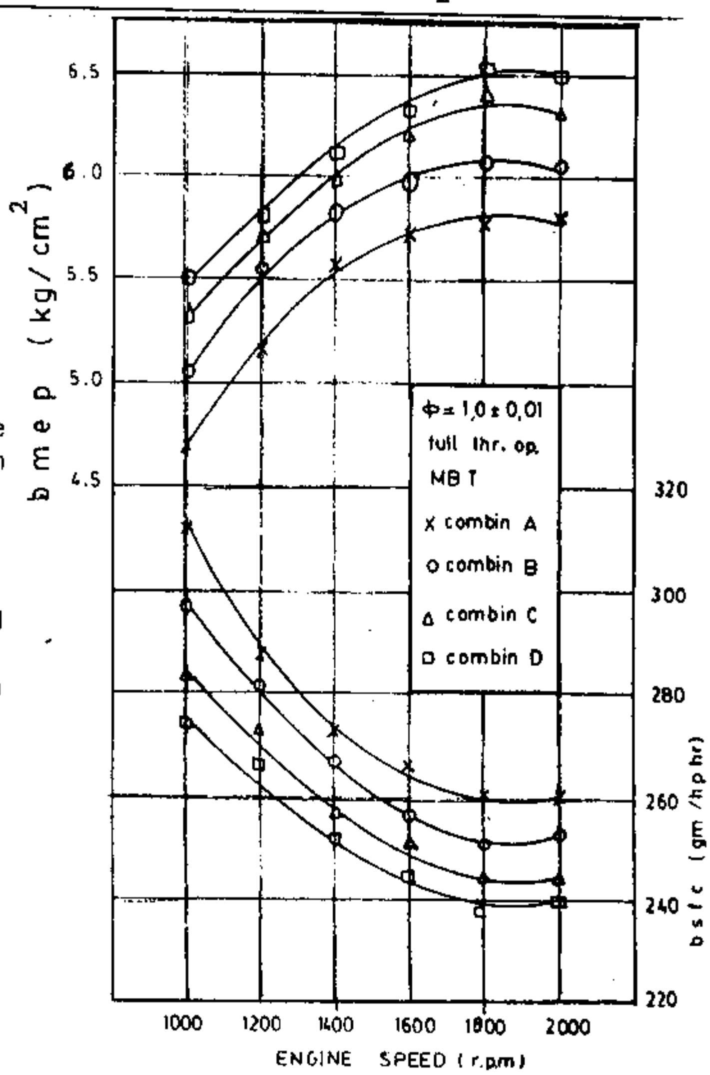
To investigate the effect of connecting passage shape on engine performance, an extensive engine test are conducted at different engine speed and different throttle opening for all geometry A, B, C, and D. Full, 3/4 and 1/2 throttle opening conditions are used to represent the common operating range of automotive engine. Shown in Fig. 5 are the characteristics of engine performance at full throttle opening and an equivalence ratio of 1.0. From these results and trends obtained at different throttle opening, it can be seen that the minimum bsfc (brake specifice fuel consumption) for all combination A, B, C and D lies at an engine speed ranges from 1600 to 1800 RPM and drifts

Table 2. Matrix of Engine Tests

	Series I	Series II
Design Parameters		
Prechamber volume (%)	17	17
Passage shape	Straight and convergent divergent nozzle	Straight and convergent divergent nozzle
Passage dimater (Cm)	0.9	0.9
Compression ratio	9	9
Spark gap projection (mm)	3.5,8,13	3.5
Operating Parameters		
Equivalence ratio	1.1 till LML	1.1, 1.0, 0.9, 0.8
Engine speed RPM	1000 and up to 2000	1500
Throttle opening	full, 3/4, 1/2	full
Spark advance	MBT	MBT
Mixture inlet temp. (°C)	30 ± 1	30 ± 1

slightly towards less speed with the throttle partly clossed. Allover the engine speed and engine load range result, the combination D, of convergent divergent nozzle and extended spark gap projection, improves the engine performance.

The effect of mixture strengh on engine performance for the different combination tested is shown in Fig. 6 to 8,. Again, it can be seen that the combination D has the lowest bsfc allover the range of mixture strength and throttle opening. The gain is increased towards the lean side. This gain is within the range of 10% at equivalence ratio of 0.8 and full throttle operation and 4% at an equivalence ratio of 1.1, compared to arrangement A. An important finding during engine tests is that engine fitted with combination C and D at full throttle opening posses ability to lean runing up to 0.7 equivalence ratio with no misfiring, Fig. 6. Similar results are



Fig(5) ENGINE PERFORMANCE FOR DIFFERENT PRECHAMBER COMBINATIONS obtained at part throttle opening as shown in Fig. 7 & 8. However, the tendency of the engine to lean burn diminshes with increased throttling. This behaviour may be due to decreased compression pressure and increaseed dillution with the residual gases.

Shown in Figs.9 to 11 are the effects of mixture strength and throttle opening on MBT spark advance at the point of minimum bsfc for all geometry investigated. From these figures, it can be noted that at any given equivalence ratio the MBT spark advance is a minimum when the engine is fitted with combination D. This indicates that more rapid combustion and increased ability for running lean with this geometry.

In order to verify the findings from the engine performance tests, a second series of tests are conducted. In this series of results, measurements of instantaneous cylinder gas pressure

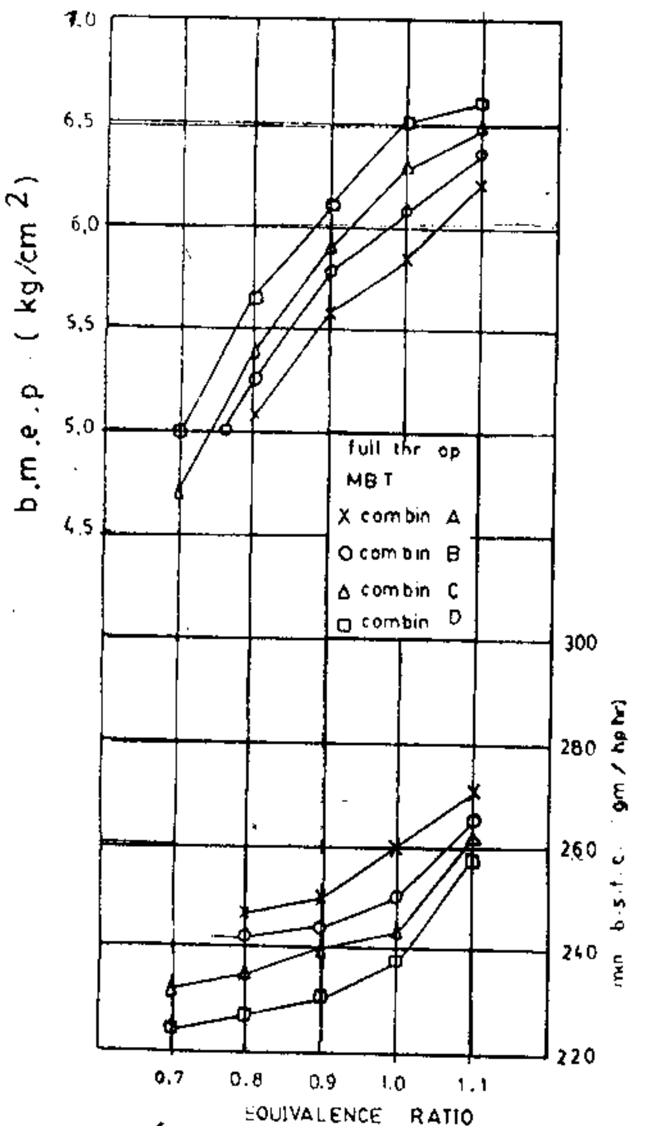


Fig (6) EFFECT OF EQUIVALENCE RATIO ON THE PERFORMANCE OF THE ENGINE WITH DIFFERENT PRECHAMBER COMBINATIONS AND AT FULL THR OP

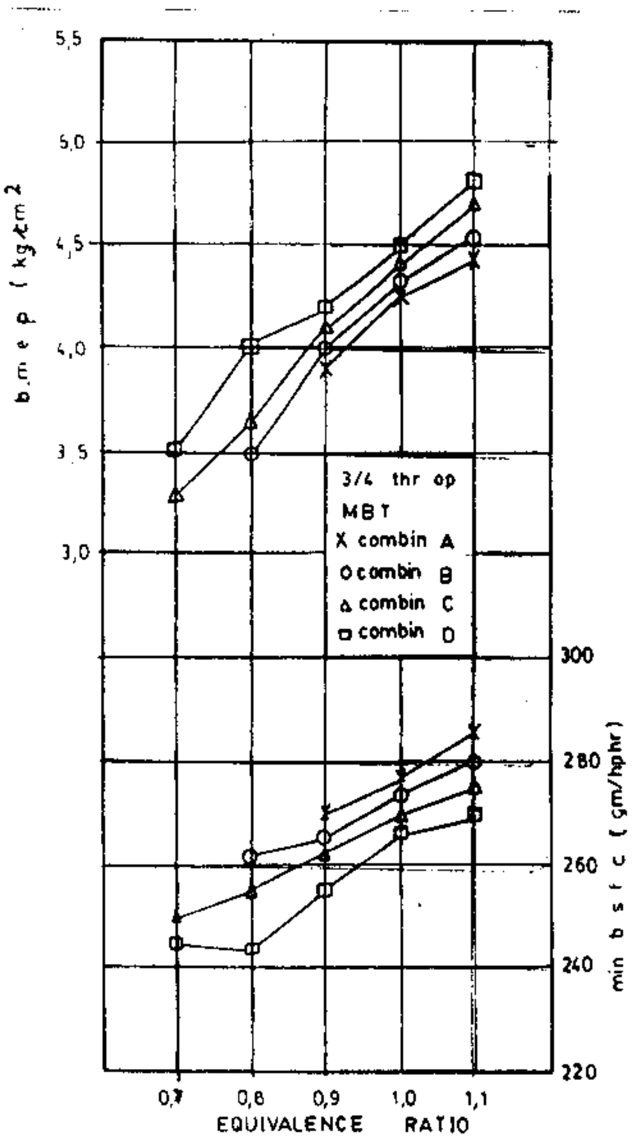
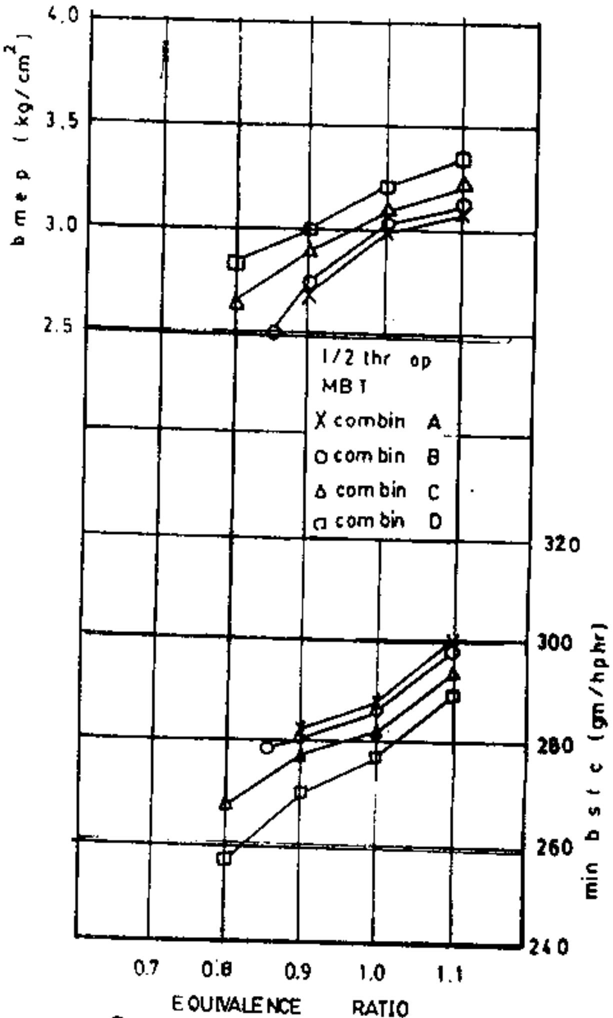
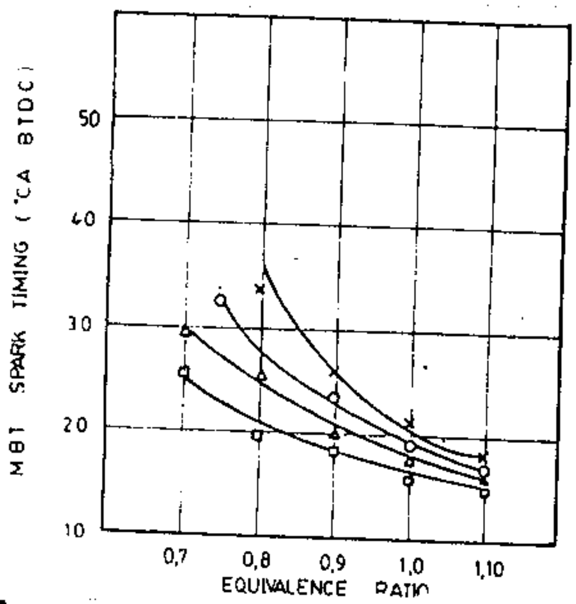


Fig (7) EFFECT OF EQUIVALENC RATIO ON THE PERFORMANCE OF THE ENGINE WITH DIFFERENT PRECHAMBER COMBINATIONS AND AT 3/4 THR OP are used to analysis the combustion process. A sample of recorded pressure traces are shown in Figs. 12 & 13. In the present analysis, the combustion process is considered to be divided into two phases. The initial phase, is xtending from the time of ignition to the instant when stable flame kernel is developed (sensible pressure rise). The second phase is the interval between the end of the initial phase and the instant of peak pressure recorded. Pressure rate traces are used to determine the duration of each phase.

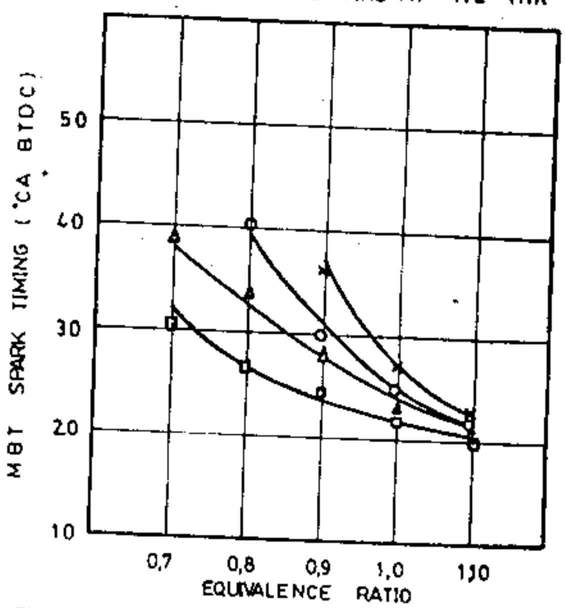
It can be seen from Figs. 14 through 17 that the combination D has a remarkable effect on both phases duration allover the equivalence ratios This may be due to improving the mixture motion in the torch chamber at the time of ignition. It can also be noted that the duration of initial combustion phase greatly



fig(8) EFFECT OF EQUIVALENCE RATIO ON THE PERFORMANCE OF THE ENGINE WITH DIFFERENT PRECHAMBER COMBINATIONS AND AT 1/2 THR OP.

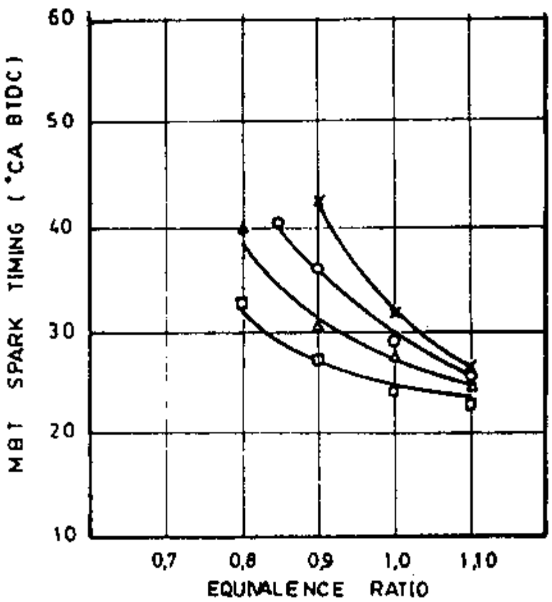


Fig(9)EFFECT OF EQUIVALENCE RATIO ON MBT SPARK TIMING FOR DIFFERNT PRECHAMBER ARRANGEMENTS (FULL THR. OP



Fig(10)EFFECT OF EQUIVALENCE RATIO ON MBT SPARK TIMING FOR DIFFERENT PRECHAMBER ARRANGEMENTS (3/4 THR. OP) dependent on the mixture strength, while the second phase is less dependent on it. As a result of decreased the combustion duration for combination D, the peak gas pressure and maximum pressure rise are slightly increased. Figs. 14 & 17.

It is well known that cyclic variation is one of the major problems in a lean burn engine. The dependence of the cyclic peak pressure variation up on the mixture strength and connecting passage shape is shown in Fig. 18. Here, we define the coefficient of cycle pressure variation CV,



Fig(11)EFFECT OF EQUIVALENCE RATIO ON MBT SPARK TIMING FOR DIFFERENT PRECHAMBER ARRANGEMENTS (1/2 THR.OP)

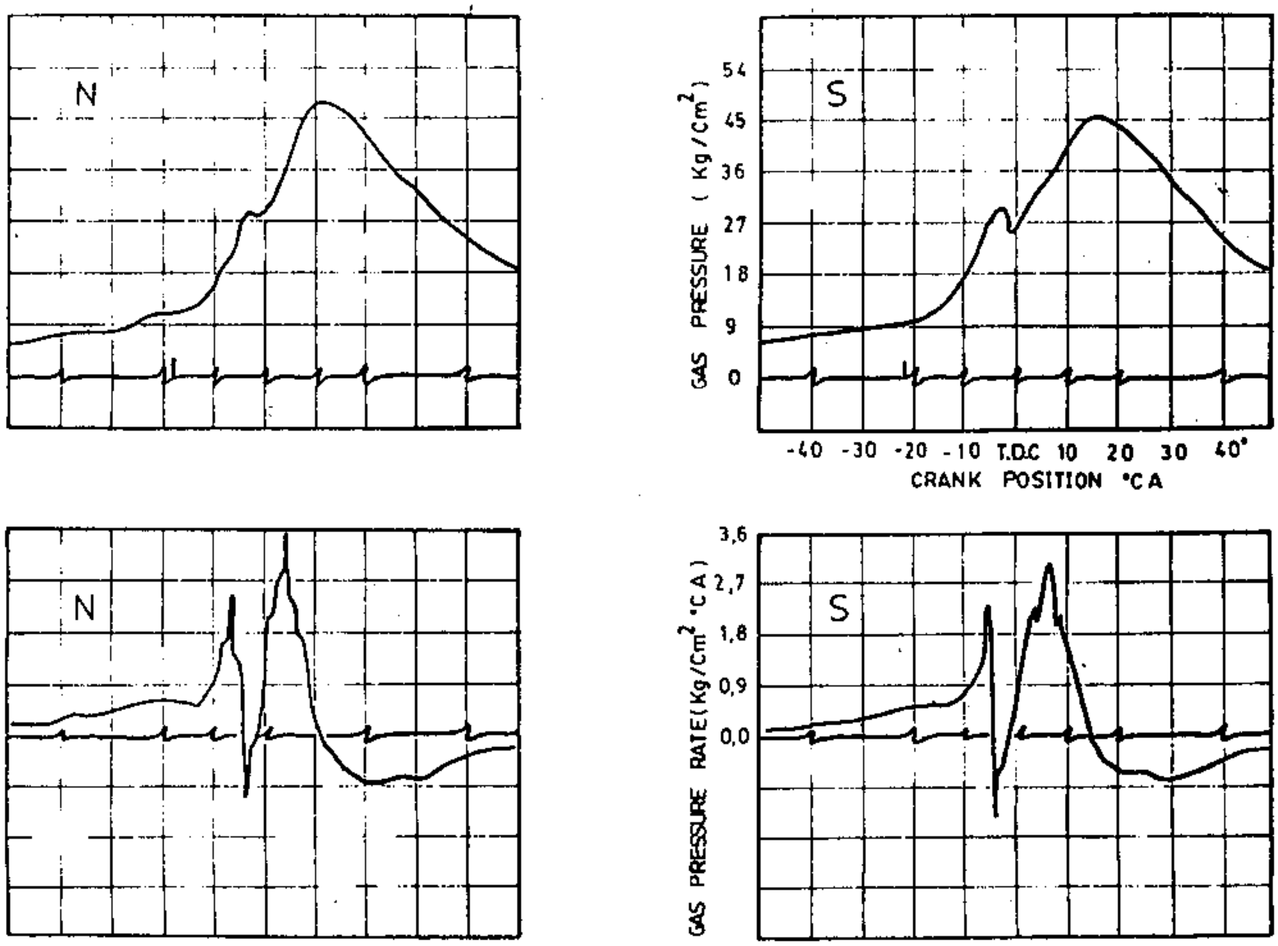


Fig.(12) GAS PRESSURE AND PRESSURE RATE (FULL THROTTLE – 1500 R PM – Φ = 1,0± 0,01 - M B T)

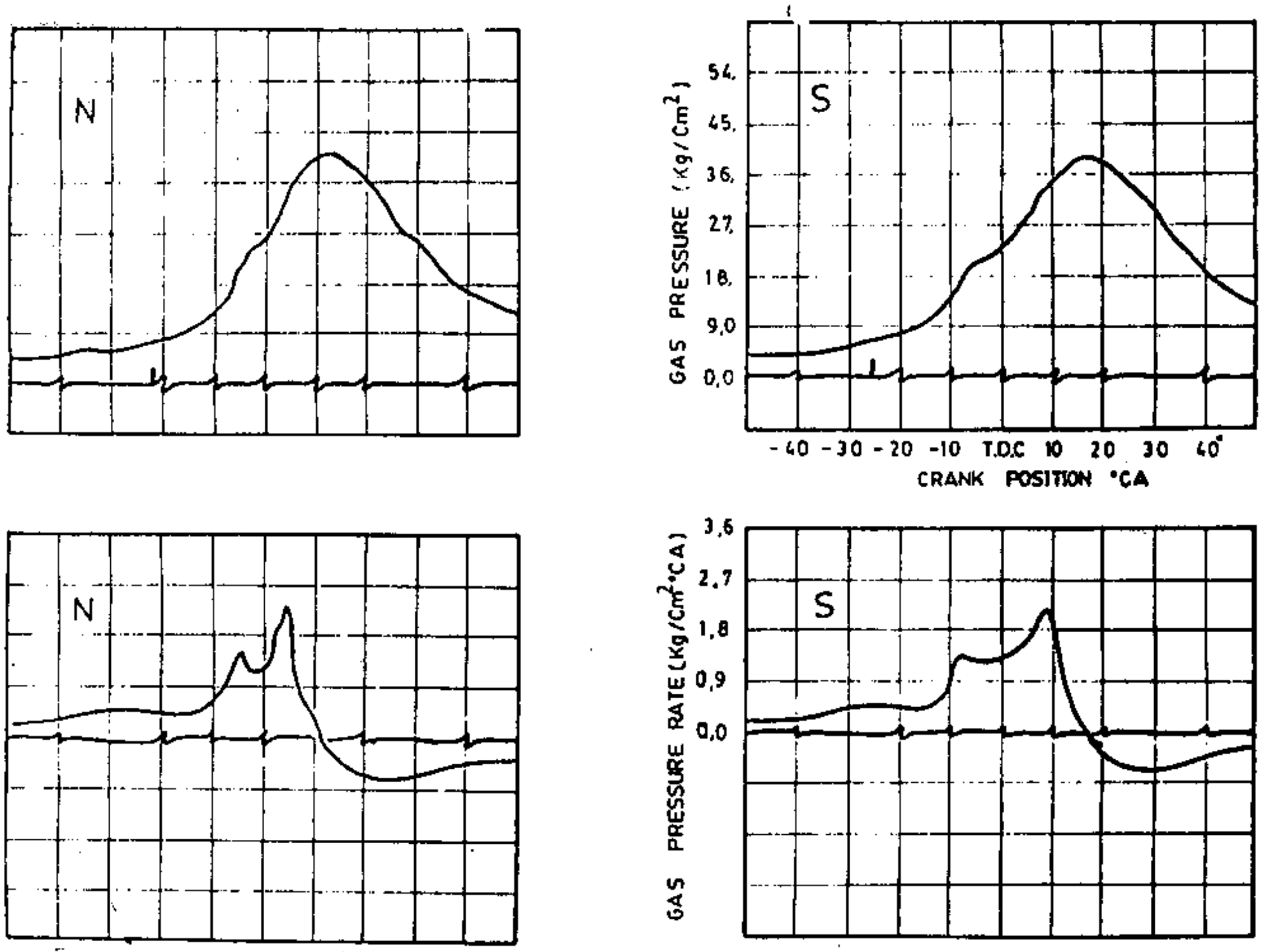


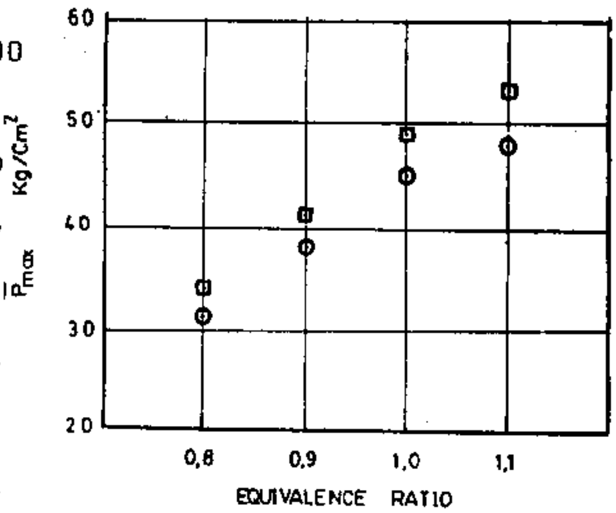
Fig (13) GAS PRESSURE AND PRESSURE RATE (FULL THROTTLE — 1500 RPM — $\phi = 0,9 \pm 0,01$ —MBT)

as

$$CV = \left(\sqrt{\frac{\sum (P_{max} - \bar{P}_{max})^2}{n}} / P_{max} \right) \times 100$$

It can be seen that fitting the engine with combination D results in supressing the cyclic variation particularly at lean equivalence ratio. This may be due to improving the mixture motion and its formation during flame development.

The mass fraction burnt is calculated from the measured pressure traces and thermodynamic analysis of the combustion process using the following relation (9).

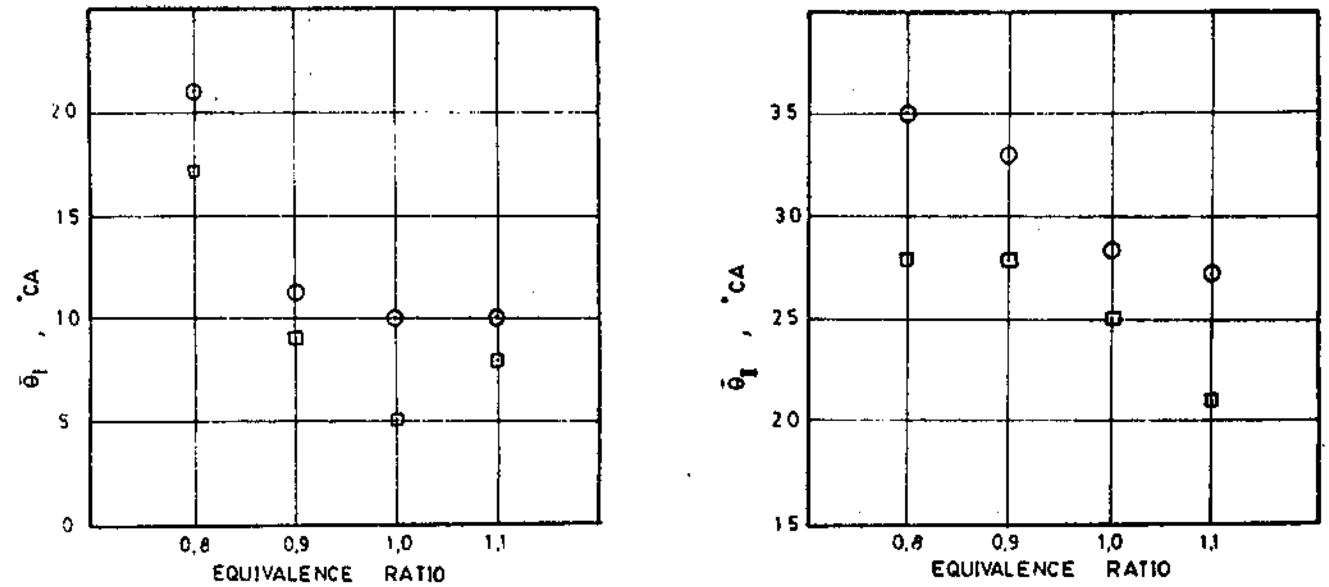


Fig(14)DEPENDENCE OF PEAK GAS PRESSUR UPON MIXTURE STRENGTH (1500 RPM, FULL THROTTLE OPEN , MBT)

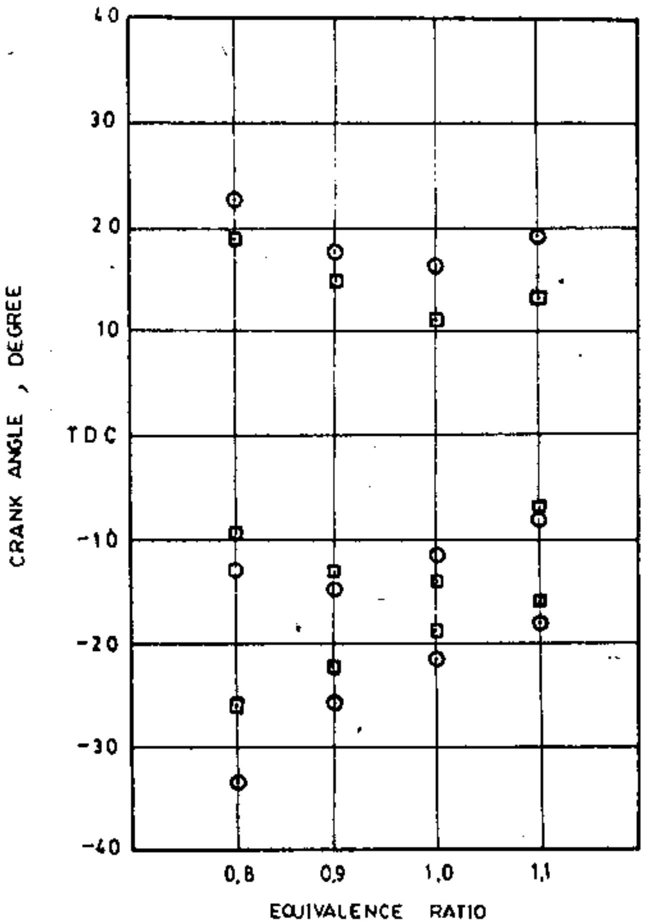
⊙ DIVIDED CHAMBER WITH STRAIGHT CONNECTING PASSAGE

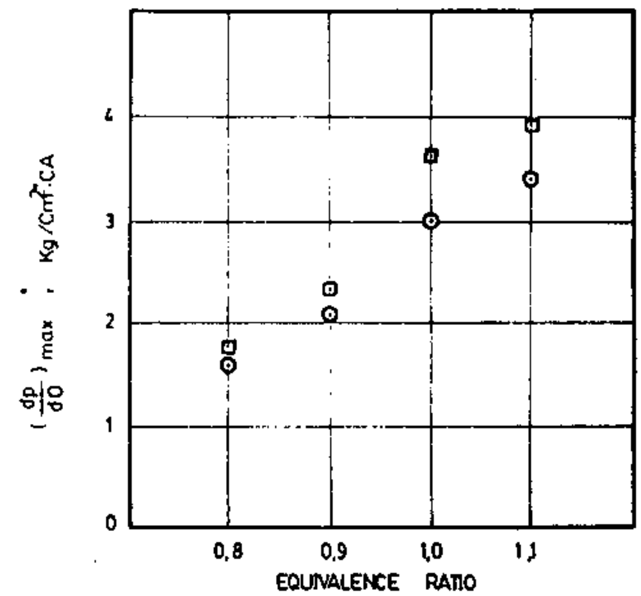
⊡ DIVIDED CHAMBER WITH CONV. DIV. CONNECTING PASSAGE

$$M_x = \frac{PV - P_oV_o + (\gamma_b - 1)W + (\gamma_b - \gamma_u)M\, C_{vu}\,(T_u - T_o)}{(\gamma_b - 1)(h_{fu} - h_{fb}) + (\gamma_b - \gamma_u)\, C_{vu}\, T_u}$$



Fig(15) DEPENDENCE OF INITIAL PHASE θ AND MAIN PHASE θ OF COMBUSTION UPON MIXTURE STRENGTH (1500 RPM, FULL THROTTLE OPEN , MBT)





Fig(17) DEPENDENCE OF MAXIMUM PRESSURE RATE UPON MIXTURE STRENGTH (1500 RPM , FULL THROTTLE OPEN , MBT)

Fig(16) INFLUENCE OF MIXTURE STRENGTH ON DURATION OF INITIAL MAIN PHASE OF COMBUSTION (1500 RPM, FULL THR OPEN, MBT)

⊙ DIVIDED CHAMBER WITH STRAIGHT CONNECTING PASSAGE
⊡ DIVIDED CHAMBER WITH CONV. DIV. CONNECTING PASSAGE

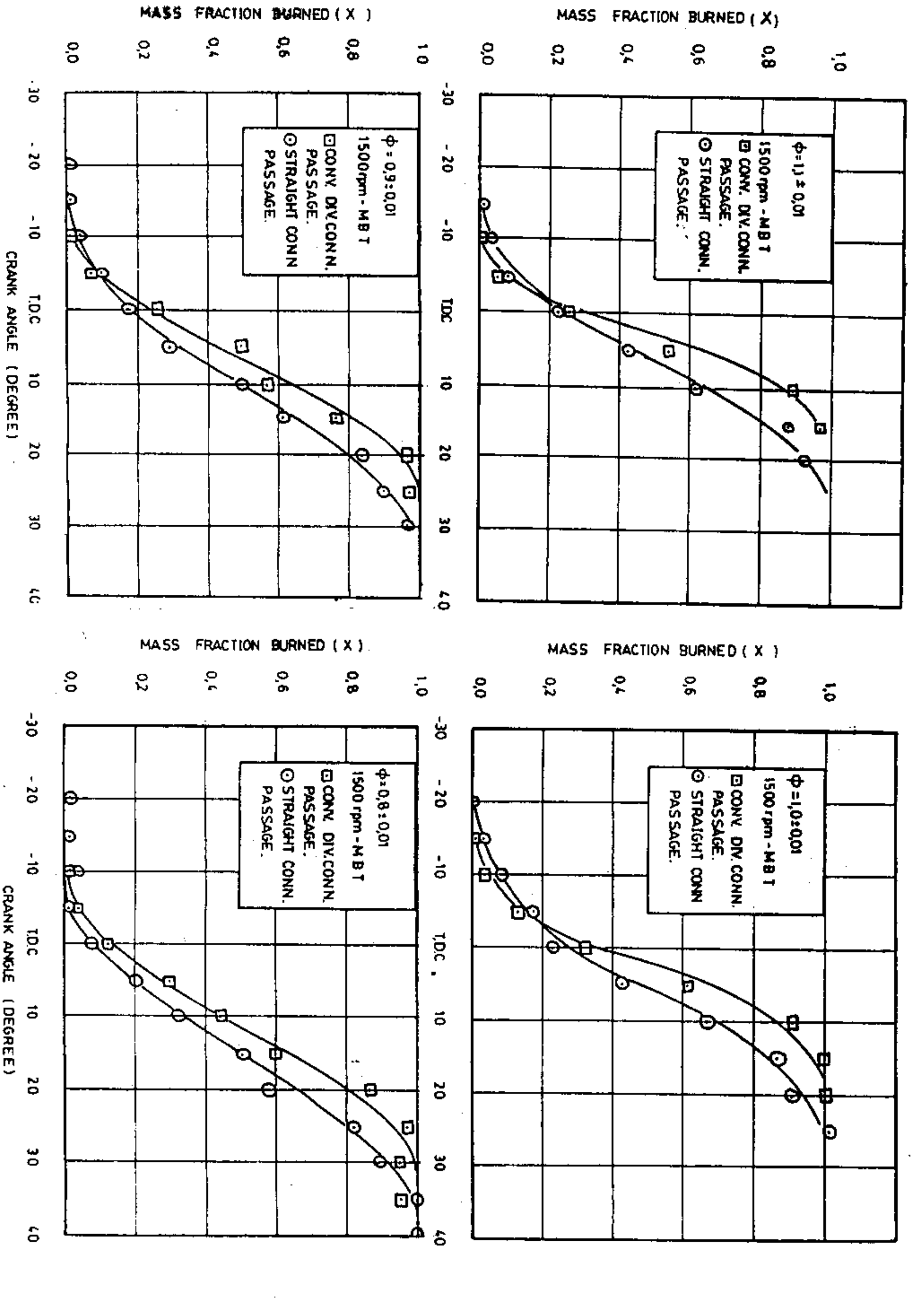


Fig (19-22) CALCULATED MASS FRACTION BURNED (X) AS FUNCTION OF CRANK ANGLE (θ)

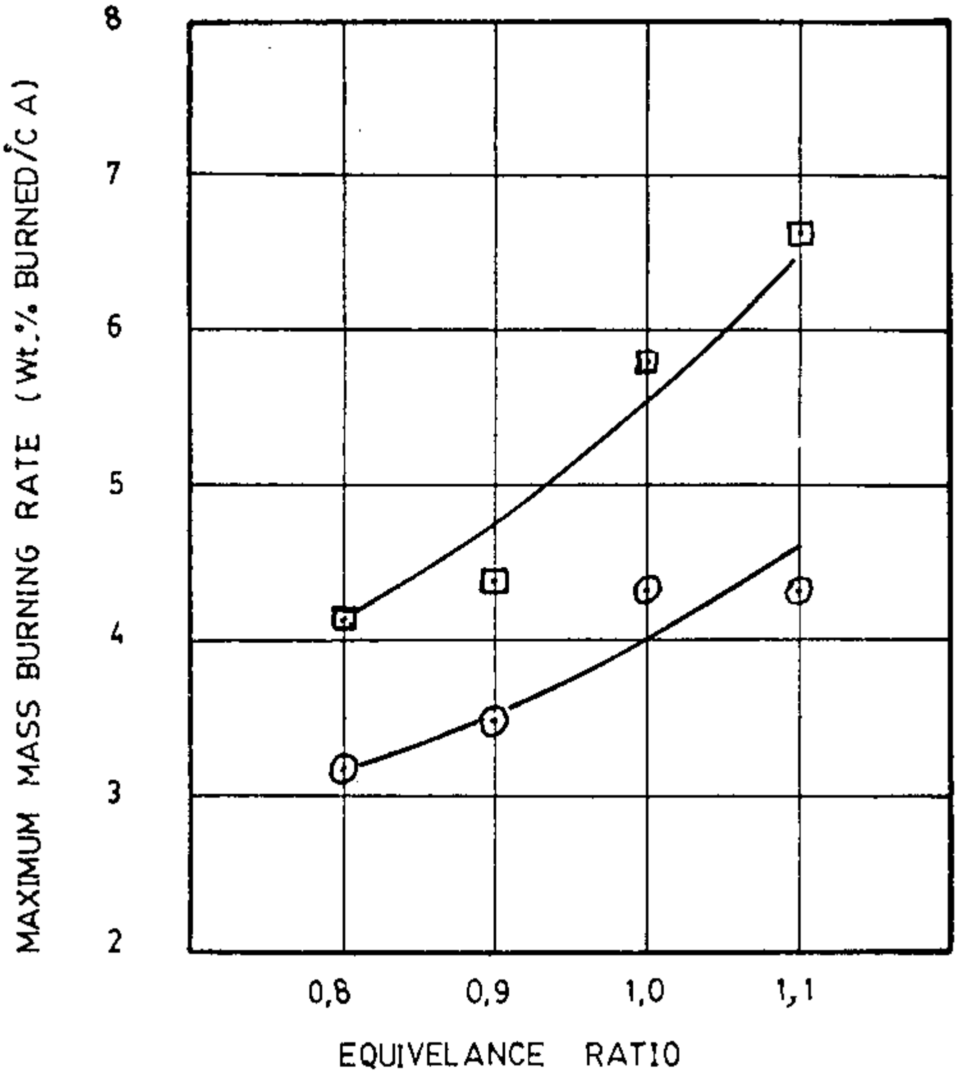


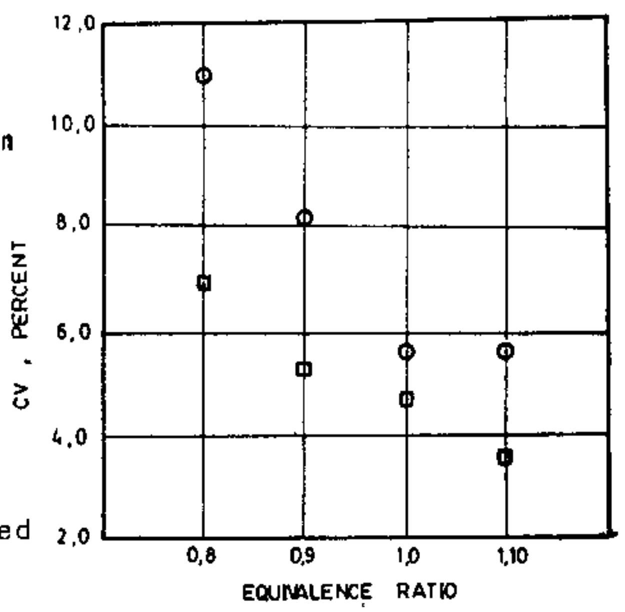
Fig (23) EFFECT OF MIXTURE STRENGTH ON MAXIMUM MASS BURNING RATE.

⊡ DIVIDED CHAMBER WITH CONV. DIV. CONNECTING PASSAGE.

⊙ DIVIDED CHAMBER WITH STRAIGHT CONNECTING PASSAGE.

Using the above relation in conjunction with pressure traces we can estimate the mass fraction burnt during the combustion cycle as shown in Figs. 19-23. During the analysis, heat transfer loss through the cylinder wall is neglected.

From these results it can be seen that, using the combination geometry of convergent divergent nozzle, the engine posses increased buring rate for all engine operation parameter considered over that the straight passage. This of course will results in increased the engine thermal efficiency and lower engine emissions.



Fig(18)DEPENDENCE OF PEAK PRESSURE COEFFICIENT OF VARIATION CV UPON MIXTURE STRENGTH (1500 RPM, FULL THROTTLE OPEN , MBT)

⊖ DIVIDED CHAMBER WITH STRAIGHT CONNECTING PASSAGE
⊡ DIVIDED CHAMBER WITH CONV. DIV. CONNECTING PASSAGE

CONCLUSIONS

Based on the experimental investigation, the results obtained concluded that the connecting passage shape and flame initiation point has a greater influence on performance and combustion characteristics of the torch ignition engine.

Both the extended spark plug electrode and convergent divergent nozzle connecting passage shape results in enhancing the combustion rate overall engine operation conditions and extended the effective lean burn limit. The increased buring rates with torch chamber burning are associated with increased turbulence levels in the combustion space. This effect the initial flame kernel development ingited by the spark in the torch chamber. The accelaration of combustion in the second phase is a function of the torch of flame issuing from the torch chamber into the main chamber through the connecting nozzle.

The benefits of accelaration of combustion results in improving the engine performance and lean buring ability. One might then also achieve lower emission levels. Thus, in particular it would be most useful to run the engine with a leaner mixture and higher compression ratio.

NOMENCLATURE AND ABBREVIATIONS

CV	Coefficient of the cyclic peak pressure variation
DC	Top dead center
P	Instantaneous gas pressure (N/m^2)
P_o	Gas pressure at instant of spark timing
V	Instantaneous cylinder volume (m^3)
V_o	Cylinder volume at instant of spark timing
T_o	Gas temperture at instant of spark timing, °K
T_u	Mean temperature of unburned gases, °K
M	Mass of gas inside the cylinder, Kg
Mx	Mass fraction burned
W	Work done on piston, N.m
Cvu	Constant volume specific heat of unburned mixture, N.m/Kg °K
h_f	Effective specific enthalpy of formation of gas mixture at absolute zero, N.m/Kg.
γ	Sepecific heats ratio.

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