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Pre-Chamber Stratified Charge Engine Combustion Studies

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MANY CLAIMS HAVE BEEN MADE recently about the stratified charge engine, including the pre-chamber engine, as a solution to the automobile emissions problem. In the past the pre-chamber engine was investigated for good fuel economy. The advent of emission control has revised interest in the pre-chamber engine utilizing a lean mixture as an alternative to the conventional engine.

This investigation was initiated to determine if the pre-chamber engine concept has inherent emission advantage over the conventional, to identify its disadvantages and to gain understanding of the pre-chamber engine combustion process.

Before looking at the test data a review of the pre-chamber engine operation is in order. Fig. 1 illustrates that engine combustion at lean air-fuel ratios (A/F) forms a minimum of carbon monoxide (CO) and nitrogen oxide (NO) and dis-

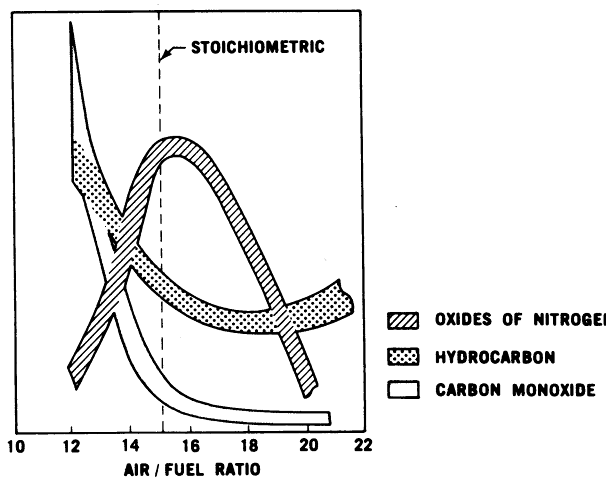

Fig. 1 - Emission concentration relationship to air/fuel ratio

ABSTRACT

Single-cylinder experiments were conducted with a 3-valve, carbureted pre-chamber stratified charge engine in comparison with a conventional engine. The pre-chamber engine operation is governed by many design and operating variables. This investigation was limited to determining the effect of overall air/fuel ratio, ignition timing and EGR on emissions and fuel economy at a single road load test condition.

It was found that, as for the conventional engine, these operating variables are also significant for the pre-chamber engine and that a compromise must be made between good fuel economy and low emissions. The main virtue of the pre-chamber engine was found to be the ability to operate at leaner overall air-fuel ratio. This resulted in lower nitrogen oxide (NO) emissions than the conventional engine without EGR. The unburned hydrocarbons (HC) were found to be higher for the pre-chamber engine up to the conventional engine lean misfire A/F ratio.

Exhaust gas introduced into the pre-chamber was found to reduce NO emissions significantly without a large corresponding increase in HC emissions as observed with the conventional engine.

Only at very low NO emissions with severely with severely retarded spark timing and/or high EGR rate did the pre-chamber engine show a fuel economy advantage over the conventional engine.

As the test program was limited to one load and speed, the results should not be construed to be typical of all modes of operation.

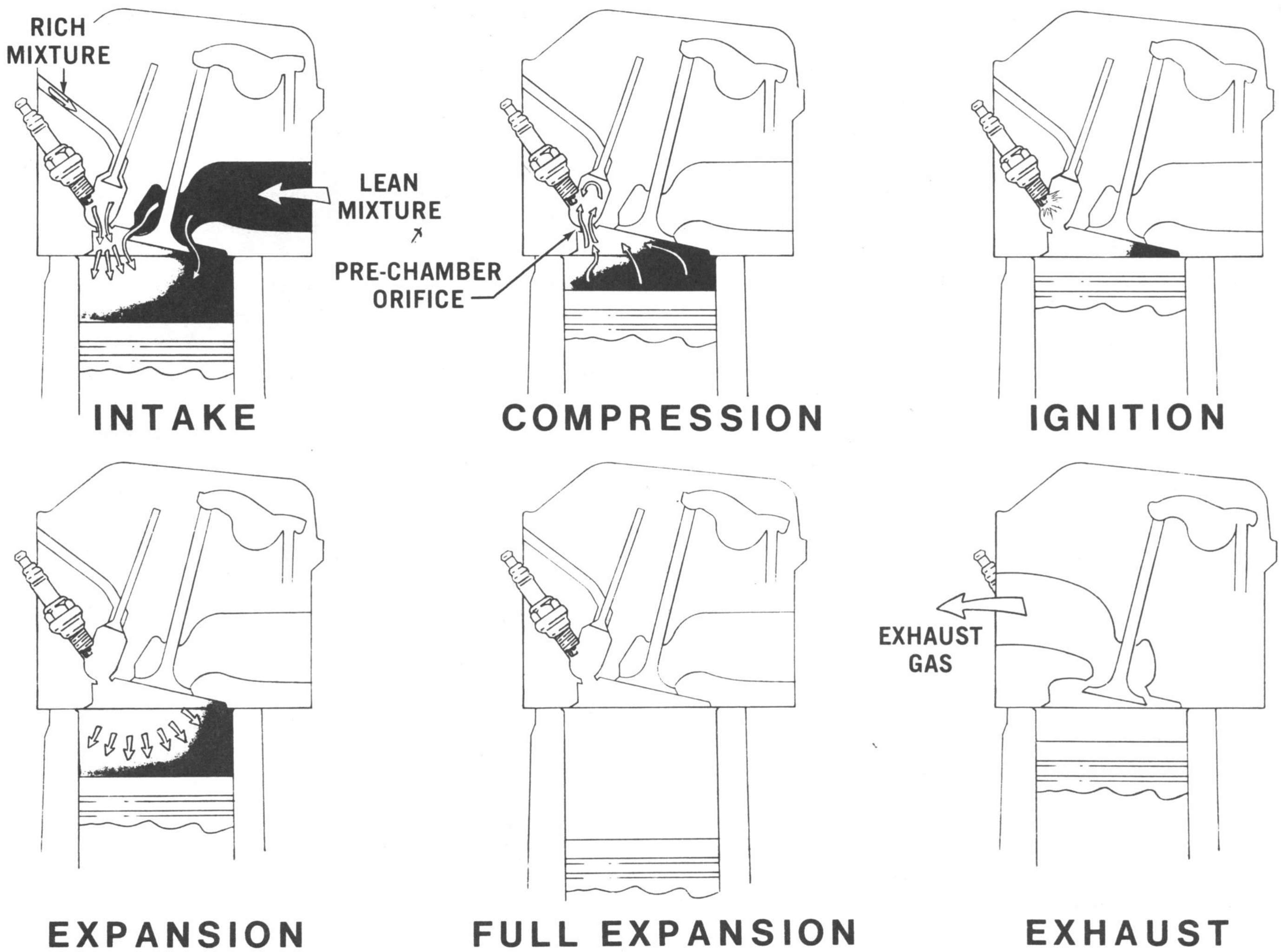

Fig. 2 - Pre-chamber stratified charge engine

charges low amounts of unburned hydrocarbon (HC). Therefore, from a steady-state emission standpoint lean combustion is very desirable. But the problem is that the lean mixture is not readily ignitable by conventional means and, if ignited in a conventional combustion chamber, the combustion process procedes at such a slow rate that the efficiency suffers drastically. Providing a richer mixture at the spark plug than in the rest of the combustion chamber eliminates the ignitability problem. In addition, some form of induced combustion turbulence would improve the combustion rate. The pre-chamber concept does all of this. A rich mixture is introduced into a separate enclosure (pre-chamber) basically for ignition. In this manner the rich mixture is mechanically prevented from mixing completely with the leaner portion. This not only maintains a richer mixture in the pre-chamber for a positive ignition, but also results in some degree of stratified combustion which is beneficial for low NO emissions (1).* An orifice between the rich pre-chamber and the lean main combustion chambers is used to induce the required amount of combustion turbulence to obtain efficient combustion of the lean mixture.

Fig. 2 describes the pre-chamber engine process. Note that the rich and lean mixtures do not stay in their respective

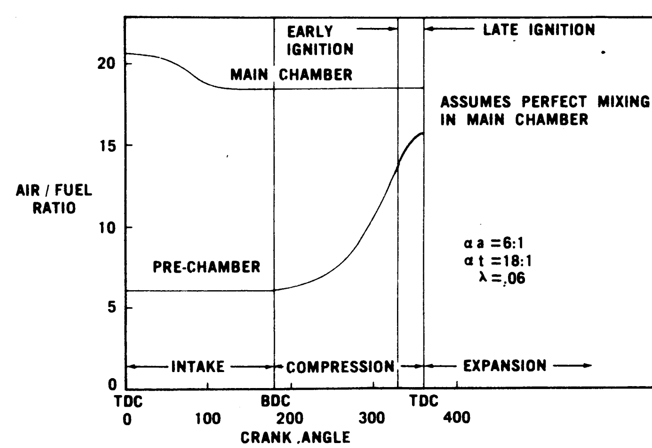

Fig. 3 - Calculated combustion chamber A/F versus crank angle

*Numbers in parentheses indicate References at end of paper.

chambers. On the intake stroke, if more than one pre-chamber volume of the rich mixture (α a) is inducted, then the excess will flow through the orifice and either partially or fully mix with the leaner main chamber mixture (α m). On the compression stroke the reverse takes place. The leaner main chamber mixture and some of the rich mixture is pushed into the pre-chamber and dilutes the original rich charge (α a). The mixture strength in the pre-chamber at the time of ignition (α ai) depends on the ignition timing, the degree of mixing taking place in the main chamber on the intake stroke and the degree of pre-chamber overfill. Fig. 3 provides the approximate α ai for a chosen ignition timing at the air flow (λ) and air-fuel ratio (α a, α m) conditions used for these tests. It is possible to obtain the same α ai by different combinations of α a, α m, λ and ignition timing the effect of which was not thoroughly investigated. It was the intent of this study to learn only about the operating parameters of α t, ignition timing and EGR in comparison with a conventional engine.

TEST EQUIPMENT AND INSTRUMENTATION

TEST ENGINE - Experiments were conducted with a 50 cu. in. single cylinder engine built on a 400 CID V-8 engine block with pistons only in the front two cylinders. Only the #1 cylinder was used for combustion while the #5 piston in the opposite bank served as counterbalance. The stock crankshaft was modified for best dynamic balance for this two cylinder operation.

A specially designed pre-chamber single cylinder head was used on the firing cylinder with the remaining cylinders blocked to complete the cooling system. A 1973 production 400 CID engine cylinder head was also tested to obtain comparative baseline data.

The head designs are illustrated in Figs. 4 and 5 with specifications covered in Appendix A. Fig. 4 shows the designed locational relationships between the pre-chamber, the orifice and the main chamber, the spark plug location relative to the orifice and the pre-chamber intake valve, and

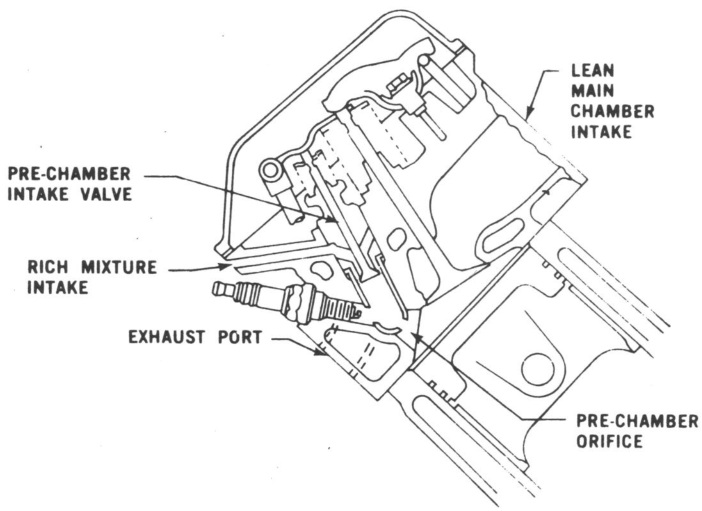

Fig. 4 - Pre-chamber cylinder head design configuration

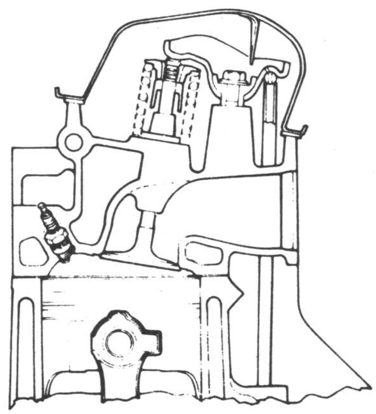

Fig. 5 - Conventional combustion chamber configuration

the proximity of the main chamber intake valve to the pre-chamber orifice. The spark plug was so located as to receive minimum of fuel impingement from the pre-chamber intake valve or the pre-chamber orifice. The pre-chamber cup was installed into a machined recess in the cylinder head with clearance calculated to give contact between cup and head for heat transfer when expanded at the higher operating temperatures. This design was to minimize spark plug fouling and to provide a heat sink for the vaporization of fuel particles in the fuel-rich charge entering the pre-chamber. The main chamber and its intake port were designed for minimum mixture turbulence prior to combustion.

Fig. 5 shows the open wedge design (squishless) conventional combustion chamber which was run for comparison purposes.

The exhaust system design permitted manual control of exhaust gas heat to the intake systems by changing the exhaust gas flow path as diagramed in Fig. 6. Figs. 6 and 7 illustrate the engine set-up and further describe the intake and exhaust system. The exhaust system was insulted with approximately .25 in. thick asbestos wrap to conserve heat energy for the intake charge pre-heating. The insulated exhaust system acted as a HC/CO reactor with the main exhaust pipe volume equal to one cylinder displacement.

A .94 inch diameter venturi Bendix 1V carburetor was used to meter fuel to the main combustion chamber and a specially constructed .25 inch diameter venturi carburetor was used for the pre-chamber with independent and manually controlled throttles and needle valve fuel adjustments for both carburetors.

Because of an unsolvable engine vibration problem which caused fuel handling difficulties the pre-chamber carburetor was isolated from the engine through a length of wire-reinforced silicone rubber hose, suspended over the engine and isolated from the engine mounting bed plate.

Indolene Clear fuel was used throughout the testing to minimize the combustion deposit formation.

The ignition system used was a conventional breaker-type automotive 12V system with a .035 inch gapped resistor spark plug.

INSTRUMENTATION - The engine was coupled to an electric dynamometer equipped with a strain gage load read-out.

The gas analysis equipment used included Beckman and Intertech analyzer for CO, CO_2, O_2 and HC analysis and a chemiluminescent NO analyzer. The HC readings were taken as N-hexane. The exhaust was sampled downstream from a mixing chamber to minimize measurement errors resulting from the single cylinder exhaust gas inhomogeneity.

The carburetor air flow measurements were made with Meriam laminar element flow meters of 20 and 4 cfm flow capacity.

Fuel flow rates were calculated from burette timed samples. The burette fuel measuring system was designed to maintain a constant pressure fuel supply during measurements.

Combustion pressure measurements were made using a tranducer installed in the main combustion chamber and displayed and analyzed with a Digital Processing Oscilloscope.

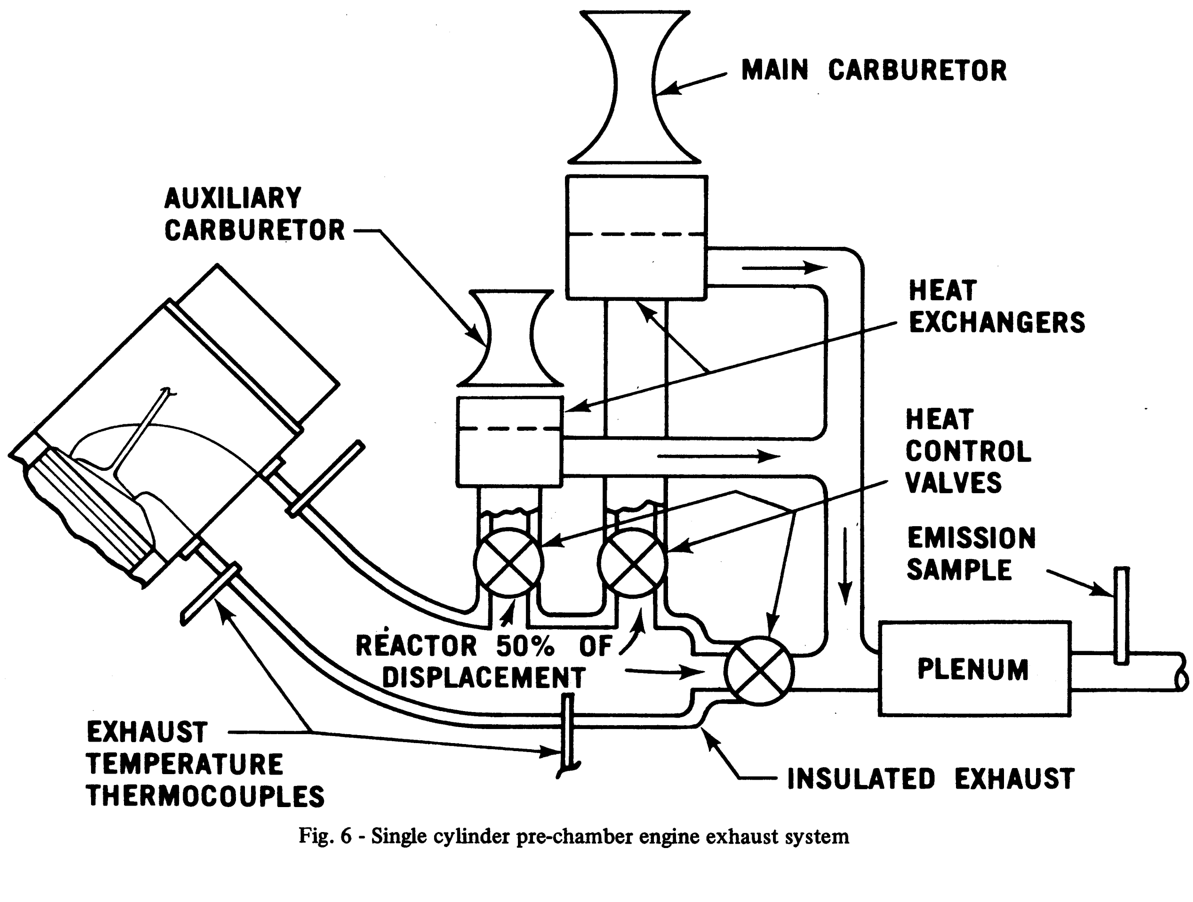

Fig. 6 - Single cylinder pre-chamber engine exhaust system

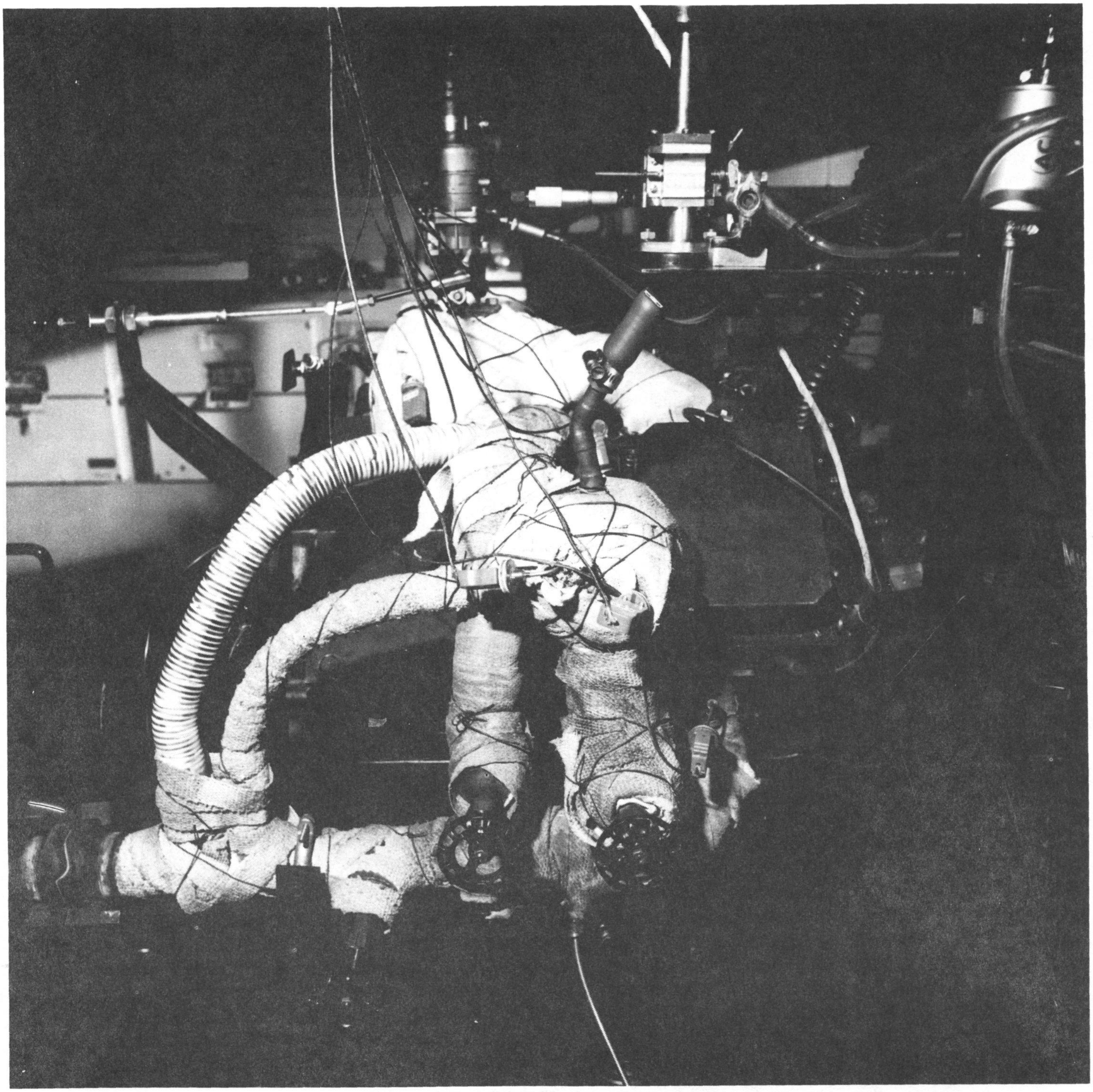

Fig. 7

A more detailed description of the instrumentation used is included in Appendix D.

TEST PROCEDURE

Except as indicated, the investigation was conducted at steady state 1500 rpm and 60 psi IMEP. This test condition can be related to a light 40 mph acceleration of a standard size vehicle, a condition which frequently occurs on a CVS vehicle emission test. Both the pre-chamber head and the conventional head were run on the same basic engine block at comparable conditions.

Engine operating conditions were manually adjusted and only one condition varied at a time. The operating conditions investigated were as follows:

1. The overall air/fuel ratio (α t)
2. Spark advance relative to minimum advance required for best torque (MBT)
3. Exhaust temperature effect on HC/CO emissions
4. Exhaust Gas Recirculation (EGR)

A pre-chamber volume with 8.9% of the total clearance volume and a 12mm diameter circular orifice was used for all pre-chamber engine tests.

To describe the true effects at a given constant operating condition the emissions are reported on a specific mass basis as described in Appendix B.

The overall air/fuel ratio was obtained by exhaust analysis using the Spindt Method (3) and by fuel and air flow measurements. The two methods were required to agree within .5 A/F. The overall air/fuel ratio data reported is based on exhaust analysis. The reported pre-chamber and main chamber α a and α m are, of necessity, calculated from fuel and air flow measurements.

Intake charge mixture temperatures were maintained in the range of 170-190°F for the main intake system (and the conventional engine) and 370-400°F for the pre-chamber intake charge measured near the intake valves.

The exhaust backpressure was maintained at a normal test cell level minimum and did not exceed 1.0 inch Hg measured at the exhaust port outlet.

The exhaust temperatures were measured at the exhaust port outlet and/or in the reactor as shown in Fig. 6.

Determination of burn time was accomplished using the method of Rassweiler and Withrow (8). Pressure rise due to combustion is analogous to the charge burning. In brief, the difference between observed pressure change and calculated pressure change due to piston motion (Δ Pc) is calculated over a small time interval and corrected to a constant volume. The fraction burned at any point during combustion is the sum of the Δ Pc from ignition to that point divided by the total sum of Δ Pc from ignition to the point where combustion is complete. Complete combustion has occurred when the observed pressure change equals the change due to piston motion.

The exact points where significant burning begins and burning ends are difficult to establish reliably. Therefore, the time to burn 10 to 90% of the charge was taken to compare the effect of operating parameters on ignition lag and burn time. Each burn and lag time calculation was made using the average of 512 consecutive p-t traces.

AIR-FUEL RATIO REQUIREMENTS

To obtain the effects of varied overall air-fuel ratio (α t) for the pre-chamber engine, as compared to a conventional combustion chamber, tests were conducted with α t leaned from 13:1 to lean limit at MBT spark advance where a substantial HC increase was observed. Other test conditions were maintained constant as shown in Table 1. The pre-chamber air-fuel ratio (α a) was maintained at a constant 6:1 and the mass air flow ratio of pre-chamber-to-main chamber (λ) was held constant at .06 which results in approximately 5 pre-chamber volumes of rich charge overflowing into the main chamber on the intake stroke. The overall air-fuel ratio was adjusted by varying the main intake A/F.

The most noticeable difference in the results is the lower NO emissions at A/F richer than 20:1 for the pre-chamber as shown in Fig. 8. The lower NO is thought to be due to stratified combustion. With the combustion initiated in the rich mixture zone of the pre-chamber, the rich condition results in low peak temperatures in the pre-chamber and a lack of excess oxygen serving to minimize NO formation. The bulk of the main combustion chamber mixture is lean, likewise producing minimum of NO with only a fraction of the total mixture consumed at high NO forming air/fuel ratios.

Table 1 - Air/Fuel Ratio Requirement Test Condition for Figs. 8 - 14

Pre-Chamber Head (Ref. Test #22-07)	Conventional Head (Ref. Test #12-06)
1500 RPM	1500 RPM
60 IMEP	60 IMEP
6:1 α a	8.6 CR
.06 λ	MBT spark
MBT spark	No EGR
12 mm dia. orifice	
8.9% pre-chamber volume	
8.5 CR	
No EGR	

Shown in Fig. 9, pre-chamber operation is accompanied by approximately a 3% fuel consumption penalty over the conventional chamber in the operating range of 16:1 to 19:1 α t. After 20:1 A/F combustion in the conventional chamber deteriorates rapidly due to misfires.

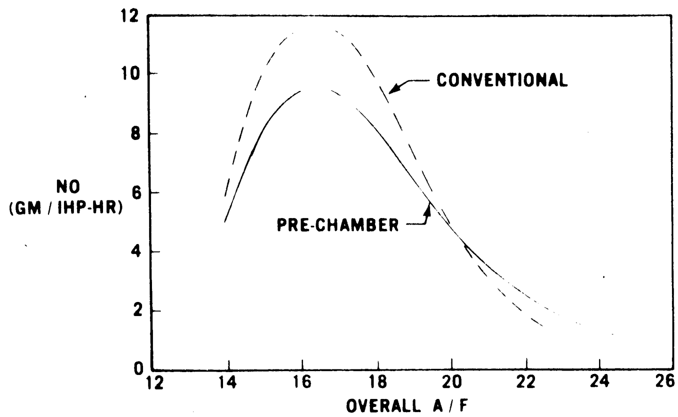

Fig. 8 - No emissions versus overall A/F at MBT ignition timing

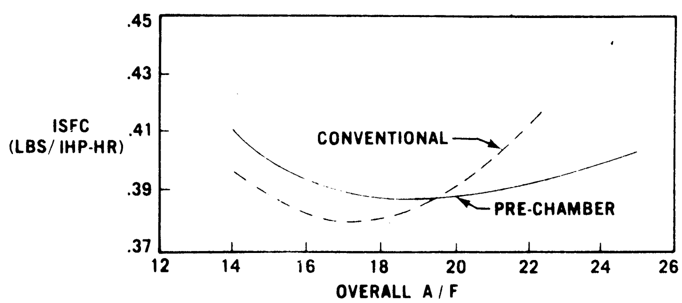

Fig. 9 - Fuel consumption versus overall A/F at MBT ignition timing

The difference in MBT ignition timing requirement for the conventional and pre-chamber is shown in Fig. 10. The difference is attributed mainly to increased ignition lag time for the conventional chamber. Calculations made from combustion chamber pressure measurements show in Fig. 11 that the ignition lag increases rapidly with lean A/F ratio at igni-

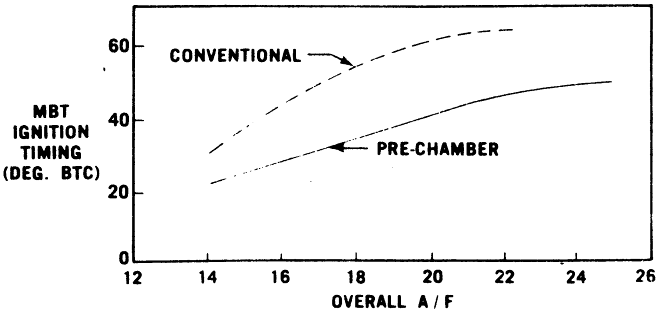

Fig. 10 - MBT ignition timing requirements

tion although the burn time may be less. The higher spark advance is required for the conventional chamber because of low mixture turbulence in the vicinity of the spark plug and a relatively leaner A/F ratio which results in high ignition lag time. The burn time for the pre-chamber condition was longer by approximately 10% throughout the A/F ratio range but may decrease with the use of smaller orifice diameter or a larger pre-chamber volume or both.

The difference in HC emissions shown in Fig. 12 at A/F ratio below 19:1 is mainly due to the difference in exhaust temperature indicated in Fig. 13 and a 20% higher combustion chamber surface-to-volume ratio for the pre-chamber engine. At A/F over 19:1, the conventional chamber HC emissions increase rapidly, indicating misfires.

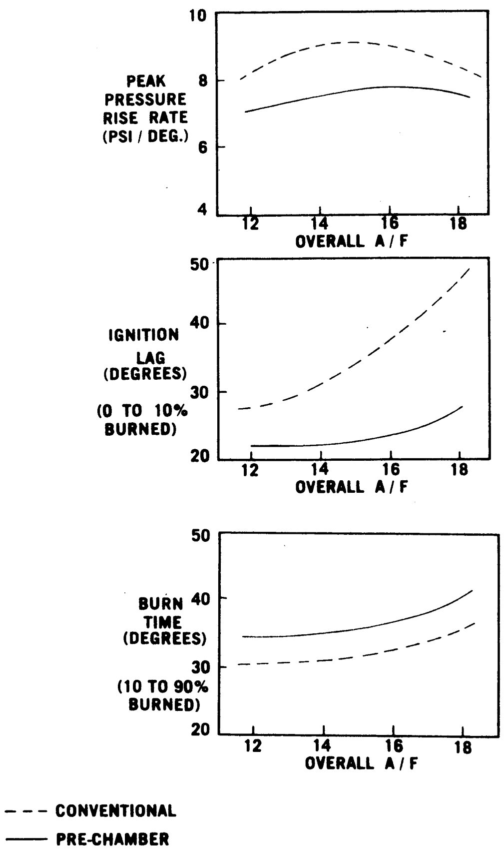

Fig. 11 - Combustion characteristics versus overall A/F at MBT ignition timing

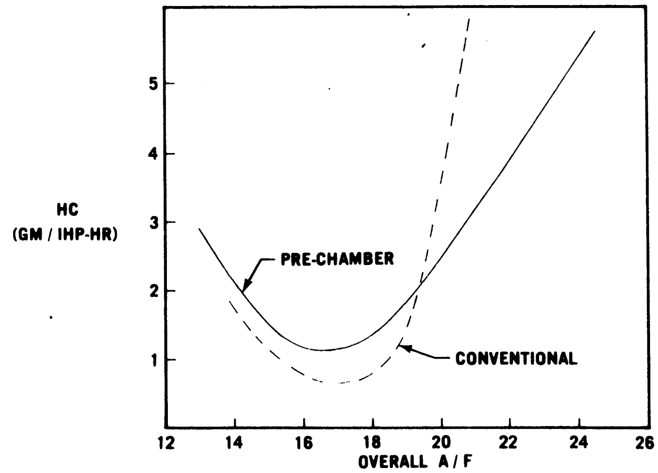

Fig. 12 - HC emissions versus overall A/F at MBT ignition timing

The lack of significant differences in CO emissions in Fig. 14 is mainly due to a lack of significant CO conversion in the exhaust at the exhaust temperatures observed. The increase in CO above 19:1 A/F ratio is a result of increased mass flow due to the leaner operation.

IGNITION TIMING

For a conventional engine retarded ignition timing has proved to be the one variable which significantly reduces HC, CO and NO emissions. This investigation was conducted to determine its significance for the pre-chamber engine. It should be noted that any change in ignition timing for the pre-chamber engine also is accompanied by a change in the pre-chamber air-fuel ratio at the time of ignition (α ai). Theoretically, at constant α a and α m, the α ai will become leaner with retarded ignition timing. Referring again to Fig. 3, a decrease in spark advance by 20 degrees from MBT (from 25° to 5° BTC) will only increase α ai by about one air-fuel ratio, (assuming perfect mixing). The significance of the change in α ai alone was not investigated.

To determine the total effect, the ignition timing was incrementally decreased from a maximum advance of MBT + 5° for both the pre-chamber and the conventional engine. All other engine conditions were maintained constant as shown in Table 2. Overall air-fuel ratios of 18:1 for the pre-chamber and 16:1 for the conventional chamber were chosen as typical for this road load condition.

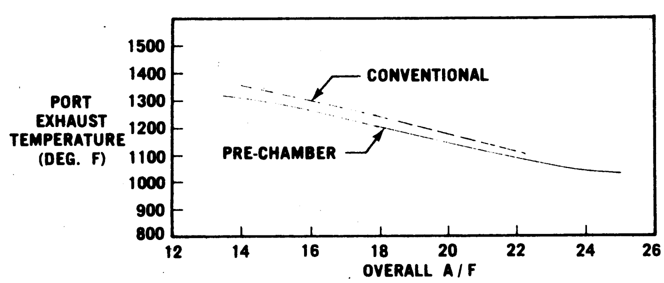

Fig. 13 - Port exhaust temperature versus overall A/F at MBT ignition timing

Table 2 - Ignition Timing Requirement Test Conditions for Figs. 15 - 22

Pre-Chamber Head	Conventional Head
(Ref. Test #22-07)	(Ref. Test #12-06)
1500 RPM	1500 RPM
60 IMEP	60 IMEP
6:1 αa	8.6 CR
18.1 αt	16:1 A/F
20.5:1 αm	No EGR
.06 λ	
8.5 CR	
12mm dia. orifice	
8.9% pre-chamber volume	
No EGR	

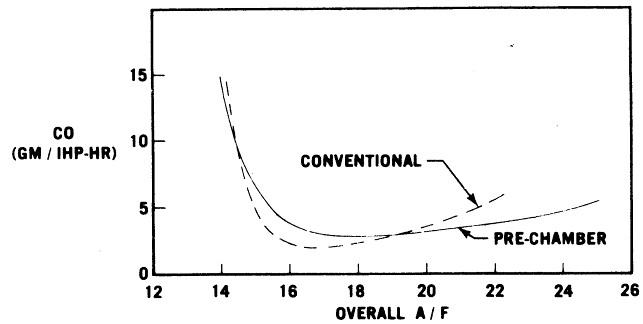

Fig. 14 - CO emissions versus overall A/F at MBT ignition timing

The results shown in Figs. 15 and 16 point out that ignition timing is critical on NO emissions for both the pre-chamber and conventional combustion chamber; but after retarding a few degrees from MBT, a rather large fuel consumption penalty is received. Because of the initial NO emission advantage of the pre-chamber engine, the conventional engine must be run more retarded from MBT to obtain the same NO emission level as the pre-chamber.

The HC emissions in Fig. 17 show a large decrease with spark retarded for both engines. This is due to increase in exhaust temperature shown in Fig. 19 which results in better post reaction. The pre-chamber has 30% higher HC emissions at MBT. At about 30° retarded, both engines have about equal HC because the exhaust temperatures approach about 1400°F with good post reaction.

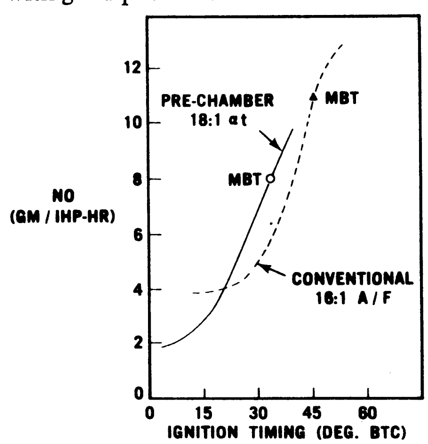

Fig. 15 - No emissions versus ignition timing

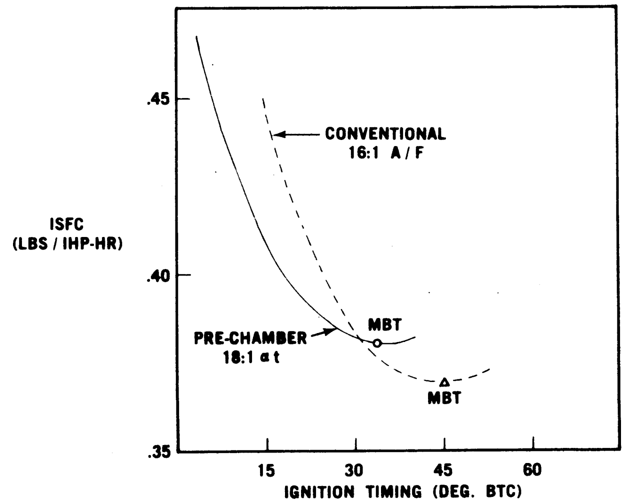

Fig. 16 - ISFC versus ignition timing

The CO emissions increase for both engines to about 10 degrees retarded ignition and decrease when retarded further. This phenomenon, shown in Fig. 18, is thought to be due to HC to CO conversion in the reactor with the temperature insufficiently high to complete the conversion to CO_2. The CO peaks for both are at about the same 1220°F reactor temperature, shown in Fig. 19. The CO increase was studied separately and will be discussed later in the text.

Fig. 20 and 21 illustrate the NO/ISFC and HC/ISFC trade-offs with spark advance. It shows that in order to get a low NO emissions with retarded spark, a large sacrifice in fuel economy must be made. At the higher NO level the conventional chamber has a fuel economy advantage. This advantage is lost, however, where spark is retarded to obtain NO levels of 4 gm/IHP-HR or less. At the 4 gm NO level the conventional engine specific fuel consumption begins to deteriorate very rapidly. By retarding spark alone, the conventional engine cannot reach the low NO level obtainable with the pre-chamber. There is a similar trade-off in HC emission control except that the pre-chamber engine at the same NO level condition shows higher HC emissions than the conventional.

The degree that combustion characteristics are affected by ignition timing is shown in Fig. 22. The peak pressures are almost identical for the pre-chamber and conventional engine

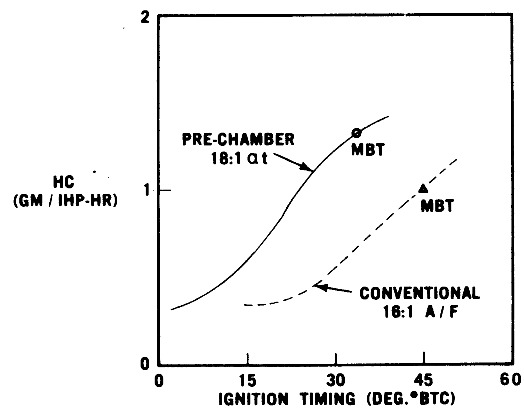

Fig. 17 - HC versus ignition timing

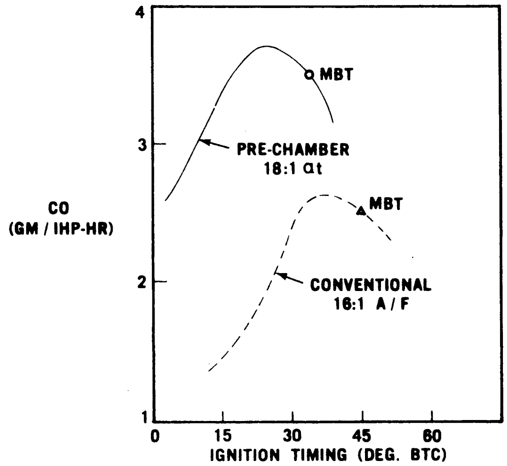

Fig. 18 - CO versus ignition timing

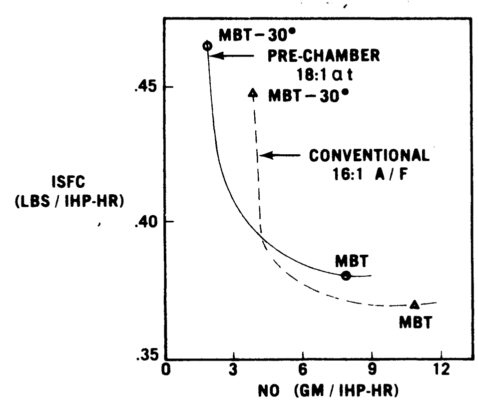

Fig. 20 - ISFC versus number at variable ignition timing

and decrease in the same proportion with retarded spark.

The peak pressure-rise-rate which is an indicator of the relative rate of burn is slightly lower (11%) for the pre-chamber engine measured in the main combustion chamber. The 10% to 90% burn time is calculated to be 17% longer for the pre-chamber engine at MBT (41 degrees vs. 34 degrees) and increases at a slower rate with retarded spark than the conventional.

The ignition lag time at MBT is 43% lower for the pre-chamber than the conventional (28 vs. 48 degrees). The reason for this difference is due to the difference in A/F ratio and mixture turbulence near the spark gap at the time of ignition as explained under A/F effects. At equal spark retard the difference in lags is maintained. Because of this ignition lag difference MBT ignition timing cannot be used as an indicator of relative burn rates between stratified and conventional combustion chambers. Instead, the pressure-rise-rate at MBT spark is a better indicator of the relative burn rates between different engines.

EXHAUST SYSTEM HC/CO CONVERSION

Because the tailpipe HC/CO emissions are greatly affected by the degree of post reaction in the exhaust system, it was of interest to measure the HC/CO emission concentrations leaving the exhaust port. To prevent the exhaust manifold reaction, the exhaust gas was cooled to 800°F with N_2 gas injected at the port. This procedure was used because an abnormally high HC concentration is measured at the port as reported by other investigators (4, 2). Therefore, a simple procedure of sampling and comparing HC and CO concentration at port and after the exhaust manifold could not be used.

It was assumed that no change in mass NO emissions would result from the N_2 injection and the NO concentration was used as a tracer gas to determine the degree the exhaust sample was diluted for the correction of measured HC/CO concentration.

The engine was run at three constant exhaust mass flow rates using different speeds and ignition timing to obtain an exhaust temperature range of 1000° to 1700°F before cooling with N_2.

To determine the amount of HC/CO conversion taking place at different reactor temperatures, the mass emissions at 800°F reactor temperature were compared to emissions without cooling. Fig. 23 illustrates the changes taking place in the exhaust system at different exhaust temperatures. Only small amounts of HC are converted below 1000°F with conversion reaching 50% at 1300°F. Surprisingly, the CO is shown to increase in the exhaust manifold with increasing

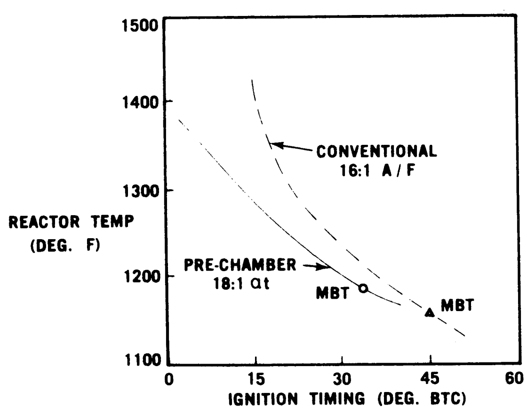

Fig. 19 - Reactor temperature versus ignition timing

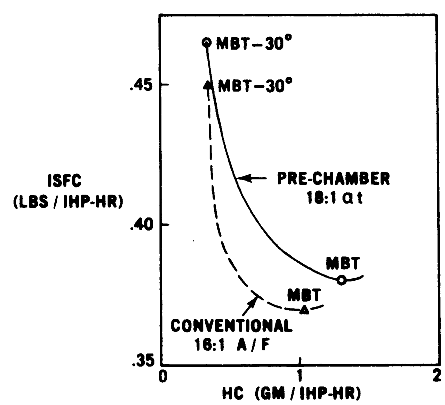

Fig. 21 - ISFC versus HC at variable ignition timing

exhaust temperature between 800°F and 1400°F with maximum increase at 1200°F. The data suggest that HC is converted to CO in the exhaust system faster than CO is converted to CO_2 at temperatures below 1200°F. Thus, until a sufficiently high exhaust temperature is reached for adequate conversion of CO, an increase in tailpipe CO emissions will take place. This means that when working with lean burn engine emissions, one must be careful not to create a CO problem when post oxidizing HC in the exhaust system. CO increases to approximately 1200°F exhaust temperature and then decreases with a break-even point at approximately 1400°F.

A rough accounting of HC converted and the increase in CO showed good correlation up to 1300°F after which CO conversion becomes significant.

Table 3 makes comparison of port emission for the conventional and the pre-chamber engine at the same overall A/F. It is evident that the pre-chamber engine emits exhaust with 30 to 40% higher HC emission concentration from the combustion chamber at the same test conditions with nearly the same exhaust temperature. The 22% larger combustion chamber surface-to-volume ratio (S/V) is thought to be the main contributor to the higher HC emissions.

EXHAUST GAS RECIRCULATION

The pre-chamber combustion process initiates in a fuel rich mixture in the pre-chamber and then passes through the orifice into the main combustion chamber which may or may not have a uniform air/fuel mixture. If some stratification does exist in the main chamber, it is likely that a richer mixture will be found at the pre-chamber orifice. This will be consumed next after the pre-chamber with ignition finally passing into the lean region. At the air/fuel ratios used for this investigation (a a = 6:1 and a m = 20.5:1), MBT ignition occurred theoretically at about a ai = 14:1 assuming perfect mixing in the main chamber.

It is apparent that the early combustion taking place at rich A/F contributes heavily to the overall NO emissions. It seemed, therefore, that EGR would be effective in reducing NO without a significant increase in HC emissions (7) if introduced into this rich initial burning charge.

This investigation was conducted at constant 1500 rpm and 60 IMEP load with 18:1 a t for the pre-chamber engine. For comparison, the conventional head was tested richer, at 16:1 A/F, because this has been shown to be typical when EGR is used.

The EGR and ignition timing were both variable. EGR was incrementally added to the intake charge up to the maximum rate supporting good combustion at MBT. At each EGR setting, ignition timing was varied from MBT +5 to MBT -30 degrees.

For the pre-chamber head EGR was first introduced only in the main combustion chamber intake system. In a second test, the EGR was introduced only into the pre-chamber intake manifold. The effect of EGR concurrently in both manifolds was not investigated. EGR for the conventional head was also introduced into the intake manifold.

The EGR rate was calculated from CO_2 measurements as described in Appendix C and is defined as follows: % EGR = (CO_2 intake - CO_2 background) / (CO_2 exhaust - CO_2 intake).

For the test conditions in Table 4, the effects of EGR at MBT spark are shown in Figs. 24-33. As indicated, the maximum EGR rate for the pre-chamber was approximately 7%, for the main combustion chamber 20% and 17% for the conventional. The latter two were maximum values because of erratic combustion and a large increase in HC emissions shown in Fig. 27. At 7% pre-chamber EGR, an HC increase was not observed, but the engine load surged sufficiently to be declared a maximum rate. This surging was probably due to the extensive dilution of the pre-chamber mixture with exhaust gas causing large variations in the ignition lag time. The 7% maximum overall rate shown is a 123% EGR rate for

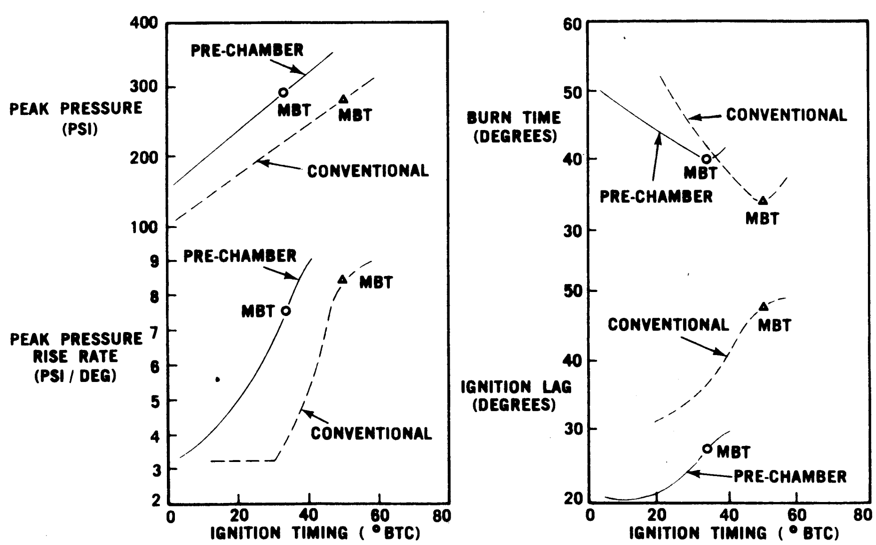

Fig. 22 - Combustion characteristics versus ignition timing

Table 3 - Comparison of Port Emissions from Pre-Chamber Head and Conventional Head HC/CO Measured After Plenum With N_2 Cooling

Conditions: αt = 18:1, λ = .06, αa = 6:1, MBT Ignition Timing

	PRE-CHAMBER				CONVENTIONAL			
RPM	IMEP psi	Exh. Temp. °F	HC ppm	CO Mole %	IMEP psi	Exh. Temp. °F	HC ppm	CO Mole %
1000	55	1110	352	.11	55	1094	199	.08
1500	67	1263	262	.11	66	1272	154	.11
2000	79	1402	193	.13	80	1363	131	.08

the pre-chamber mixture. This means that the mixture inducted in and through the pre-chamber consisted of one part of 6:1 air/fuel mixture and 1:23 parts of exhaust gas. The actual dilution of pre-chamber charge at the time of ignition was not measured in this study but from other reported data (7) it is estimated that gas/fuel ratios in the range of 17-19:1 existed in the pre-chamber at ignition.

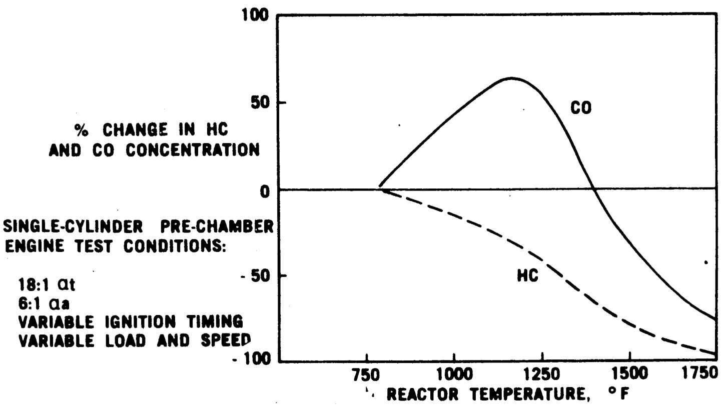

Fig. 23 - HC/CO conversion rate in exhaust reactor

Fig. 24 illustrates the degree that ignition timing must be advanced for MBT condition with increasing EGR at a constant air/fuel ratio. Like trends are shown for all three tests—increasing MBT spark requirement with EGR due to increased burn time and increased ignition lag. The pre-chamber EGR condition shows an exceptionally rapid increase in MBT requirement. Calculations from pressure-time measurements taken for the combustion cycle show that the major cause for the increase in the pre-chamber MBT spark requirement is ignition lag with burn duration remaining relatively constant. Fig. 25 shows the calculated ignition lag and the burn duration for 10% to 90% of the combustion chamber mass. The high ignition lag measured indicates a large amount of EGR present at ignition resulting in a lean gas/fuel ratio.

Comparison of reactor temperatures, presented in Fig. 26, show basically that at this test condition the reactor for the conventional engine was doing a better job, increasing in temperature as higher HC concentrations are received at sufficiently high temperature for HC to CO conversion. HC and CO data in Figs. 27 and 28 also show this and agree with the earlier discussion of data in Fig. 23.

The HC emissions increase rapidly for main chamber EGR and the conventional chamber with EGR due to incomplete

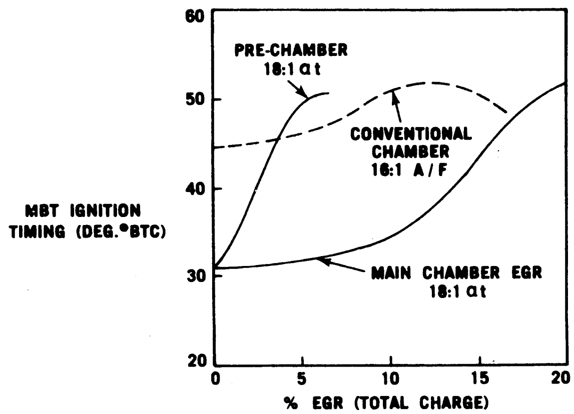

Fig. 24 - MBT spark timing requirements versus % EGR

Table 4 - EGR Evaluation (Ref. Test #22-07) Test Conditions for Figs. 24 - 37

Pre-Chamber Head	Conventional Head
1500 RPM	1500 RPM
60 IMEP	60 IMEP
6:1 αa	8.6 CR
20.5:1 αm	16:1 A/F
18:1 αt	
.06 λ	
8.5 CR	
12mm dia. orifice	
8.9% pre-chamber volume	

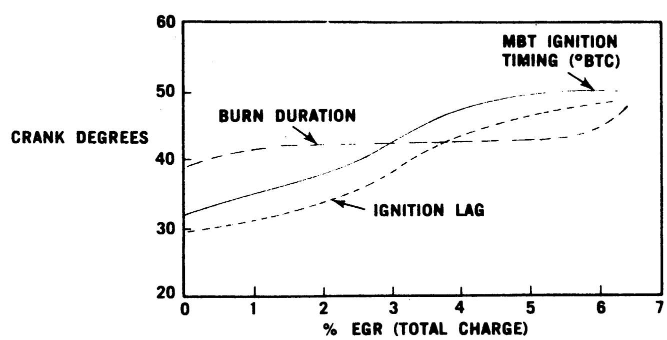

Fig. 25 - Pre-chamber EGR effect on combustion characteristics

combustion of the lean end gas diluted by the EGR.

Fig. 29, NO emission results, shows the extreme effectiveness of pre-chamber EGR in reducing NO emissions as postulated at the beginning of the text. In comparison, large amounts of EGR are required for the main chamber and the conventional chamber resulting in high HC emissions at a comparable low NO level.

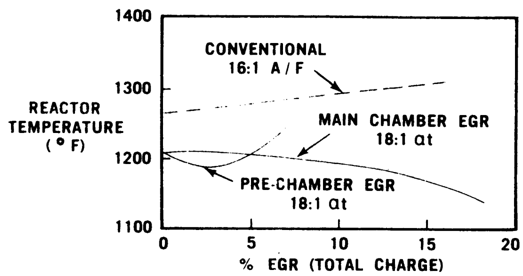

Fig. 26 - Reactor temperature versus % EGR at MBT ignition timing

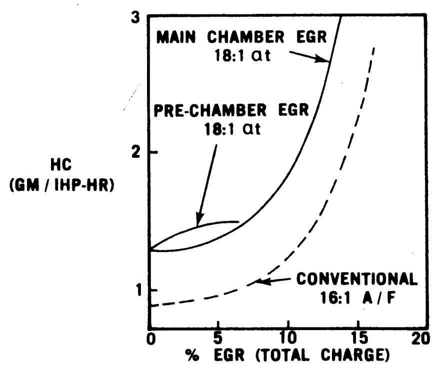

Fig. 27 - HC emissions versus % EGR at MBT ignition timing

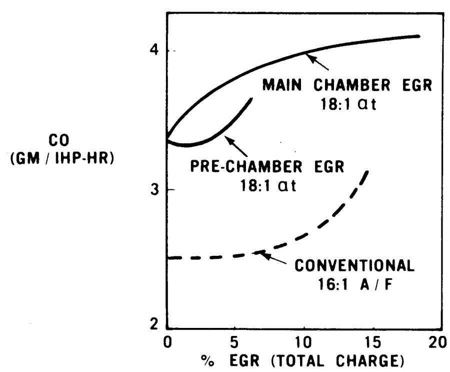

Fig. 28 - CO emissions versus % EGR at MBT ignition timing

Fig. 30 shows that specific fuel consumption is oppositely affected. The pre-chamber EGR with the largest NO decrease shows the fastest increase in ISFC with increasing EGR. A cross plot in Fig. 31 shows that a definite trade-off exists for decrease in NO emissions to an increase in fuel consumption with increasing EGR at MBT spark. It is interesting to note that the same relationship exists both for pre-chamber and main chamber EGR, both tested at 18:1 a t and shown as one line. The conventional run at 16:1 A/F shows the same trend at low EGR rates and the curves displaced only by the initially higher NO level due to the difference in A/F and the stratification effect disucssed earlier.

Fig. 32 expresses the NO and HC trade-off with EGR on a percent basis. At MBT spark and a more than 50% reduction in NO with EGR, the HC increase is more rapid for the conventional than the pre-chamber.

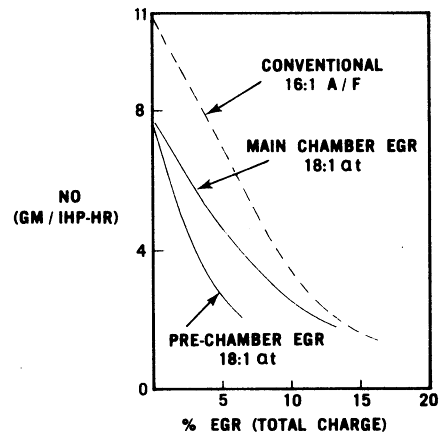

Fig. 29 - NO emissions versus % EGR at MBT ignition timing

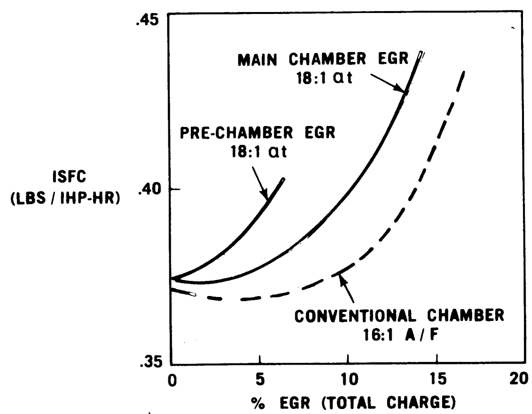

Fig. 30 - ISFC versus % EGR at MBT ignition timing

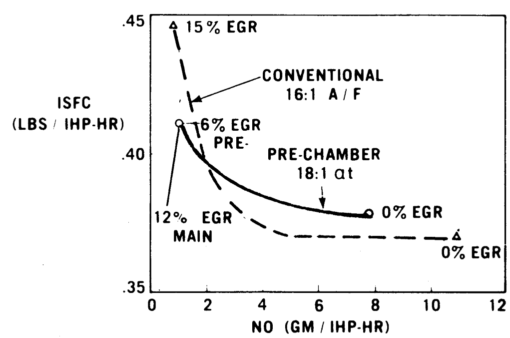

Fig. 31 - Fuel consumption relationship with NO emissions at MBT ignition timing and variable EGR rate

In NO vs. ISFC trade-off with EGR at MBT spark, shown in Fig. 33, the pre-chamber has a lower percentage loss in ISFC at over 85% decrease NO level. At a smaller NO decrease, the conventional has a smaller increase in ISFC. Figs. 34 through 37 are crossplots of data at variable EGR rates and retarded spark. To this point only their individual effects have been demonstrated. The following analysis will extract the best compromise combination.

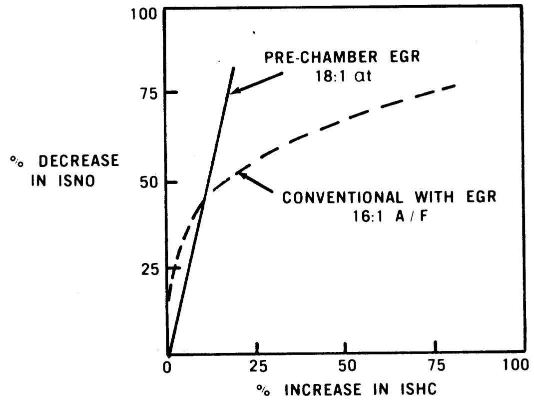

Fig. 32 - NO and HC emission trade-off with variable EGR and MBT ignition timing

advantage over the conventional in minimum ISFC, again occurring at about the 1.3 gm HC level.

Fig. 36 shows that the crossing over point where both engines are equal in ISFC occurs at about the 1.5 NO level. At lower than 1.5 NO the pre-chamber engine has lower ISFC but at a higher level the conventional is lower in ISFC.

Fig. 37 shows how HC emissions compare when both

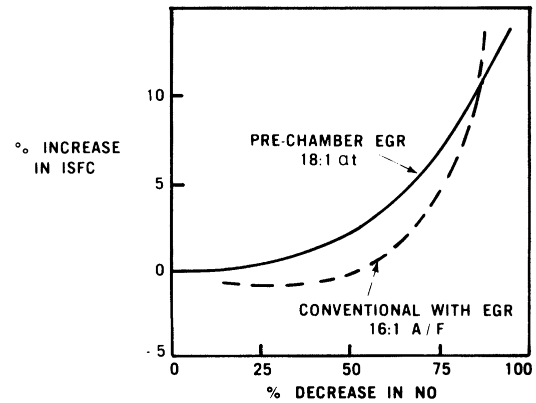

Fig. 33 - NO emission and ISFC trade-off with variable EGR and MBT ignition timing

At a constant 2 gm/IHP-HR NO level, in Fig. 34, HC emissions and ISFC increase rapidly as more EGR is introduced in conjunction with less ignition retard. At this test condition it is to HC advantage to operate with less EGR and more spark retard. At the higher NO levels, the pre-chamber engine is at a disadvantage because it cannot obtain the low ISFC that is possible with the conventional. The pre-chamber engine is 3% higher in minimum ISFC which is at about the 1.3 gm HC level.

At 1 gm/HP-HR NO emission level, shown in Fig. 35, the positions are reversed and the pre-chamber engine has a 6%

engines are operated at the lowest ISFC possible for a given NO level with varied spark and EGR. The difference in ISFC is not great in the low 1 to 2 ISNO range.

In summary, EGR introduced into the pre-chamber is more effective than EGR introduced in the main chamber because much less EGR is required to reach a given NO level and results in a minimum increase in HC. The pre-chamber engine, in combination with EGR introduced in the pre-chamber and retarded spark, has lower ISFC, at less than 1.5 ISNO emission level and the same HC emissions, than a conventional engine.

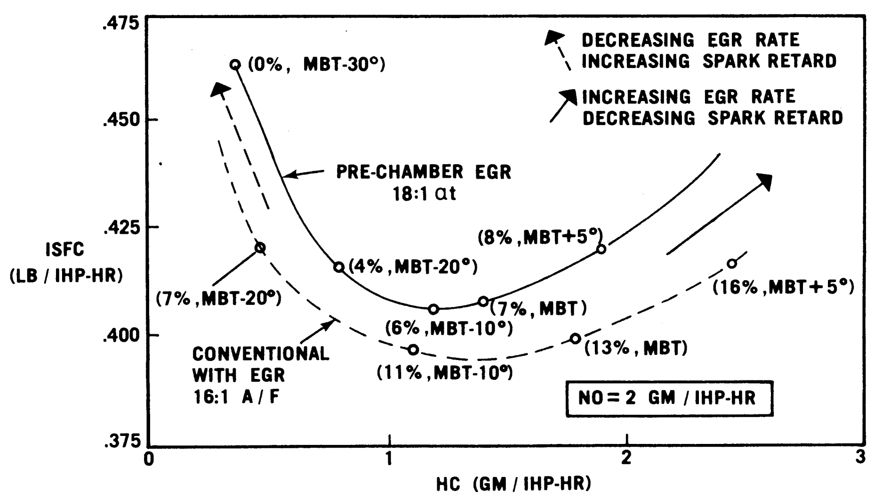

Fig. 34 - ISFC relationship with HC emissions at a constant 2 GM/IHP-HR NO emission level—variable EGR and ignition timing

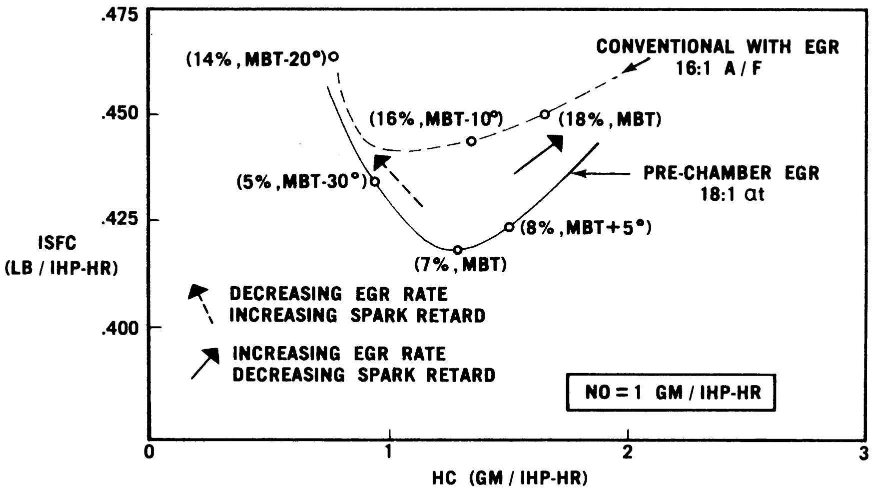

Fig. 35 - ISFC relationship with HC emissions at a constant 1 GM/IHP-HR NO emission level–variable EGR and ignition timing

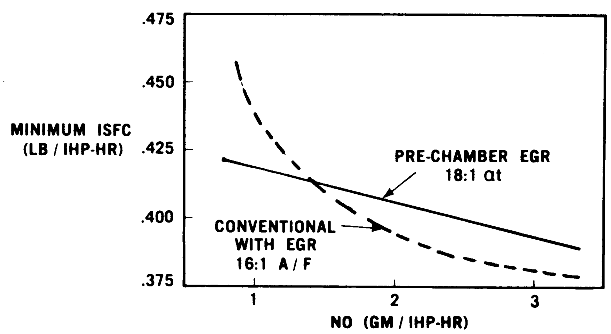

Fig. 36 - Minimum ISFC for given NO emission level with variable ignition timing and EGR rate

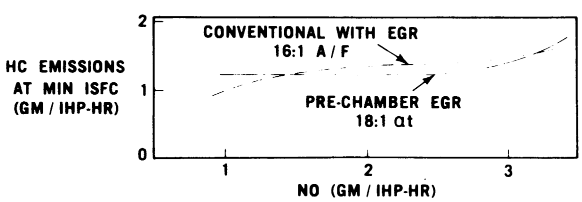

Fig. 37 - HC emissions at the minimum ISFC point for a given NO level with variable ignition timing and EGR rate

SUMMARY AND CONCLUSIONS

A pre-chamber engine and a conventional chamber engine were compared from the emission and economy standpoint on a single cylinder test at one "road load" operating condition. It was found that both engines respond similarly to the operating variables of A/F ratio, spark advance and exhaust gas recirculation. The main difference found was that the pre-chamber engine can be operated at a leaner overall A/F ratio than the conventional engine which results in lower NO emissions without exhaust gas recirculation. An added bonus of the leaner combustion is the presence of an adequate amount of O_2 in the exhaust gas for post oxidation of HC and CO in the exhaust system without secondary air injection.

From the limited single-cylinder tests conducted on this particular design of pre-chamber engine, in comparison to a conventional production engine, the following conclusions are reached:

1. The pre-chamber engine was found to have the ability to operate at 3 to 4 overall air/fuel ratios learner than the conventional engine. The conventional engine would misfire at ratios leaner than 19:1, whereas the pre-chamber engine operated satisfactorily at 23:1.

2. At MBT spark timing in the air/fuel range of 14 to 19, the pre-chamber engine shows lower NO emissions than the conventional engine but higher levels of HC emissions and poorer fuel economy.

3. With both the conventional and pre-chamber engines, only residual CO is present in the exhaust gas at leaner than 16:1 A/F ratio. At operating conditions where exhaust gas temperature is at about 1250°F CO is formed from post oxidation of HC in the exhaust system and, because of the low initial concentration level, a large percentage increase in tailpipe CO is observed.

4. EGR can be effectively utilized in reducing NO emissions from the pre-chamber engine. When EGR is introduced into the pre-chamber, a substantial drop in NO emission results without any significant increase in HC emissions.

5. At the low NO emission levels of <1.5 GM/IHP-HR the pre-chamber engine with EGR has lower fuel consumption than the conventional engine also wth EGR. At the higher NO emission levels, the conventional engine has lower fuel consumption.

NOMENCLATURE

A/F = air/fuel ratio of the conventional engine or overall ratio of the pre-chamber engine

αt = overall air/fuel ratio of the pre-chamber engine
αm = main chamber air/fuel ratio of the pre-chamber engine
αa = pre-chamber air/fuel ratio
λ = pre-chamber to main chamber mass air flow ratio
β = pre-chamber to total fuel flow ratio
αai = pre-chamber theoretical air/fuel ratio at the time of spark ignition
αmi = main chamber theoretical air/fuel ratio at the time of spark ignition
EGR = exhaust gas recirculation
MBT = minimum spark advance for best torque

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APPENDIX A
ENGINE SPECIFICATIONS

CONVENTIONAL ENGINE

Bore	4.00 in.
Stroke	4.00 in.
Compression Ratio	8.6:1
Displacement	50.26 cu. in.
Combustion Chamber Surface-to-volume ratio at TDC	5.58 in.$^{-1}$
Intake Valve Dia.	2.05 in.
Exhaust Valve Dia.	1.66 in.
Valve Lift	.43 in.
Camshaft Timing	256° Duration 17° BTC I.O. 21° ATC E.C. 38° Overlap

PRE-CHAMBER ENGINE

Same as conventional except:

Total Combustion Chamber Surface-to-volume ratio	6.73 in.$^{-1}$
Pre-Chamber Cup Volume	.598 cu. in. (8.9%)
Pre-Chamber Orifice Dia.	.472 in. (12mm)
Pre-Chamber Cup Wall Thickness	.08 in.
Pre-Chamber Cup I.D.	.70 in.
Exhaust Valve Dia.	1.63 in.
Main Intake Valve Dia.	1.93 in.
Pre-Chamber Intake Valve Dia.	.57 in.

APPENDIX B
CALCULATION OF SPECIFIC MASS EXHAUST EMISSIONS

Indicated data is used to allow comparison of engine combustion chamber performance between engines with different mechanical efficiencies. Since the single cylinder pre-chamber engine has higher friction than 1/8 of that of an 8-cylinder engine, the performance only on indicated basis is meaningful.

Expressing emissions on a specific basis allows for comparison of emission levels among different engines.

The specific emissions are calculated using exhaust concentration (HC, CO or NO) times its molecular weight times the exhaust flow rate and divided by the horsepower.

$$\text{HC,GM/IHP-Hr} = \frac{(1.30343 \times 10^{-3})\,(\text{HC, as a hexane, PPM})\,(\text{Air Flow \#/Hr})\,(\text{Moles Exh./Mole Air})^{*}}{\text{IHP}}$$

$$\text{CO,GM/IHP-Hr} = \frac{(4.3860)\,(\text{CO\%})\,(\text{Air Flow \#/Hr})\,(\text{Moles Exh./Moles Air})^{*}}{\text{IHP}}$$

$$\text{NO,GM/IHP-Hr} = \frac{(7.2036 \times 10^{-4})\,(\text{NO, PPM})\,(\text{Air Flow \#/Hr})\,(\text{Moles Exh./Mole Air})^{*}}{\text{IHP}}$$

*From Ref. 5.

APPENDIX C
EGR RATE CALCULATIONS

The formula used for calculation of "Percent EGR" is:

$$\%\,\text{EGR} = \frac{CO_2\ \text{intake} - CO_2\ \text{background}}{CO_2\ \text{exhaust} - CO_2\ \text{intake}} \qquad (1)$$

Where:

CO_2 intake = Mole percent of CO_2 in the intake manifold with recirculated exhaust.

CO_2 background = Mole percent of CO_2 in the intake manifold without recirculated exhaust. This includes the effects of ambient CO_2 and CO_2 from valve overlap effects.

CO_2 exhaust = Mole percent of CO_2 in the exhaust gas to be recirculated.

For applications where secondary air is introduced before the exhaust to be recirculated is picked up, the "Percent EGR" should be corrected by the formula.

$$\%\,\text{EGR} = \frac{R\,(1 - O_2/21)}{1 + R\,(O_2/21)}$$

Where:

R = Percent EGR calculated by formula (1)
O_2 = Mole percent of oxygen in the recirculated exhaust gas

This definition gives the relationship between EGR flow and fresh fuel/air mixture. It is important to note that EGR rates higher than 100% are possible with this definition. This condition exists whenever the EGR mass flow rate exceed the mass flow of fresh fuel and air.

APPENDIX D
DESCRIPTION OF TEST INSTRUMENTATION

COMBUSTION MEASUREMENTS

The Tektronix Digital Processing Oscilloscope System used for compiling the combustion data consisted of the following components shown.

7704A Oscilloscope	4911 Tape Perforator and Reader
P7001 Processor	021-0117-00 Interface
119-0465-00 Data Acquisition Processor	7A18 Amplifier
4010-1 Terminal	7B53A Time Base
4610 Hard Copier	

The D.P.O System has the capability of inputting, digitizing and sorting in the computer four waveforms simultaneously. The A-D converter has a digitizing rate of ~ 6 us/data point and can store consecutive cycles from an engine below 2000 rpm.

The computer can be user programmed in a variety of ways. The language is DEC BASIC modified to include commands for data input from the oscilloscope and also special routines for waveform analysis.

The turn around time for data analysis is about 10 minutes for the combustion calculations. This allows for on-the-spot interpretation and validation of data while engine is running.

The cylinder pressure measurement was made with a Kistler Model 618-A5 pressure transducer installed in the main combustion chamber.