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EVALUATION OF PRECHAMBER SPARK IGNITION ENGINE CONCEPTS

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EVALUATION OF PRECHAMBER SPARK IGNITION ENGINE CONCEPTS

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ABSTRACT

This report presents a review of the performance, emission, and operational characteristics of prechamber (or divided chamber) spark-ignition engine concepts including an analysis and evaluation of the applicability of these concepts to new automotive and stationary engines and retrofit installations. Relative to conventional automotive engines, prechamber engines exhibit very low carbon monoxide emissions and show some reduction in the emission of oxides of nitrogen. However, the hydrocarbon emission from prechamber engines is similar to that of conventional engines employing noncatalytic emission control systems, indicating a need for aftertreatment devices such as lean thermal reactors or catalytic converters. The fuel consumption of vehicles equipped with prechambers is similar to or slightly better than that of equivalent conventional vehicles at comparable levels of emission control.

SECTION 1

HIGHLIGHTS

An examination and summarization was made of available information pertaining to the design, performance, emission, and fuel consumption characteristics of automotive and stationary prechamber spark-ignition engine concepts. A considerable amount of relevant technical data was obtained during the data acquisition phase of this study and is presented in the body of the report. An analysis and evaluation of these data resulted in the following findings:

1. Prechamber or divided chamber spark-ignition engines are modifications of conventional engines in which a rich air-fuel mixture is generated in the spark plug region of the prechamber while a lean mixture, or even pure air, is inducted into the main chamber. The resulting two-stage combustion process permits stable operation of the engine at very lean overall air-fuel ratios, accompanied by lower peak combustion temperatures, lower dissociation losses, and lower throttling losses. These conditions are conducive to low HC, CO, and NO_x emissions, low fuel consumption, and, possibly, reduced sensitivity to fuel octane number.

2. While many different prechamber concepts have been pursued by numerous investigators for more than half a century, only very few designs have now reached an advanced state of research and development. In particular, these include Honda Motor Company's automotive CVCC (Compound Vortex Controlled Combustion) system, which has been in mass production in Japan since December 1973, and Fairbanks Morse

Engine Division's opposed piston heavy-duty stationary gas engine, which has been sold commercially since 1952. Currently, most domestic and foreign automobile manufacturers are expending considerable efforts on the development of prechamber engines for potential future application to one or more of their vehicle models.

3. The complexity of the known prechamber concepts varies from very simple configurations such as the Cornell spark plug to sophisticated designs consisting of a camshaft-actuated prechamber intake valve combined with carburetion or fuel injection. The automakers are concentrating their efforts on the latter configuration, whereas a number of other organizations, including the Combustion Control Subsidiary of Systron Donner and Teledyne Continental Motors, are experimenting with retrofittable designs employing a pressure-actuated prechamber intake valve and an auxiliary carburetor/fuel vaporizer arrangement.

4. Similar to conventional spark-ignition engines, hydrocarbons (HC), oxides of nitrogen (NO_x), and carbon monoxide (CO) are the principal known pollutant species emitted from prechamber spark-ignition engines. Other pollutants emitted from these engines include primarily aldehydes and odor at levels comparable to conventional spark-ignition engines. Because of the lean operation, the CO emission from prechamber engines is generally very low compared to conventional automotive engines and similar to conventional stationary gas engines which are always adjusted for lean operation. Conversely, the NO_x emission from current prechamber engines is somewhat below that of conventional engines employing exhaust gas recirculation, while the HC levels are similar, indicating the need for exhaust aftertreatment devices such as lean thermal reactors or oxidation catalysts to meet future emission standards.

5. In tests over the Federal Driving Cycle, a number of automobiles equipped with three-valve prechamber engines and thermal exhaust reactors have demonstrated reasonable emission control consistent with acceptable fuel economy. It appears that prechamber engine-powered vehicles in the 2000-lb weight class would be capable of achieving emission levels of about

0.4 gr/mile HC, 2-4 gr/mile CO, and 0.6 - 1.0 gr/mile NO_x with fuel economy similar to equivalent conventional 1974 light-duty vehicles. Conversely, vehicles in the 5000-lb weight class, equipped with three-valve prechamber engines, would have NO_x emissions of about 1.4 - 1.7 gr/mile while HC and CO would be unchanged from the levels of the 2000-lb vehicle.

6. Further reduction of the NO_x emission from prechamber engines would be possible by means of exhaust gas recirculation (EGR) and/or spark timing retard. However, this would be accompanied by a significant increase in the specific fuel consumption and HC emission of the engine. Therefore, consideration of other approaches might prove to be beneficial, including optimization of the prechamber geometry, turbulence levels in the prechamber and main chamber, size and shape of the prechamber orifice, and prechamber air-fuel ratio.

7. While a number of investigators have reported improvements up to 25 percent in the specific fuel consumption of their engines relative to conventional engines, based on engine-only tests, these trends could not be duplicated by the concept developers in vehicle installations. Over the Federal Driving Cycle, the fuel economy of most prechamber engine-powered vehicles tested to date has been comparable to that of conventional vehicles, although the emissions from prechamber engines were generally lower. It is conceivable, however, that some improvement in fuel economy might be achieved by optimizing certain engine design parameters and/or by permitting higher emission levels.

8. The manufacture of automotive prechamber engines incorporating cam-actuated third-valves would require significant modifications and additions to existing engine production lines. Because of the related high capital investment requirements and the long lead time of the machine tool industry, prechamber engines would probably be phased in gradually, starting with one or two lines. General Motors has indicated that this might be accomplished within 24 to 30 months after program approval.

9. The initial and operating costs of three-valve prechamber engines are comparable to the respective costs of conventional engines incorporating catalytic emission

control systems. Conversely, the maintenance cost of these prechamber engines would be lower considering the periodic catalyst replacements that might be required in conventional vehicles to meet future emission standards. While the manufacturing cost of prechamber systems employing pressure-actuated intake valves would be somewhat lower, the performance potential and durability of these concepts has not been adequately demonstrated.

10. The power output capability of automotive prechamber engines operating with very lean overall air-fuel mixtures is 5 to 25 percent lower than that of equivalent conventional engines. However, the power of the engine could be restored by increasing its displacement at the expense of an increase in unit cost. This approach is being pursued by some investigators while others contemplate the use of mixture enrichment as a means of increasing the power output of the engine in the high-load regime.

11. While the durability of Honda's CVCC engines has been shown to be similar to that of conventional engines, problems have been encountered in a number of prechamber engine configurations employing a pressure-actuated prechamber intake valve. These include overheating of the prechamber unit, failure and erratic operation of the third valve, and marginal vehicle driveability.

12. Based on tests conducted by a number of investigators including Honda, Ford, and General Motors, the driveability of vehicles equipped with prechamber engines is expected to be comparable to that of equivalent vehicles powered by conventional engines.

13. Detonation-like noise has been encountered in a number of prechamber engine designs independent of the octane rating of the fuel used. Conversely, most other prechamber engines exhibit noise characteristics similar to conventional spark-ignition engines. Some reduction in the noise level has been achieved in one engine by optimizing the size and shape of the communicating passage between the prechamber and main chamber.

14. While lower fuel octane requirements were reported by a number of investigators, Honda's CVCC engine and the three-valve concepts under development at General Motors have octane requirements similar to equivalent

conventional engines. There is some evidence, however, that the knocking characteristics of prechamber engines might be altered by modifying certain prechamber design details such as the geometry of the prechamber and the turbulence level in the prechamber and main chamber. As a result, lower octane gasoline might be applicable, which would be beneficial from a crude oil usage point of view.

15. Satisfactory prechamber engine operation on JP-4, CITE, and diesel fuels was achieved by Teledyne Continental Motors and Stanford University. Because of the attendant crude oil savings, further investigations should be conducted to determine the effect of these fuels on the performance and emissions of the engine over wide ranges of transient and steady-state operating conditions.

16. For economic reasons, the prechamber configurations incorporating a cam-actuated third-valve and/or a fuel injection system are not considered to be applicable as retrofit devices for in-use vehicles. While the Cornell spark plug and a number of prechamber concepts incorporating unscavenged prechambers or pressure-actuated prechamber intake valves are potential retrofit candidates for automotive engines, insufficient information is available regarding their cost, durability, performance, and emission potentials to permit a meaningful assessment at this time.

17. Although prechamber retrofitting of heavy-duty stationary engines in the field might be possible in principle, it appears that this approach would not be economically feasible considering the high cost of conversion and the uncertainties regarding the associated benefits in terms of emission and fuel consumption reduction.

18. The principal advantage of the three-valve prechamber engines under development by the automakers would appear to be their ability to meet the 1977 federal emission standards without the use of a catalytic converter. Based on the current state-of-the-art technology, NO_x emission levels below about 1.5 gr/mile would be difficult to achieve in standard size (4500 to 5000-lb) production automobiles without incurring substantial losses in fuel economy. This same NO_x level versus fuel economy problem is exhibited in conventional engines employing exhaust gas recirculation for NO_x control. Therefore,

the domestic automobile manufacturers may proceed very cautiously in arriving at a decision regarding the future of these prechamber engines unless there is a high probability that future NO_x emission standards would not be lower than about 1.5 gr/mile.

SECTION 2

INTRODUCTION

The concept of a prechamber engine dates back to the first oil engine developments of the pre-diesel era. While early attempts of incorporating prechambers into spark-ignition engines were largely unsuccessful, many organizations and individuals have shown renewed interest in the concept during the past several years, primarily because of its low emission and fuel consumption potential.

This study was initiated with the objective of summarizing and evaluating the available information pertaining to the applicability of prechamber concepts to light-duty automotive and heavy-duty stationary spark-ignition engines considering both new engine designs and retrofit installations.

To fulfill the objectives of this study, the effort was divided into two phases. The first phase was concerned with the compilation and review of applicable information acquired from (1) the open literature, and (2) discussions with engine manufacturers and other organizations and individuals active in prechamber engine research and development. In the second phase of the study, a summarization and evaluation was made of all data acquired in the first phase.

The results of this study are presented in the following order and context: Section 3 includes a brief discussion of the various stratified charge engine concepts devised to date and examines the emission and specific fuel consumption characteristics of prechamber engines in general.

Section 4 reviews the state of the art of many prechamber engine configurations which have been or still are under development by a number of automobile and stationary engine manufacturers and other organizations. Special attention is focused on the performance, emissions, and fuel economy of prechamber engines and comparisons are made with conventional engines, whenever possible. Section 5 presents an evaluation of prechamber engines with respect to performance and economics.

A compilation of prechamber engine patents granted by the United States Patent Office between 1914 and 1974 is presented in Appendix A. Appendix B lists those organizations and individuals that contributed to this study either directly by providing useful engine test data, or indirectly through general discussions of the combustion, emission, performance, and operational characteristics of prechamber engines. Appendix C presents metric system conversion factors.

SECTION 3

STRATIFIED CHARGE ENGINE APPROACHES

3.1 GENERAL ENGINE DESCRIPTION

In principle, the stratified charge engine is a modification of the conventional spark-ignition engine. The principal design difference consists of the application of two-stage combustion in which a rich air-fuel mixture is generated around the spark plug and a lean mixture in the remaining zones of the combustion chamber. This staged combustion process permits operation of the engine at very lean overall air-fuel ratios which is conducive to low emissions, good fuel economy, and reduced sensitivity of the engine to fuel octane number. The stratified charge engines can be divided into two distinct classes: open chamber engines and prechamber or divided chamber engines.

3.1.1 Open Chamber Stratified Charge Engines

In the open chamber configurations, exemplified by the Texaco TCCS and Ford PROCO engines, a single combustion chamber is employed similar to that of conventional spark-ignition engines (Refs. 3-1 and 3-2). In the TCCS system, an air swirl is set up in the cylinders by means of directional intake porting combined with special piston cup designs. Fuel is injected into each cylinder toward the end of the compression stroke. Upon ignition of the swirling, rich mixture surrounding the spark plug, the burning charge expands into the outer regions of the combustion chamber where the heterogeneous combustion process is then completed in an oxygen-rich environment. The rate of combustion is controlled by varying the rate of fuel injection (Ref. 3-3).

In the Ford PROCO engine, the fuel is injected into the piston cup during the compression stroke to permit vaporization of the fuel while forming a stratified charge. The rate of combustion is controlled by the flame speed and the volume of the air-fuel mixture formed in the engine (Ref. 3-3).

Open chamber stratified charge engines are not considered in this report.

3.1.2 Prechamber Stratified Charge Engines

The prechamber stratified charge engine or divided chamber engine, exemplified by Honda's CVCC engine concept, employs two interconnected combustion chambers per cylinder (Ref. 3-4). During the suction stroke of the piston, a fuel-rich mixture is inducted into the generally smaller prechamber, while the main chamber is charged with a lean mixture or even pure air. In principle, both carbureted and fuel injected configurations are feasible. Upon ignition in the prechamber, hot gases expand into the main chamber where the combustion process is then carried to completion. The principal advantage of prechamber engines over conventional engines is their ability to operate with very lean overall air-fuel mixtures resulting in low emissions, particularly NO_x. However, because of the less favorable combustion chamber surface-to-volume ratio combined with high turbulence, the heat losses of this engine tend to be higher than in conventional designs.

While unthrottled operation of the engine would be desirable from an efficiency point of view, some throttling might be required at light loads to achieve acceptable driveability and HC emission characteristics.

The benefits in terms of emission reduction and fuel economy improvement that might be realized in a particular design, depend upon the tradeoffs between the heat losses and the inherently higher thermodynamic cycle efficiency obtained with operation in the lean air-fuel mixture regime.

3.2 HISTORIC DEVELOPMENT

The concept of a prechamber spark-ignition engine dates back to several patents granted by the U.S. Patent Office in the early 1920s. While the divided chamber engine patented by Ricardo in 1918 (Ref. 3-5) was successfully operated on gaseous and prevaporized light distillate fuels, incorporation of this concept into other spark-ignition engines proved to be very disappointing. As a result, the engine manufacturing industry showed little interest in the development of the concept, although many patents were subsequently issued throughout the world. The most significant patents granted by the United States Patent Office are listed in Appendix A.

Heavy-duty stationary prechamber gas engines have been marketed by Fairbanks Morse Engine Division of Colt Industries for more than twenty years. More recently a number of organizations and individuals have become involved in the research and development of light-duty prechamber engines for potential use in automotive applications. In particular, the Honda Motor Company of Japan has conducted extensive design, development, and test work on its CVCC engine during the past several years. This engine is now in production and is being utilized by Honda as the standard power plant in its 1975 model year Civic automobile exported to the United States.

3.3 PERFORMANCE CHARACTERISTICS

The thermal efficiency of spark-ignition (Otto cycle) engines increases with increasing compression ratio and air-fuel ratio, as shown in Figure 3-1 (Ref. 3-3). As indicated, leaning the mixture results in a substantial improvement in thermal efficiency. This is accompanied by a reduction in the combustion temperature, dissociation effects, and heat-losses permitting further improvement in the efficiency of the engine. However, in conventional engines the flame speed decreases with increasing air-fuel ratio and as a result, the combustion process is extended over a longer time period causing a loss in thermal efficiency (Ref. 3-6).

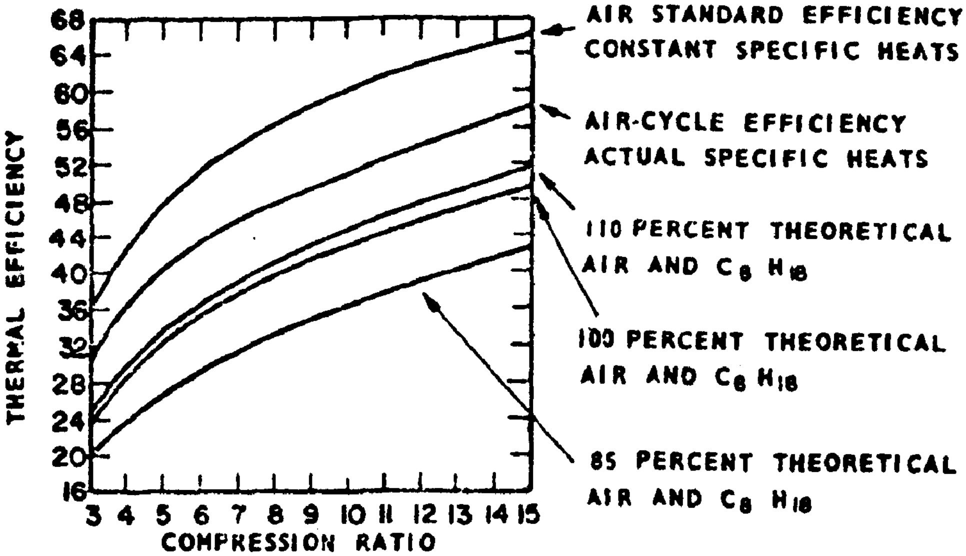

Figure 3-1. Thermal efficiency versus compression ratio and air-fuel ratio (Ref. 3-3)

While efforts are being conducted by the automotive industry and by certain research organizations and individuals to extend the lean limit of conventional automotive engines by means of improved carburetors, intake manifolds, and combustion chamber designs, the projected gains are not sufficient to meet future NO_x emission goals. Conversely, prechamber engines can be operated efficiently with very lean mixtures. In this case, the flame speed of the lean main chamber charge is increased substantially by the multiple ignition sources and by high turbulence levels created by the hot and highly reactive combustion products emanating from the prechamber.

3.4 EMISSION CHARACTERISTICS

Stratified charge engines operating with lean air-fuel mixtures have the potential of reduced exhaust emissions relative to conventional spark-ignition engines. This is illustrated in Figure 3-2, showing the specific mass emissions of hydrocarbons (HC), carbon monoxide (CO), and oxides of nitrogen (NO_x) as a function of the air-fuel ratio supplied to the engine (Ref. 3-7). At very low air-fuel ratios (rich mixtures) NO_x is low, HC and CO emissions are high, and specific fuel consumption is high. As the air-fuel ratio increases, NO_x rises, reaching a maximum about 10 percent above stoichiometric, accompanied by declining HC and CO. With further leaning of the mixture, NO_x decreases rapidly while CO and HC increase.

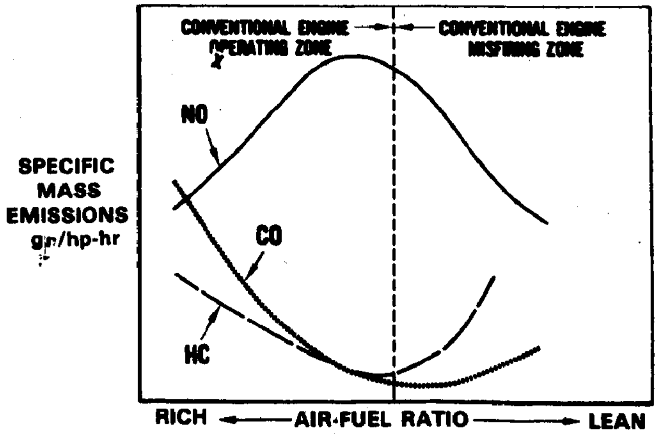

Figure 3-2. Specific mass emissions versus air-fuel ratio (Ref. 3-7)

The rate of NO_x formation is primarily determined by three factors: peak combustion temperature, residence time at high temperatures, and oxygen availability. In prechamber engines, the combustion process commences in the fuel-rich prechamber near the top dead-center position of the piston. Because of a lack of oxygen, very little NO is formed under these conditions. As combustion proceeds in the prechamber, the burning charge is expanded into the main chamber where an adequate amount of oxygen is available to complete the chemical reactions. Since the air in the main chamber is relatively cool, the NO formation reactions are rapidly quenched, hence minimizing NO_x. In addition, the rapid rate of energy release in divided chamber engines permits the use of retarded spark timing which further inhibits the formation of NO_x.

The HC emitted from internal combustion engines is primarily related to quenching of the oxidation reactions in the wall region of the combustion chamber and in ultra-lean zones of the air-fuel charge in the main chamber. In addition, the fuel captured in various engine crevices has been shown to contribute to the emission of unburned HC (Ref. 3-8). Conversely, the higher turbulence level in the main chamber created by the hot prechamber gases tends to reduce the effect of the quench layer (Ref. 3-7). Based on currently available information, the raw HC emitted from prechamber engines is high and aftertreatment devices such as catalytic converters or thermal reactors would be required to meet future HC emission standards.

Carbon monoxide is the result of a deficiency in oxygen during the combustion process. While sufficient oxygen is ultimately available in prechamber engines to complete the CO reactions, the oxygen may not reach the CO molecules before the temperature of the gases in the chamber has declined to a value too low for oxidation to proceed. In general, the CO concentration emitted from prechamber engines is quite low. However, in the very lean air-fuel mixture regime, the specific mass emission of CO tends to increase again with increasing air-fuel ratio because of the high specific air flow rates (pounds of air per horsepower-hour) associated with ultra-lean engine operation.

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SECTION 4

PRECHAMBER ENGINE CONCEPTS

This section of the report reviews the state of the art of a number of prechamber spark-ignition engine concepts which have been or still are under development by a number of automobile manufacturers, stationary engine manufacturers, and other organizations.

Subsection 4.1 presents technical information relative to the various prechamber engine configurations that are currently being considered by the automobile industry for potential use in light-duty vehicle applications. These include development efforts by Ford Motor Company, General Motors Corporation, Honda Motor Company, and Volkswagenwerk A. G. Subsection 4.2 is concerned with the prechamber work conducted by other industrial firms, including Combustion Control Subsidiary of Systron Donner Corporation, Eaton Corporation, Phillips Petroleum Company, Teledyne Continental Motors, Thermo Electron Corporation, and Walker Manufacturing Company. Subsection 4.3 treats the prechamber engine research and development programs conducted by a number of universities and research organizations. These include The Aerospace Corporation, California State University at Sacramento, Cornell University, Stanford University, University of California at Berkeley, University of Rochester, University of Wisconsin, and a number of Russian developments. The status of the stationary prechamber engine developments conducted by Fairbanks Morse Engine Division of Colt

Industries and other manufacturers of stationary engines is discussed in Subsection 4.4, while pertinent technical information on a number of other promising domestic and foreign prechamber concepts is presented in Subsection 4.5.

4.1 AUTOMOBILE MANUFACTURERS

4.1.1 Ford Motor Company

The Ford Motor Company has been involved in the development of a number of different prechamber engine concepts for some time. The configurations considered to date fall into one of the following three categories: (1) small prechambers occupying less than 5 percent of the total clearance volume, (2) medium-size prechambers utilizing about 15 percent of the clearance volume, and (3) large prechambers with a volume of more than 20 percent of the clearance volume. Since all these configurations are in various phases of development, the available information is sketchy and the interim data presented in this section do not necessarily reflect the ultimate capability of a given design.

4.1.1.1 System Description

4.1.1.1.1 Small Prechamber

The Ford torch ignition engine, illustrated in Figure 4 consists of the torch chamber cavity containing the spark plug, and the main combustion chamber which communicates with the prechamber through a number of small orifices (Refs. 4-1 and 4-2). A small fraction of the homogeneous air-fuel mixture supplied to the engine by means of a conventional carburetor enters the torch chamber during the compression stroke. Upon ignition of the prechamber charge, the pressure in the prechamber rises rapidly, forcing the hot combustion gases into the main chamber through the communicating passages.

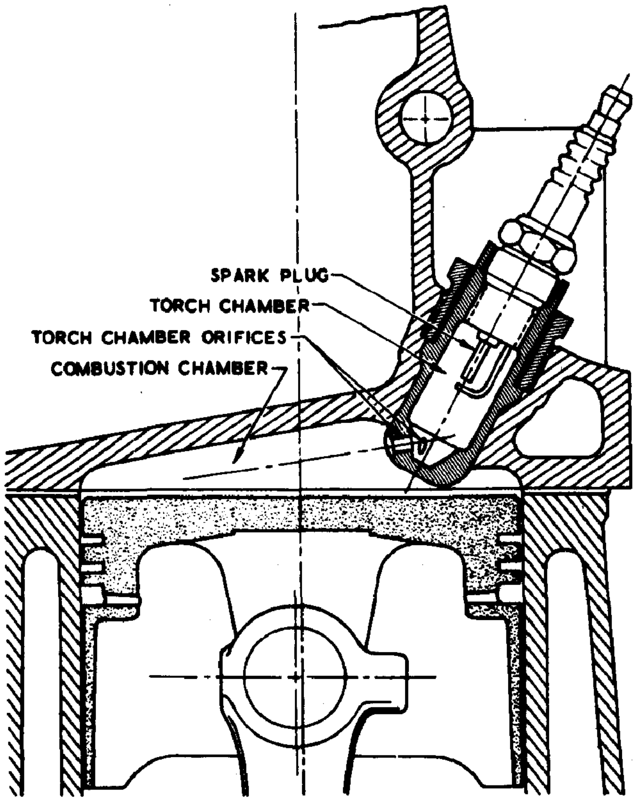


Figure 4-1. Ford torch ignition engine (Ref. 4-1)

Penetration of the hot gas jets deep into the main chamber assures rapid combustion of the lean main chamber charge.

Preliminary torch ignition tests were conducted by Ford on a 1972 Gran Torino automobile equipped with a 351 CID V-8 engine, automatic transmission, and a rear axle ratio of 2.75. The engine had a compression ratio of 8.0 and incorporated a cold-start spark advance system, a heat control valve, and air injection into the exhaust ports (Ref. 4-1). To provide a better understanding of the torch ignition combustion characteristics, Ford has plans to investigate the critical prechamber parameters using a single-cylinder engine.

4.1.1.1.2 Medium-Size Prechamber

To date, Ford has experimented with two different medium-size prechamber configurations: (1) a carbureted three-value prechamber engine and (2) a prechamber with fuel injection.

The three-valve engine employs two separate carburetors to supply the air-fuel mixture to the main chamber and to the prechamber. The main chamber carburetor is set for lean mixture operation, and a small, pressure-fed carburetor supplies a rich mixture to the precombustion chamber which occupies about 8.9 percent of the total clearance volume. The chamber is equipped with a small secondary intake valve (the third valve) which is actuated simultaneously with the main intake valve. The spark plug is also located in the prechamber. Upon ignition in the prechamber, the combustion gases expand into the main chamber through a small orifice and ignite the lean charge in the main combustion chamber. The three-valve concept was tested in a modified 400 CID V-8 engine as shown in Figure 4-2 (Ref. 4-3). In this particular configuration, the precombustion chamber was equipped with a thin liner to minimize heat losses, and special pistons with contoured edges were utilized to decrease the turbulence level created during the intake stroke (Ref. 4-2). Also, exhaust port liners and a thermal reactor were added to improve HC and CO control.

In the fuel-injected engine configuration, the prechamber, occupying a volume of 12 percent of the total clearance volume, was designed for installation in the spark plug well with minor rework of the cylinder head. Again, the main chamber was supplied with a lean mixture through a standard carburetor, while the mixture in the prechamber was enriched by means of a direct, low-pressure fuel injection system which incorporated a constant-flow, solenoid-operated injector. The prechamber fuel flow rate was controlled by varying the injection

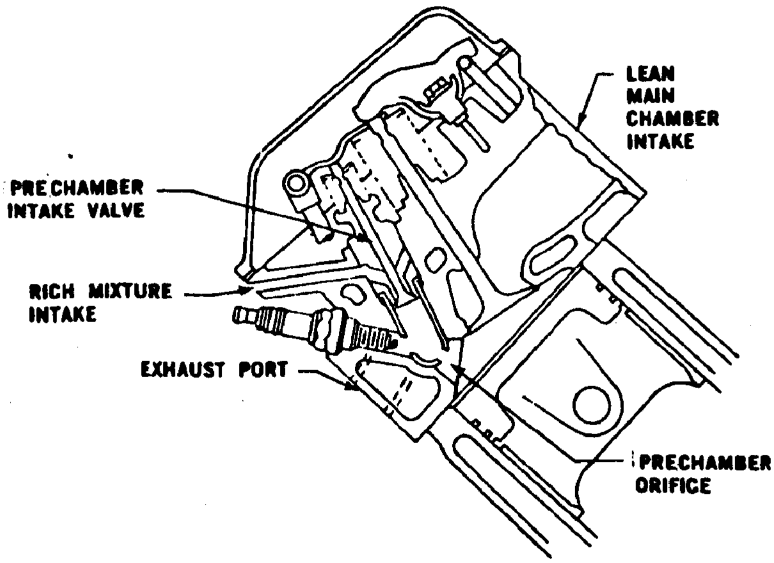


Figure 4-2. Ford experimental three-valve prechamber engine configuration (Ref. 4-3)

timing. This particular configuration was tested in a four-cylinder 140 CID engine having a compression ratio of 7.25 (Ref. 4-2).

4.1.1.1.3 Large Prechamber

In this concept, air is inducted into the engine cylinder through a standard intake valve, while the fuel is injected into the large prechamber by means of a medium pressure injector. To date, this concept has been evaluated in a single-cylinder engine. Additional development work on this concept is planned in conjunction with a V-8, 400 CID engine (Ref. 4-2).

4.1.1.2 Emission Characteristics

4.1.1.2.1 Small Prechamber (Torch Ignition)

Emission test data from the 1972 Gran Torino vehicle equipped with Ford's torch ignition engine and operated in accordance with the 1975 Federal Test Procedure are listed in Table 4-1 (Ref. 4-1). In these tests, the engine was operated with variable exhaust gas recirculation (EGR) rates and with air injection into the exhaust ports. As indicated, the HC, CO, and NO_x emissions were below the 1976 federal standards (1.5 gr/mile HC; 15 gr/mile CO; 3.1 gr/mile NO_x). While HC and NO_x were below the 1976 California standards (0.9 gr/mile HC; 2.0 gr/mile NO_x), CO was above the California standard (9.0 gr/mile) in two of the four tests reported.

Table 4-1. EMISSIONS FROM 1972 GRAN TORINO WITH TORCH IGNITION ENGINE - 1975 FEDERAL TEST PROCEDURE; 4500-lb INERTIA WEIGHT (Ref. 4-1)

Test No.	Exhaust emissions, gr/mile			Fuel economy, mpg	EGR flow, %	Remarks
	HC	CO	NO_x			
36	0.81	15.0	1.6	11.1	3-12	15:1 A/F, 6° BTDC timing with air injection
38	0.87	7.2	1.4	11.0	3-12	16:1 A/F, 6° BTDC timing with air injection
69	0.80	9.2	1.8	10.5	2-10	16:1 A/F, 4° BTDC timing with air injection
70	0.88	6.7	1.9	10.3	3-10	16:1 A/F, 4° BTDC timing with air injection

4.1.1.2.2 Medium-Size Prechamber

A considerable amount of development work was performed by Ford Motor Company on its three-valve prechamber concept utilizing a single-cylinder engine and a 400 CID V-8 engine (Ref. 4-3).

Initial tests on the V-8 engine were designed to evaluate the effect of air-fuel ratio, ignition timing, and EGR on the specific fuel consumption and HC, CO, and NO_x emissions. Based on these tests, a number of important facts have become apparent relative to prechamber engine combustion. These include:

1. Prechamber engines can be operated at overall air-fuel ratios that are about 3 to 4 units leaner than conventional spark-ignition engines.
2. The burning time in the prechamber engine is approximately 10 percent longer than in conventional-engines, while the ignition lag is approximately 50 percent shorter, permitting the use of more spark retard in prechamber engines.
3. At MBT (minimum advance for best torque), the prechamber shows about 30 percent lower NO_x emissions than the conventional engine, accompanied by higher levels of HC and CO and a loss in fuel economy of about 5 percent or less.
4. Introduction of EGR into the prechamber can result in substantially lower NO_x emissions without significant increases in HC or CO.
5. In the low NO_x regime (< 1.5 gr/bhp-hr), the prechamber engine with EGR has lower specific fuel consumption than a conventional engine utilizing EGR. Conversely, at higher NO_x levels, the conventional engine is more efficient.

Test data from the carbureted three-valve perchamber work are presented in Tables 4-2 and 4-3 (Ref. 4-1). Table 4-2 shows the dynamometer calibration of the engine when adjusted to maintain an exhaust temperature of 1400°F and a NO_x level of

Table 4-2. FORD 400 CID THREE-VALVE PRECHAMBER CALIBRATION (Ref. 4-1)

Engine speed, rpm	Initial main intake depr. in. Hg	IMEP load, psi	Overall A/F	Secondary mixture A/F	Spark timing
800	12	38	16.5:1	10:1	12° ATC[a]
1000	12	43	16.7:1	10:1	6° ATC[a]
1500	9	58	17.0:1	10:1	5° BTC[b]
1500	6	68	17.5:1	10:1	5° BTC[b]
2000	6	74	18.5:1	10:1	14° BTC[b]
2500	6	76	18.5:1	10:1	23° BTC[b]

[a]After top-center

[b]Before top-center

Table 4-3. PRELIMINARY HOT-START CVS VEHICLE TEST DATA; CARBURETED THREE-VALVE PRECHAMBER ENGINE; 4500-lb INERTIA WEIGHT (Ref. 4-1)

		Emissions, gr/mile			
Vehicle	Engine	HC	CO	NO_x	Fuel economy, mpg
1973 Ford LTD Automatic	400 CID 8.5 C.R.	0.43	5.69	1.62	9.1

1.3 gr/bhp-hr. The initial emission data obtained with this calibration over the CVS hot-start cycle are shown in Table 4-3. As indicated, the HC, CO, and NO_x emissions are below the 1976 federal and

California standards. However, inclusion of the cold-start phase of the Federal Test Procedure would result in higher HC and CO levels. On the other hand, further improvements in the emissions might be achieved by means of additional development work.

The emission tests conducted on the 140 CID engine equipped with fuel-injected prechambers were not promising, showing emission levels of 4 gr/bhp-hr NO_x; 15 gr/bhp CO; and 1.37 gr/bhp-hr NO_x (Ref. 4-2).

4.1.1.2.3 Large Prechamber

Dynamometer test data from the single-cylinder/large prechamber engine are listed in Table 4-4 (Ref. 4-2), showing indicated specific mass emissions at 1500 rpm and two-load settings. The HC and CO emission levels are quite high and would have to be reduced substantially to meet current and future emission standards. Conversely, the observed NO_x level is rather low.

Table 4-4. EMISSIONS FROM SINGLE-CYLINDER ENGINE WITH LARGE PRECHAMBER (Ref. 4-2)

Engine speed, rpm	IMEP, psi	HC, gr/ihp-hr	CO, gr/ihp-hr	NO_x gr/ihp-hr	ISFC, lb/ihp-hr
1500	40	1.2	5.0	0.5	0.395
1500	70	1.1	6.0	0.5	0.377

4.1.1.3 Fuel Consumption Characteristics

4.1.1.3.1 Small-Size Prechamber

According to Table 4-1, the fuel consumption of the 1972 Grand Torino equipped with Ford's torch ignition engine varied between 10.3 and 11.1 mpg. While no corresponding data are available for standard 1972 Torinos, EPA data indicate that the 1973 Torino with the same standard engine and inertia weight had a fuel economy of 9 mpg over the 1975 FTP. Assuming 5 to 10 percent better fuel economy for 1972 cars, this would correspond to approximately 10 mpg for a standard 1972 Torino. Thus, there appears to be no loss in fuel economy, and there may be even a small gain for the torch ignition engine relative to a standard engine at the same emission levels.

4.1.1.3.2 Medium-Size Prechamber

As indicated in Table 4-3, the fuel economy of the three-valve prechamber vehicle over the Federal Driving Cycle is 9.1 mpg. This is comparbale to the 8.8 mpg obtained with a similar vehicle incorporating a standard 400 CID engine and tested with an inertia weight of 5000 lb. Another 1972 vehicle with a 390 CID engine was tested over this cycle using an inertia weight of 4500 lb. This vehicle achieved a fuel economy of 9.5 mpg. Thus, the three-valve prechamber engine appears to have fuel consumption characteristics similar to conventional engines with moderate emission control, and may prove to be superior to conventional engines adjusted to meet more stringent standards.

While comparable fuel consumption data are not available for the prechamber engine with fuel injection, limited tests conducted on a four-cylinder engine indicate that this concept might be inferior to standard engines.

4.1.1.3.3 Large-Size Prechamber

The fuel consumption data available for the large prechamber engine are limited to single-cylinder dynamometer data. These data look promising and indicate that the fuel consumption of this concept should be equal to or perhaps better than that of a standard engine when tested at the same emission levels.

4.1.1.4 Vehicle Performance Characteristics

The torch ignition engine, when installed in a 1972 Torino, demonstrated acceptable driveability in the range of 5 to 7. The driveability index used by Ford covers the range between 0 and 10, where the number 10 represents the best driveability. For standard cars, driveability indexes greater than or equal to 5.5 are considered acceptable, while luxury cars should have driveability indexes above 6.

Preliminary data from four-cylinder and eight-cylinder three-valve prechamber engine/vehicle configurations indicate acceptable driveability characteristics.

While the large prechamber engine concept has undergone some durability and driveability testing in vehicle installations, detailed performance data are currently not available.

4.1.1.5 Potential Problem Areas

The principal prechamber engine development goal established by Ford was the achievement of the 1977 federal emission standards (0.41 gr/mile HC; 3.4 gr/mile CO; and 2.0 gr/mile NO_x), combined with acceptable driveability and fuel economy without the use of other emission control devices such as catalysts, EGR, and exhaust manifold air injection. None of the prechamber engine configurations tested to date has completely satisfied this goal.

Most likely the torch ignition engine requires some EGR (~ 10%) as well as a thermal reactor for HC and CO control. The durability of the prechamber and spark plug has not yet been demonstrated.

The three-valve prechamber configuration might have the best chance of approaching Ford's development goal. However, the 400 CID prechamber engine requires a small thermal reactor for HC and CO control. Sufficiently low NO_x levels might be achieved in future designs to meet the 0.4 gr/mile limit without EGR. Achievement of the 1977 HC and CO standards is expected to require incorporation of a catalytic or thermal reactor, particularly for larger engines. In addition, some intake air preheating and exhaust insulation might be required to minimize HC and CO during the cold-start phase of the test cycle.

The three-valve prechamber introduces additional complexity in the engine and a high degree of extensive manufacturing development would be required to reduce system cost.

The prechamber with fuel injection was found difficult to control because of the very small quantities of additional fuel supplied to the prechamber. This resulted in rough engine operation, misfire, and loss of power, leading to program cancellation.

The large prechamber which relies on stratification of the injected fuel to generate a rich mixture near the spark plug and a lean mixture in the remaining part of both chambers has a number of unresolved problems related to fuel injection, fuel atomization, charge stratification, and wall wetting.

4.1.1.6 Current and Projected Status

While a number of Ford's prechamber engine concepts have been subjected to substantial development efforts, none of these designs is ready for application to production engines.

The small torch ignition prechamber concept requires additional emission control to meet future emission standards. However, in principle, the concept is applicable as a retrofit device to existing engines at an undetermined increase in cost. Further development work on this engine is planned by Ford.

The three-valve prechamber engine concept is probably the most advanced and successful configuration tested by Ford to date. Because of its favorable emission and fuel consumption characteristics, Ford intends to continue the development of that concept which eventually might be used by Ford in a new production engine.

The future of the large prechamber is currently unknown. Further work on this concept is being conducted by Ford using a larger, single-cylinder engine. Also, Ford was negotiating a research and development contract with Ricardo and Company of England, covering work on prechamber spark-ignition modifications of the Comet diesel engine. However, unless this concept proves to be superior to Ford's open chamber stratified charge (PROCO) engine with respect to performance and cost, the prospects of this configuration remain doubtful.

4.1.2 General Motors Corporation

Stratified charge engine research and development work at General Motors dates back to the patent disclosure in 1921 of a three-valve prechamber engine (Ref. 4-4). The principal design objective of this early engine was the achievement of improved fuel economy combined with a lower fuel octane requirement relative to the conventional spark-ignition engines that were in production to that time. With the advent of automotive exhaust emission regulations, General Motors has resumed its prechamber engine development efforts and is in the process of conducting both multicylinder and single-cylinder engine work (Refs. 4-5 and 4-6). The multicylinder engine programs are primarily aimed at the development of an engine suitable for mass production. Conversely, the single-cylinder efforts are intended to enhance General Motors' fundamental understanding of the combustion and fluid dynamic processes occurring in prechamber engines and to permit the measurement of certain thermodynamic and flow field parameters which are required as input to the mathematical engine model currently under development at General Motors.

Initial emission and performance data released by General Motors on these two programs are discussed in the following subsections.

4.1.2.1 Multicylinder Engine Program

4.1.2.1.1 Engine Description

One of the multicylinder prechamber engine configurations currently under development at General Motors is depicted in Figure 4-3 (Ref. 4-5). In this design, which is known as the General Motors jet ignition stratified charge engine (JISCE), a rich air-fuel mixture is inducted into the small prechamber through a cam-actuated third valve, while a lean mixture is fed to the main combustion chamber. A modified carburetor is utilized which has separate throats and metering systems for the prechamber and main chamber flow circuits. To assure good combustion in both chambers, the engine incorporates an early fuel evaporation (EFE) system in the intake manifold similar to that employed in many 1975 model year automobiles. Ignition of the combustible air-fuel mixture is accomplished by means of a conventional spark plug located in each prechamber. Upon ignition, a jet of fuel-rich combustion gases is expelled from the prechamber into the main chamber through a restricting orifice. The combustion process is then completed in the main chamber. In some of the tests, a thermal reactor or catalytic converter was incorporated in the exhaust system for further HC and CO reduction.

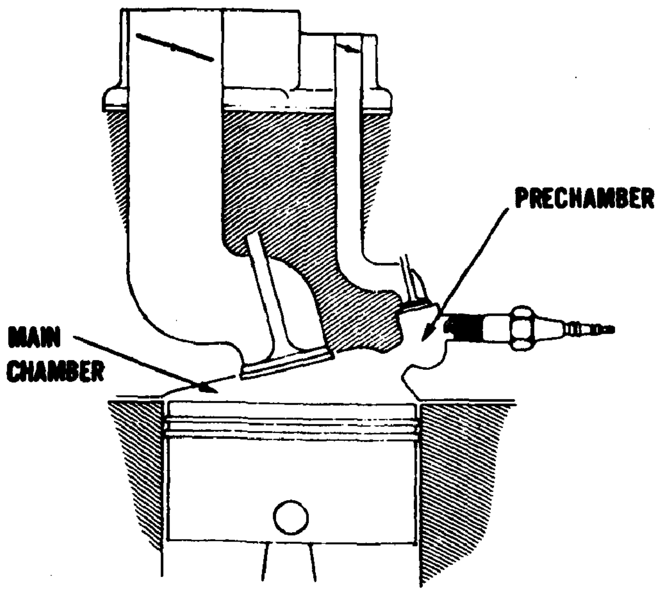


Figure 4-3. General Motors jet ignition stratified charge engine schematic (Ref. 4-5)

Based on many laboratory tests, General Motors has identified a number of important prechamber design and operating parameters which have a significant effect on the performance, emissions, fuel economy, and response characteristics of the engine. These include the prechamber volume and shape, nozzle size and shape, prechamber and overall air-fuel ratios, and prechamber and main chamber flow rates. While these parameters are considered to be proprietary by General Motors, they have indicated that relatively small prechamber volumes (less than 10 percent of the total clearance volume) are utilized in all its JISCE designs currently under consideration (Ref. 4-7).

4.1.2.1.2 Materials and Manufacturing

Although the prechamber engine work at General Motors has not progressed much beyond the feasibility development stage, they have conducted initial studies relative to materials, production line modifications, and cost requirements associated with the potential introduction of JISCE engine-powered automobiles.

Except for the prechamber, the materials projected for use in the manufacture of JISCE engines would be identical or similar to those used in current automotive engines. Most likely the prechamber and the interconnecting flow passage would be fabricated from austenitic steel. Considering the relatively small quantities of material required for the prechambers, General Motors does not foresee any difficulties in securing an adequate supply of this chromium-based alloy (Ref. 4-7).

According to General Motors, the manufacture of prechamber engines would require the incorporation of significant modifications in the existing production lines. For example, machining of the cylinder head (which includes the prechambers) would be considerably more involved than in current production engines and might require the addition of new machine tooling. Other engine components that would have to be redesigned include the dual-flow carburetor, the prechamber intake valve, and the valve-actuating mechanism (Ref. 4-7).

General Motors feels that the added complexity of the prechamber engine and the requirement of new machine tooling would raise the cost of this engine relative to that of current production engines (Ref. 4-7). However, the magnitude of this cost increase has not yet been evaluated by General Motors. In view of the substantial capital investment requirements associated with future prechamber engine mass production and the long production leadtimes of the machine tool industry, General Motors feels that only a gradual phasing-in of prechamber engines would be feasible, rather than simultaneous conversion of all engine lines. While the future of prechamber engines at General Motors is dependent upon the successful completion of the current development programs and future emission regulations, General Motors feels that production of one or two prechamber engine lines might be feasible within about 24 to 30 months after program approval (Ref. 4-7).

4.1.2.1.3 Applications

To date, General Motors has incorporated its JISCE prechamber concepts into a number of experimental 140 CID and 350 CID engines. These modified engines were incorporated into vehicles which were tested on the chassis dynamometer in accordance with the 1975 Federal Test Procedure using a variety of rear axle ratios and inertia weights. The 140 CID engine/vehicle configuration was tested with inertia weights of 2500 lb and 3000 lb, while the 350 CID engine/vehicle was tested with 4000-lb and 5000-lb inertia weights. In addition, General Motors has conducted a limited amount of vehicle road testing (Ref. 4-5).

4.1.2.1.4 Emission Characteristics

Table 4-5 presents HC, CO, and NO_x emission data from three experimental 350 CID prechamber engine/vehicle configurations which were operated on gasoline over the Federal Driving Cycle using an inertia weight setting of 5000 lb (Ref. 4-5). Similar HC and CO emissions and somewhat lower NO_x levels were obtained with propane. Test configuration No. 1 incorporated an early fuel evaporation (EFE) system and some spark advance during the cold-start phase of the driving cycle to minimize the inherently high HC and CO exhaust concentrations characteristic of engine cold starts. In addition to EFE and cold spark advance, configuration No. 2 included a thermal reactor, while a catalytic converter was employed in configuration No. 3 (Ref. 4-5).

To maintain acceptable vehicle driveability, the overall air-fuel ratio supplied to the engine during these tests was controlled

Table 4-5. EMISSION TEST DATA FROM GENERAL MOTORS 350 CID PRECHAMBER ENGINE AND 350 CID CONVENTIONAL ENGINE POWERED AUTOMOBILES; 1975 FEDERAL TEST PROCEDURE; 5000-lb INERTIA WEIGHT (Ref. 4-5)

Engine configuration	EGR[a]	AIR[b]	EFE[c]	Cold spark advance	Thermal reactor	Catalytic converter	No. of cars	No. of tests	Average emissions, gr/mile			Emission range, gr/mile		
									HC	CO	NO_x	HC	CO	NO_x
Prechamber No. 1	no	no	yes	yes	no	no	1	2	0.9	4.5	1.7	0.7-1.0	4.3-4.6	1.6 1.7
Prechamber No. 2	no	no	yes	yes	yes	no	1	5	0.26	3.0	1.5	0.2-0.33	2.9-3.3	1.4 1.6
Prechamber No. 3	no	no	yes	yes	no	yes	1	3	0.19	0.9	1.5	0.17-0.2	0.8-1.0	1.4 1.5
1974 Production	yes	yes	-	no	no	no	4	8	1.2	25.2	1.9	0.9-1.8	18.1-32.9	1.6 2.5

[a]Exhaust gas recirculation system
[b]Air injection reactor system
[c]Early fuel evaporation system

to about 20:1. Conversely, with increasing power output demand, the air-fuel ratio was reduced until both prechamber and main chamber mixture ratios were identical at wide-open throttle. In this case, the engine operated in a homogeneous mode similar to a conventional spark-ignition engine.

Test data from a number of 1974 production vehicles equipped with conventional 350 CID engines also are listed in Table 4-5, for comparison. These engines employed exhaust gas recirculation (EGR) for NO_x control and an air injection reactor (AIR) system for added HC and CO control. The principal component of the AIR system is a vane-type air pump to provide compressed air for injection into the exhaust manifold.

As indicated in Table 4-5, the average raw CO emissions from the prechamber engine are less than 20 percent of those emitted by the conventional engine, reflecting the lean air-fuel ratio operation of the prechamber engine. However, the raw HC emission of the prechamber engine is only slightly lower than that of the conventional engine and exceeds the 1977 federal emission standard by a wide margin. In an effort to meet this standard, General Motors has added a thermal reactor (configuration No. 2) or a catalytic converter (configuration No. 3), and has demonstrated substantially lower HC and CO emissions with these installations. It should be noted, however, that the data shown in Table 4-5 are from new systems and do not include allowances to account for performance degradation of the emission control system due to mileage accumulation, prototype-to-prototype production slippage, and production quality control tolerances.

Additional NO_x reduction has been achieved by General Motors by further leaning the mixture and/or by incorporating exhaust gas recirculation (EGR). However, the observed improvements were accompanied by sizable losses in fuel economy and rapidly increasing

HC emissions. This is illustrated in Figure 4-4, showing measured HC and fuel economy data over the Federal Driving Cycle as a function of the selected NO_x level (Ref. 4-5). It is conceivable that the HC emissions might be reduced somewhat by means of spark retard at the expense of additional degradation in fuel economy. The EGR utilized in these tests was inducted into the intake manifold just below the main chamber carburetor. EGR addition into the prechamber proved to be less desirable, causing combustion roughness.

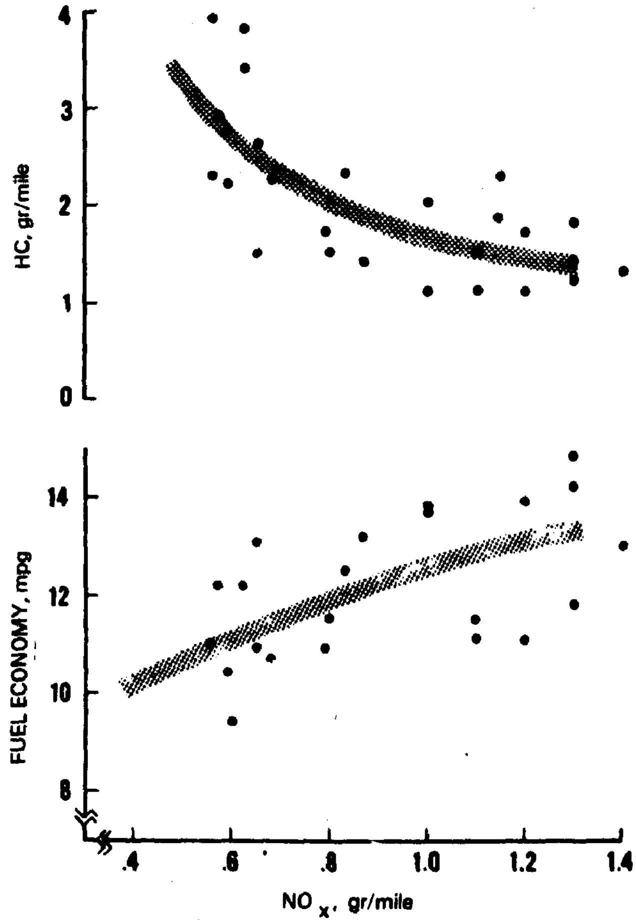


Figure 4-4. Effect of selected NO_x level on HC emission and fuel economy over the Federal Driving Cycle; 4000-lb inertia weight; 350 CID JISCE engine (Ref. 4-5)

4.1.2.1.5 Fuel Consumption Characteristics

According to General Motors, the fuel economy of the prechamber powered vehicle without exhaust control devices is similar to that of the 1974 production vehicles listed in Table 4-5. However, under certain steady-state operating conditions, the specific fuel consumption of the uncontrolled prechamber engine has been shown to be slightly better than that of the conventional engine (Ref. 4-7). Conversely, the fuel economy of engine/vehicle configurations No. 2 and No. 3 (Table 4-5) was somewhat lower than for configuration No. 1, because retarded spark settings were utilized to achieve higher exhaust gas temperatures and higher HC and CO conversion efficiencies of the exhaust reactor systems.

Currently, all General Motors prechamber engines are being operated with intake throttling similar to conventional spark-ignition engines. While unthrottled operation would be beneficial from a fuel savings point of view, the driveability of the vehicle in this mode has been shown to be unacceptable (Ref. 4-7).

4.1.2.1.6 Engine and Vehicle Performance

Stratified charge engines, like their conventional counterparts, suffer from poor combustion when operated at low load, resulting in high HC and CO emissions. The degradation in combustion quality is attributed by General Motors to the high residual gas fraction remaining in the cylinder at the high manifold vacuums prevailing under these conditions.

Vehicle acceleration also poses a potential problem area in prechamber engines because of the requirement of power enrichment under these conditions. As a result, the benefits of low emissions derived from lean engine operation are largely lost during these operating periods. Since the Federal Driving Cycle includes a number of rather severe transients, the overall emission performance of the engine could be adversely affected (Ref. 4-5).

The torque versus speed correlation of the General Motors prechamber engine at wide-open throttle is similar to that of the nonmodified engine, although the prechamber engine suffers a small torque loss of the order of 1 to 2 percent. This loss is attributed to the higher heat loss in the prechamber, combined with a small reduction in engine compression ratio due to the addition of the prechamber and the pressure drop across the passage connecting the prechamber and main chamber. Of course, in a new engine design, this torque loss would be somewhat lower since the compression ratio could be increased to the level of conventional engines.

Based on the available test data, General Motors has determined that the octane requirement of its prechamber engine at wide-open throttle is similar to that of equivalent conventional engines. In this case, the air-fuel ratio of the mixtures supplied to the prechamber and main chamber is identical (of the order of 13:1) and essentially equal to the mixture ratio used in conventional engines.

4.1.2.1.7 Potential Problem Areas

While General Motors' current prechamber engine design is quite complex relative to conventional spark-ignition engines, no major production problems are anticipated by General Motors except for the need of some new machine tooling and the related capital investment requirements. However, prechamber conversion of current production engines would be quite involved, requiring major redesign efforts (Ref. 4-5).

Aside from the higher projected production cost of prechamber engines, General Motors is concerned about the prospects of mass-producing the dual carburetor used in its prechamber engine. This unit demands very close tolerances and uses an advanced control system to assure delivery of the desired air-fuel mixtures to both the prechamber and main chamber of each cylinder under all operating conditions. According to General Motors, the development of a prototype carburetor which is designed to meet these requirements is in progress (Ref. 4-7).

Other potential problem areas include the relatively high HC and CO emissions, particularly at NO_x levels below about 1.5 gr/mile. In this case, the use of a thermal reactor or catalytic converter would be required to meet the 1975-1977 federal emission standards (Ref. 4-5).

4.1.2.1.8 Current and Projected Status

Based on the multicylinder carbureted jet ignition prechamber engine work conducted to date, General Motors has concluded that prechamber engines have inherent advantages over conventional engines in terms of extending the lean limit and achieving lower CO and NO_x emission levels (Ref. 4-5). The prechamber engine has the potential of meeting a 2 gr/mile NO_x standard with reasonable fuel economy without the use of EGR. However, after-treatment devices would be required to meet the 1977 HC and CO standards. In the opinion of General Motors, promulgation of more stringent NO_x standards might result in a substantial reduction of the prechamber engine development efforts currently pursued by the automotive industry.

Although the current engine production lines would require modification, General Motors foresees no insurmountable production-related problems associated with the manufacture of prechamber engines. However, the production cost of prechamber engines would be higher than that of current conventional engines because of the more complex cylinder head and carburetor designs projected for prechamber engines.

While some road testing has been conducted by General Motors on a number of prechamber powered vehicles, formal driveability test programs have not yet been initiated. However, prechamber engines are expected to show the same durability characteristics as conventional engines. Similar projections were made by General Motors relative to the driveability of prechamber engine powered automobiles.

Future development work on General Motors' jet ignition prechamber engines is scheduled to include further optimization of the prechamber size and shape, development of a suitable dual carburetor, and testing of fuel-injected prechamber engines of the type currently under consideration by Volkswagen of Germany. In addition, emphasis will be directed towards unthrottled engine operation to improve the fuel consumption characteristics of the current configurations.

4.1.2.2 Single-Cylinder Engine Program

4.1.2.2.1 Engine Description

General Motors' single-cylinder jet ignition stratified charge engine is shown schematically in Figure 4-5 (Ref. 4-6). The engine represents a single-cylinder modification of a General Motors 400 CID V-8 engine, employing a special cylinder head and actuating mechanism for the third valve which controls the prechamber flow. Important engine design parameters are listed in Table 4-6. The spherical prechamber, which is located on one side of the cylinder, has a diameter of 0.875 in., resulting in a prechamber volume of about 5.5 percent of the total clearance volume. The prechamber is connected to the main chamber by means of a small communicating passage. Two passage sizes, 0.375-in. diameter and 0.125-in. diameter, have been evaluated to date. In all tests, a fuel-rich mixture was supplied to the prechamber through an auxiliary cam-actuated

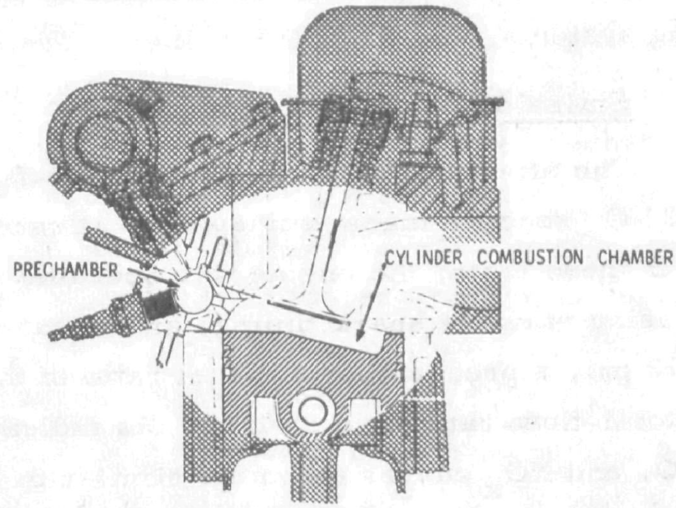


Figure 4-5. General Motors single-cylinder jet ignition stratified charge engine (Ref. 4-6)

Table 4-6. GENERAL MOTORS SINGLE-CYLINDER JET IGNITION STRATIFIED CHARGE ENGINE GEOMETRY (Ref. 4-6)

Parameter	Value
Bore, mm	104.65
Stroke, mm	95.25
Connecting rod length, mm	168.25
Compression ratio	7.95
Prechamber volume ratio	0.055
Prechamber valve lift, mm	2.57
Prechamber valve duration, rad	0.7π
Prechamber orifice diameter, mm	9.53 and 3.17

intake valve. The prechamber valve timing was adjusted such that its peak lift coincided with that of the main chamber valve.

Propane was used exclusively in General Motors' single-cylinder test work. The fuel and air flow-rates into the prechamber and main chamber were controlled by independent fuel-metering systems.

4.1.2.2.2 Emission Characteristics

The effect of the overall engine air-fuel ratio on the HC, CO, and NO_x specific mass emissions is illustrated in Figure 4-6 (Ref. 4-6). In these tests, the engine was operated at a constant speed of 1600 rpm using variable spark timing, an indicated mean effective pressure of 70 psi, a prechamber air-fuel ratio of 8.0, and a prechamber-to-total-flow-rate ratio of 0.01. As indicated, the shapes of the HC, CO, and NO_x curves are quite similar to those of conventional engines. NO_x decreases rapidly with increasing air-fuel ratio, while HC and CO reach a minimum at an air-fuel ratio of about 17.5, and increase again for leaner mixtures.

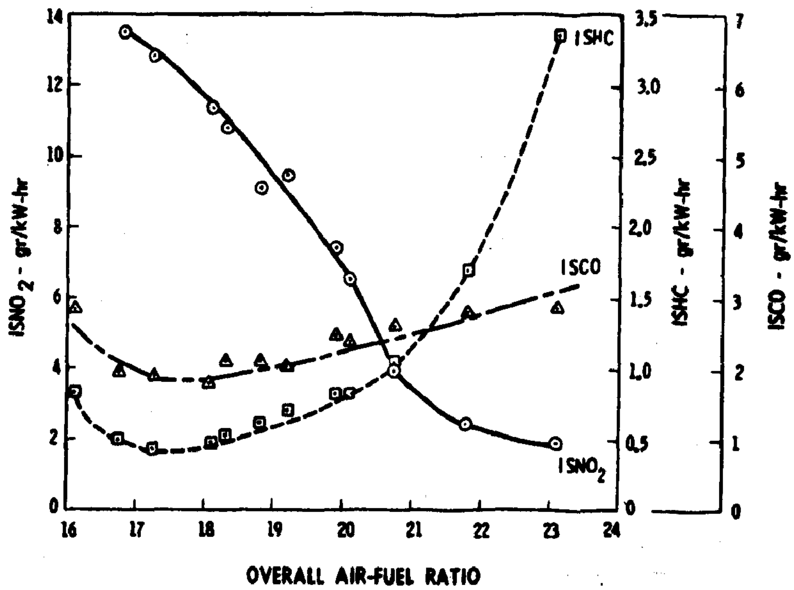


Figure 4-6. HC, CO, and NO_x specific mass emissions versus overall air-fuel ratio; General Motors single-cylinder jet ignition stratified charge engine (Ref. 4-6)

Tests conducted over a range of prechamber-to-main-chamber-flow-rate ratios indicate that HC and CO change very little, while NO_x tends to increase with increasing flow-rate ratio.

Figure 4-7 shows the effect of the prechamber supply air-fuel ratio on the emissions, using a prechamber-to-total-flow-rate ratio of 0.066 and an overall air-fuel ratio of 22. Reducing the prechamber air-fuel ratio from 22 (corresponding to homogeneous mixture operation of the engine) to about 8 has very little effect on the emissions. However, further reduction of the air-fuel ratio to about 3 is accompanied by a substantial increase in CO, a moderate reduction in NO_x, and essentially constant HC. Somewhat different trends were obtained for a flow-rate ratio of 0.0075. The differences in trends are attributed to the different mixture equivalence ratios in the prechamber at the time of ignition and the mixture stratification in the main chamber which occurs as a result of excess mixture supplied to the prechamber for flow-rate ratios above about 0.007. Since the volume of the prechamber is only about 0.7 percent of the total cylinder displacement, some of the flow entering through the prechamber valve passes out into the main chamber. The composition of the mass pushed back into the prechamber during the following compression stroke is strongly affected by the air-fuel ratio of the mixture in the vicinity of the communicating passage.

Besides affecting the mixture strength in the prechamber, the relative size of the prechamber has an influence on the degree of mixture stratification achieved in both the prechamber and main chamber. Since the fluid mechanics involved are extremely complex, General Motors has initiated the development of a mathematical model in order to gain a better understanding of these processes. The model involves the application of the mass and energy conservation equations in differential form and requires a number of meaningful flow field assumptions. These include the use of (1) ideal gas correlations, (2)

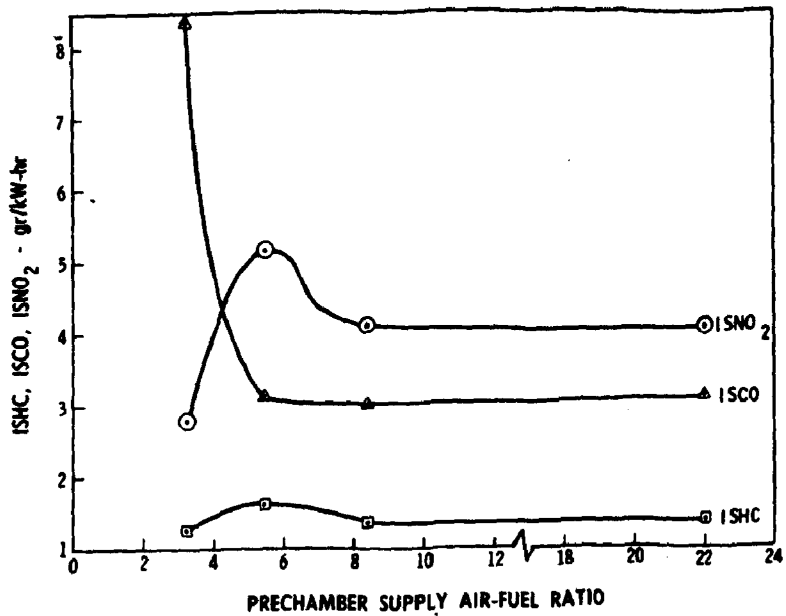


Figure 4-7. HC, CO, and NO_x specific mass emissions versus prechamber supply air-fuel ratio; prechamber flow-rate ratio 0.066; General Motors single-cylinder jet ignition stratified charge engine (Ref. 4-6)

homogeneous gas composition in each chamber except during the combustion process, (3) quasi-steady-state isentropic flow across the intake valve and the communicating passage, (4) convective heat transfer into the prechamber walls in accordance with Woschni's correlation, (5) separate heat release rates in the prechamber and main chamber using Wiebe functions, and (6) a four-zone combustion model which divides the prechamber and the main chamber into burned and unburned zones. The model determines NO_x in accordance with the Zeldovich mechanism, which was extended to account for the flow of NO from the prechamber into the main chamber (Ref. 4-6).

4.1.2.2.3 Current and Projected Status

Comparison of theoretical and experimental data indicates that the model, in its present state, has the capability of adequately describing the combustion processes occurring in the General Motors JISCE prechamber engine. Further plans in the single-cylinder engine area include additional basic experiments to determine the effect of various prechamber design and operating parameters on the performance of the engine, as well as incorporation of certain improvements into the engine model.

4.1.3 Honda Motor Company

The Honda CVCC (Compound Vortex Controlled Combustion) prechamber engine concept has been under development since 1969. Initially, strong turbulence and vortex effects were sought by Honda to improve the combustion process. However, later developments were directed away from the vortex concept toward controlled mixture stratification. While the CVCC designation was retained by Honda, it is no longer representative of the concept. The engine is in mass production and is used by Honda in its 1975 Civic automobile exported to the United States.

4.1.3.1 Engine Description

The Honda prechamber occupies a small fraction of the total combustion chamber volume (~ 10%). As illustrated in Figures 4-8 and 4-9, it has a separate intake valve (third valve) which is operated by a separate cam and rocker arm arrangement (Ref. 4-8).

The CVCC engine incorporates a three-barrel carburetor. The small barrel supplies a fuel-rich mixture (air-fuel ratio approximately 8:1) to the prechamber, while the other two barrels supply a lean mixture (air-fuel ratio approximately 20:1) to the main chamber. At the end of the compression stroke, the spark plug is surrounded by a rich mixture, while a near-stoichiometric mixture is formed in the vicinity of the prechamber outlet (Ref. 4-9). Ignition takes place in the prechamber by means of a conventional spark plug. The lean mixture in the main chamber is then ignited by the hot gases

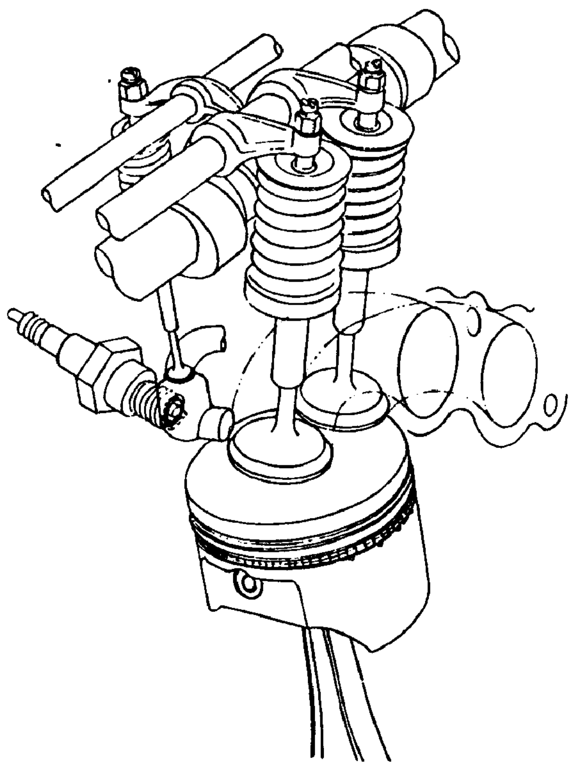


Figure 4-8. Honda CVCC divided chamber stratified charge engine (Ref. 4-8)

emanating from the prechamber passage. As a result, the peak combustion temperature and the rate of NO_x formation are reduced without adversely affecting the oxidation of HC and CO. This is illustrated in Figure 4-10, showing the engine cylinder pressure and temperature of a CVCC and standard Honda engine as a function of the crank angle (Ref. 4-8).

To minimize the emissions of HC and CO during a cold start, the intake manifold is heated by the exhaust gases. This results in a rapid rise of the mixture temperature to the desired level under all driving conditions.

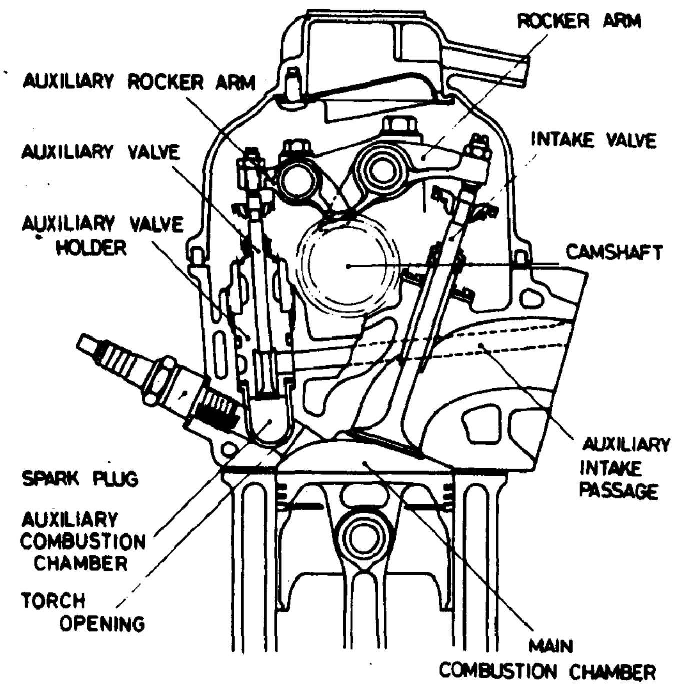


Figure 4-9. Cutaway of Honda CVCC cylinder head (Ref. 4-9)

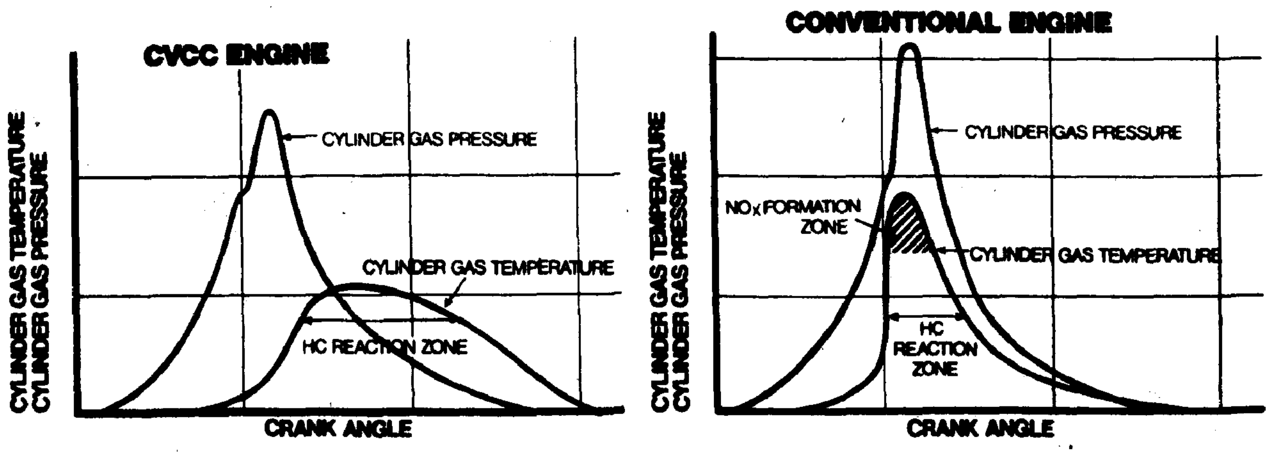


Figure 4-10. Cylinder gas pressure and temperature distributions; Honda CVCC and conventional engines (Ref. 4-8)

The oversized exhaust manifold is of dual-wall construction to increase the residence time of the exhaust gases and to maintain a high temperature environment for the purpose of reducing HC and CO. Thus, the exhaust manifold performs the function of a lean thermal reactor.

To gain a better understanding of the effect of certain engine and operating parameters on the performance and emissions of the CVCC engine, Honda has performed considerable research and development work on its 2-liter, four-cylinder CVCC engine (Refs. 4-9 and 4-10). Based on this work, Honda has concluded that engine performance is strongly affected by a number of parameters. These include (1) the air-fuel ratio of the prechamber and main chamber, (2) the ratio of prechamber to total clearance volume, (3) the ratio of prechamber flow passage to volume, and (4) the positioning of the prechamber relative to the main chamber. It was fortuitous that the lowest values of NO_x were obtained at relatively low values of specific fuel consumption. Further improvement in fuel economy has been obtained by Honda at the expense of higher NO_x levels.

Honda has formulated a mathematical model of the CVCC combustion chamber which permits the analysis of the three air-fuel ratio zones formed in the engine as a function of the important operating variables. These zones include a rich region in the prechamber, a near-stoichiometric region near the prechamber flow passage, and a lean region in the remaining part of the main chamber. This model was used to compute the combustion temperature, and the HC, CO, and NO_x emissions of the engine over a range of operating points (Ref. 4-10). In the main, good agreement was achieved between the theoretical and experimental data.

4.1.3.2 Emission Characteristics

A considerable amount of emission data is available from vehicles equipped with the Honda CVCC system. These include a 1975 Honda Civic, a 1972 General Motors Vega, modified for CVCC operation, and a 1973 Chevrolet Impala, incorporating CVCC. Pertinent engine/vehicle design and operating data are listed in Table 4-7. These vehicles were tested without additional emission control devices, such as catalysts, EGR, and secondary air injection into the exhaust manifold.

Table 4-7. CVCC ENGINE/VEHICLE CONFIGURATIONS

	Honda CVCC	Experimental Vega/CVCC	Experimental Impala/CVCC
Vehicle	1975 Civic	1972 G.M. Vega	1973 G.M. Impala
Transmission	4-speed manual	4-speed manual	3-speed automatic
Accessories	-	-	air cond., power steering, power brakes
Cylinder number	4	4	8
Displacement, CID	119	140	350
Power output (std.), hp	-	72	160
Power output (CVCC), hp	65	73	160
Compression ratio	8.0:1	8.0:1	8.5:1
Ignition timing	3° BTDC at 900 rpm		-

Emission data from three different 2-liter Honda CVCC Civic vehicles are presented in Table 4-8 (Ref. 4-11). The tests were conducted by the EPA in accordance with the 1975 Federal Test Procedure, using inertia weights of 2000 lb and 3000 lb. The data listed in the table represent averages from a number of test runs conducted on each vehicle.

Table 4-8. EMISSIONS AND FUEL ECONOMY FROM THREE 2-LITER HONDA CVCC VEHICLES; 1975 FTP (Ref. 4-11)

Vehicle	Inertia weight, lb	Emissions, gr/mile			Fuel economy, mpg	
		HC	CO	NO_x	1975 FTP	1972 FTP
Low-mileage car No. 3652	2000	0.18	2.12	0.89	22.1	21.0
Low-mileage car No. 3652	3000	0.28	3.08	1.56	19.4	18.7
50,000-mile car No. 2034	2000	0.24	1.75	0.65	21.3	19.8
Low-mileage car No. 3606 (backup)	2000	0.23	2.00	1.03	20.7	19.5

As indicated, the 2-liter Honda CVCC vehicle meets the 1977 federal emission standards (0.41 gr/mile HC; 3.4 gr/mile CO; 2.0 gr/mile NO_x). Of particular significance is the very low NO_x level of the vehicle when tested at the 2000-lb inertia weight setting and the fact that the system shows very little emission degradation after 50,000 miles. Similar results were obtained on four other vehicles, showing increases of about 16 percent for HC, 12 percent for CO, and 4 percent for NO_x after 50,000 miles (Ref. 4-12). Comparison of the data from vehicle No. 3652 indicates that the emissions increase by about 50 to 70 percent when the inertia weight setting is increased from 2000 lb to 3000 lb.

Steady-state emissions from vehicle No. 3652 are presented in Table 4-9 (Ref. 4-11). As indicated, HC and CO decrease with increasing vehicle speed (and load), while NO_x increases with increasing speed. These trends are attributed to the higher temperatures and pressures obtained at higher engine loads.

Table 4-9. STEADY-STATE EMISSIONS AND FUEL ECONOMY OF 2-LITER HONDA CVCC VEHICLE NO. 3652 (Ref. 4-11)

Vehicle speed, mph	Gear	Emissions, gr/mile			Fuel economy mpg
		HC	CO	NO_x	
Idle	-	0.06[a]	0.23[a]	0.02[a]	12.5[a]
15	2nd	0.08	1.92	0.44	21.0
30	3rd	0.01	0.67	0.50	29.2
45	4th	0.007	0.41	0.75	32.1
60	4th	0.005	0.36	0.645	33.0

[a]Idle data reported in grams/minute

Exhaust particulate data obtained from Honda CVCC and standard vehicles are listed in Table 4-10 (Ref. 4-11). As indicated, the airborne particulate mass emitted by the Honda CVCC vehicle, which was operated on lead-free gasoline, is comparable to that emitted by the 1972 Chevrolet. The total nonairborne particulate mass collected in the Dow dilution system, 27 ft downstream of the tailpipe, during all Honda testing (146 miles) equalled 1.9086 grams or 0.013 gr/mile.

The EPA certification data for the 1.5-liter 1975 Honda CVCC vehicle are presented in Table 4-11 (Ref. 4-10). This vehicle incorporated a four-speed manual transmission and was tested with an inertia weight of 2000 lb. As indicated, the emissions are well within the 1975 federal emission standards, showing no deterioration with mileage accumulation. Slightly lower HC and CO emissions have been reported by Honda (Ref. 4-10).

Table 4-10. AIRBORNE PARTICULATE EMISSIONS FROM HONDA CVCC AND CONVENTIONAL VEHICLES (Ref. 4-11)

Vehicle	Fuel lead content, gr/gal	Particulate emission, gr/mile		
		1975 FTP	Hot 1972 FTP	60 mph steady-state
Honda CVCC	lead-free	0.036	0.040	0.012
1972 Chevrolet	lead-free	-	-	0.009
1971 Chevrolet	0.5	-	-	0.021
1970 Chevrolet	3.0	-	-	0.110

Table 4-11. EMISSIONS OF 1.5-LITER HONDA CVCC 1975 CERTIFICATION VEHICLE; 1975 FTP; 2000-lb INERTIA WEIGHT (Ref. 4-10)

Vehicle	Test miles	Emissions, gr/mile			Fuel consumption mpg
		HC	CO	NO_x	
Certification	4000[a]	0.56	4.34	1.26	27.5
Certification	50,000[b]	0.38	4.05	1.07	25.1
-	Low	0.24	2.42	1.38	25.5

[a]Production vehicle

[b]Prototype vehicle

As previously noted, Honda has incorporated the CVCC system into a Chevrolet Vega and Chevrolet Impala vehicle. The emission data from these vehicles are listed in Tables 4-12 and 4-13, respectively, along with the emissions of the corresponding nonmodified

Table 4-12. EMISSIONS FROM VEGA CVCC AND STANDARD VEGA VEHICLES; 1975 FTP; 2500-lb INERTIA WEIGHT (Ref. 4-10)

Vehicle	Emissions, gr/mile			Fuel economy, mpg
	HC	CO	NO_x	
Vega CVCC low mileage	0.26	2.62	1.16	18.9
Standard Vega (1972 Calif. Spec.)	2.13	10.60	3.80	17.2

Table 4-13. EMISSIONS FROM CHEVROLET IMPALA CVCC AND STANDARD IMPALA VEHICLES (Ref. 4-13)

Test No.	Test procedure	Inertia weight, lb	Emissions, gr/mile			Fuel economy mpg	Comments
			HC	CO	NO_x		
1	1975 FTP	5000	0.27	2.88	1.72	10.5	EPA data
2	1975 FTP	5000	0.23	5.01	1.95	-	High level of CO caused by flooding of prechamber carburetor
3	1975 FTP	5000	0.80	2.64	1.51	-	High level of HC caused by a false hot start
4	1975 FTP	5000	0.32	2.79	1.68	-	
-	1972 FTP hot start	5000	0.18	2.34	1.87	11.5	
-	1972 FTP	5500	1.[illegible]	2.56	2.13	10.4	High HC caused by irregular hot start
-	1975 FTP	5000	1.56	19.33	2.42	10.5	Standard Impala 1973 model

vehicles. The Vega was tested in accordance with the 1975 Federal Test Procedure using a 2500-lb inertia weight. As indicated in Table 4-12, the HC and CO emissions of the Vega/CVCC vehicle are below the 1978 federal emission standards, while NO_x is below the 1977 standard. Comparison of the emissions of the CVCC vehicle with those from the standard Vega vehicle illustrates the superior emission performance of the CVCC concept. Also, the Impala CVCC configuration shows considerably lower emissions than the standard Impala, particularly HC and CO. As expected, the relatively small increase in the inertia weight from 5000 lb to 5500 lb has little effect on the emissions of the Impala vehicle.

Aldehyde emissions from the Impala CVCC vehicle and two standard vehicles (Plymouth Duster and Ford Maverick) are shown in Table 4-14 (Ref. 4-13). As indicated, the average aldehyde-to-hydrocarbon ratio of the CVCC vehicle is 0.063, which is similar to that of the Duster and somewhat higher than for the Maverick.

Table 4-14. ALDEHYDE AND HC EMISSIONS FROM CHEVROLET IMPALA CVCC AND STANDARD PLYMOUTH DUSTER AND FORD MAVERICK VEHICLES; MBTH METHOD (Ref. 4-13)

Vehicle	Test No.	Engine displacement CID	Test procedure	HC emission (composite bag), gr/mile	Aldehydes, gr/mile	Ratio of aldehydes to HC
Impala CVCC	2	350	1975 FTP	0.23	0.0286	0.124
	3		1975 FTP	0.80	0.0355	0.044
	4		1975 FTP	0.32	0.0195	0.061
	2H		1972 FTP	1.38[a]	0.0338	0.024
1973 Duster	b	225	1975 FTP	1.80	0.116	0.065
1973 Maverick	b	302	1975 FTP	2.25	0.104	0.046

[a] Split bag

[b] Averages of three tests

4.1.3.3 Fuel Consumption Characteristics

Fuel economy data for three 2-liter Honda CVCC vehicles tested in accordance with the 1972 and 1975 Federal Test Procedure are listed in Table 4-8. For an inertia weight setting of 2000 lb, the average fuel consumption of the vehicles when measured with the 1972 FTP was 20.1 mpg, which is about 18 percent lower than the 23.8 mpg determined by the EPA for the average 2000-lb 1973 model year certification automobile. As expected, the fuel economy of the CVCC vehicle is slightly higher when measured by means of the 1975 FTP. With respect to the standard Honda Civic vehicle, the fuel economy of the CVCC-powered Civic is about 10 percent lower (Ref. 4-11). On the other hand, for an inertia weight of 3000 lb, the fuel economy of the Honda Civic vehicle with the 2-liter CVCC engine is about 15 percent better than that of the average 3000-lb 1973 certification vehicle. However, when making these comparisons, consideration must be given to the difference in power output capability of the average 3000-lb car and the 2-liter Honda CVCC. This is further illustrated by comparing the data listed in Tables 4-8 and 4-11 which show substantially higher fuel economy for the less powerful 1.5-liter engine.

Table 4-12 presents fuel economy data for the Vega CVCC and standard Vega vehicles as determined by the 1975 Federal Test Procedure. As indicated, the fuel economy of the CVCC vehicle was 18.9 mpg, which represents an improvement of about 10 percent over the standard Vega.

The fuel economy of the Chevrolet Impala CVCC and other standard vehicles in the 5000-lb-inertia-weight class is shown in Table 4-15 (Refs. 4-8 and 4-13). As indicated, the fuel consumption of the CVCC vehicle is comparable to that of the standard Impala and is approximately 10 percent better than the average 1973 vehicle considered in the table.

Table 4-15. COMPARISON OF 350 CID CVCC IMPALA FUEL ECONOMY WITH SIMILAR HOMOGENEOUS CHARGE GASOLINE POWERED 1973 VEHICLES (Refs. 4-8 and 4-13)

Vehicle	Engine displacement, CID	Inertia weight, lb	Axle ratio	Fuel economy, mpg
Impala CVCC	350	5000	3.08	10.4[b]
Chevrolet Impala[a]	350	5000	3.08	10.5[c]
Pontiac Catalina[a]	350	5000	3.23	8.1[b]
Oldsmobile Delta 88[a]	350	5000	3.08	9.9[b]
Oldsmobile Cutlass Supreme Vista Cruiser[a]	350	5000	3.23	9.4[b]
Buick LaSabre[a]	350	5000	3.08	10.5[b]
Standard Vehicle Average				9.5[b]

[a]1973 Certification data

[b]1972 Federal Test Procedure

[c]Computed from bag 1 and bag 2 data

In summary, the CVCC prechamber combines low emissions with good fuel economy. In smaller engines (2 liters or less), there appears to be some fuel economy penalty when compared with 1973 certification cars, due, perhaps, to problems related to accurate metering of the small prechamber fuel flows. However, in larger engines, the CVCC prechamber offers equal or better fuel economy than standard engines.

4.1.3.4 Vehicle Performance Characteristics

In general, the driveability and performance of vehicles equipped with CVCC engines are comparable to those of conventional vehicles. For example, the driveability of the 1.5-liter Honda CVCC

Civic was excellent, with the only peculiarity being a slight difference in engine noise and some loss in engine braking.

According to the EPA, which tested the 2-liter Honda CVCC on the road, the engine was very responsive and the acceleration of the vehicle was very good. No driveability problems were encountered during those tests. The vehicle easily maintained expressway speeds with adequate passing power in reserve. Honda reported 0.25-mile acceleration times of 17.8 sec.

Similar results were observed on the Vega CVCC vehicle. Pertinent performance data of the Vega CVCC and standard Vega vehicles are listed in Table 4-16.

Table 4-16. VEGA PERFORMANCE COMPARISON; 2500-lb INERTIA WEIGHT (Ref. 4-8)

Vehicle	Original Vega (1972 Calif. Spec.)	Vega CVCC (improved April 1973)
Max. power	70 bhp/4500 rpm	70 bhp/4500 rpm
Max. torque	108 ft-lb/4500 rpm	101 ft-lb/2000 rpm
Idle speed, rpm	700	800
Max. speed, mph	92	92
Acceleration time, (SS 0.25 mile), sec	19.7	20.1

No vehicle performance data are available for the Chevrolet Impala CVCC. However, the maximum horsepower and torque of the Impala CVCC were slightly higher than for the standard 1973 Impala, indicating that the performance of the CVCC vehicle should be at least equal to or better than that of a standard vehicle.

The fuel octane requirement of the CVCC is similar to that of a standard engine. Tests carried out with gasoline/methanol mixtures indicated reduced NO_x, but higher HC relative to operation on gasoline.

4.1.3.5 Potential Problem Areas

The substantial mileage accumulated during the development and preproduction phases of the Honda CVCC system gave ample opportunity for the solution of any problems which might have been encountered at various stages of the development cycle.

While no problems have been reported on the CVCC engine, the high exhaust gas temperature ($\sim 1400°F$) associated with late combustion requires the use of better materials for the exhaust valves and exhaust manifolds relative to conventional engines.

Since accurate mixture ratio control is required for the prechamber and the main chamber, sophisticated carburetors are needed to achieve the desired emission and fuel economy levels. This requires close manufacturing tolerances which might create problems in mass production.

4.1.3.6 Current and Projected Status

The Honda Civic vehicle equipped with a CVCC prechamber engine has been in production in Japan since December 1973. This vehicle is now being exported to the United States.

Honda is continuing its efforts related to prechamber improvements with an objective of reducing NO_x closer to the 0.4 gr/mile limit. However, achievent of this goal in production engines appears to be doubtful, since a margin must be left to account for engine-to-engine variability and degradation during mileage accumulation. While some reduction in NO_x has been achieved by means of EGR into the prechamber and main chamber, this approach has resulted in a loss in fuel economy and is currently not regarded by Honda to be a viable solution to the NO_x problem.

Future work will be concerned with improved mixture control for both chambers and further refinement in the combustion chamber geometry. It appears doubtful that fuel injection would be introduced because of the difficulty in metering the very small quantities of fuel required for the prechamber.

In addition to having licensing agreements with Ford Motor Company and Chrysler Motor Corporation, Honda is negotiating with other manufacturers regarding licensing arrangements.

Since incorporation of the CVCC prechamber system would require major modifications to the engine, it is not considered to be a retrofit device for use in existing engines. However, current production engines could be factory-converted to CVCC as demonstrated by Honda on a Vega and Impala vehicle.

Since the CVCC prechamber concept is new compared to conventional automotive engines, further performance improvements of the CVCC might be expected with further development.

4.1.4 Volkswagenwerk A. G.

Prechamber research and development work has been in progress at Volkswagen for a number of years. While most of the published data are from single-cylinder engines, a number of vehicle test programs have been conducted using multicylinder prechamber engine installations.

4.1.4.1 Engine Description

To date, Volkswagen has investigated a number of prechamber engine designs incorporating a variety of prechamber shapes and sizes. These configurations utilize a fuel-rich prechamber mixture provided by a separate carburetor or injection pump, and a fuel-lean main chamber mixture prepared by an independent fuel supply system.

Initially, Volkswagen experimented with a prechamber volume of about 5 percent of the total clearance volume which was formed by a recess in the cylinder head, as shown in Figure 4-11(a) (Ref. 4-14). This arrangement provided some improvement in lean operation by extending the limiting fuel-air equivalence ratio to about 0.63 from a value of about 0.75 for conventional engines. Subsequently, some improvement was obtained with a separate prechamber which was connected to the main chamber through a small passage, as shown in Figure 4-11(b) (Ref. 4-14). In both designs, fuel was injected into the prechamber to form a fuel-rich mixture, while fuel-lean mixtures in the main combustion chamber were obtained from a manually controlled injection system located in the air inlet pipe. While the operational lean limits could be further extended in the second design, the transition to full load caused difficulties. However, the results were encouraging enough to warrant further prechamber engine research.

A concept similar to the Honda CVCC was tested also by Volkswagen. In this design, the fuel was supplied to the prechamber by carburetion using a separate inlet valve (third valve) for flow control, as shown in Figure 4-11(c). The auxiliary carburetor supplied an approximately stoichiometric mixture to the prechamber while the standard carburetor provided a lean mixture to the main chamber. The lean limit of this engine could be extended to air-fuel equivalence ratios of about 0.55 without incurring engine misfire and an appreciable increase in HC (Ref. 4-14). However, the hydrocarbon level remained high over the entire range and carbon monoxide did not achieve the low levels expected for lean mixture operation (Ref. 4-15).

Based on this work, a spherical prechamber concept (second-generation PCI engine) was designed by Volkswagen, as shown in Figure 4-12 (Ref. 4-16). The prechamber volume was about 25 to 30 percent of the total clearance volume and the prechamber was connected to the main combustion chamber by a relatively large passage.

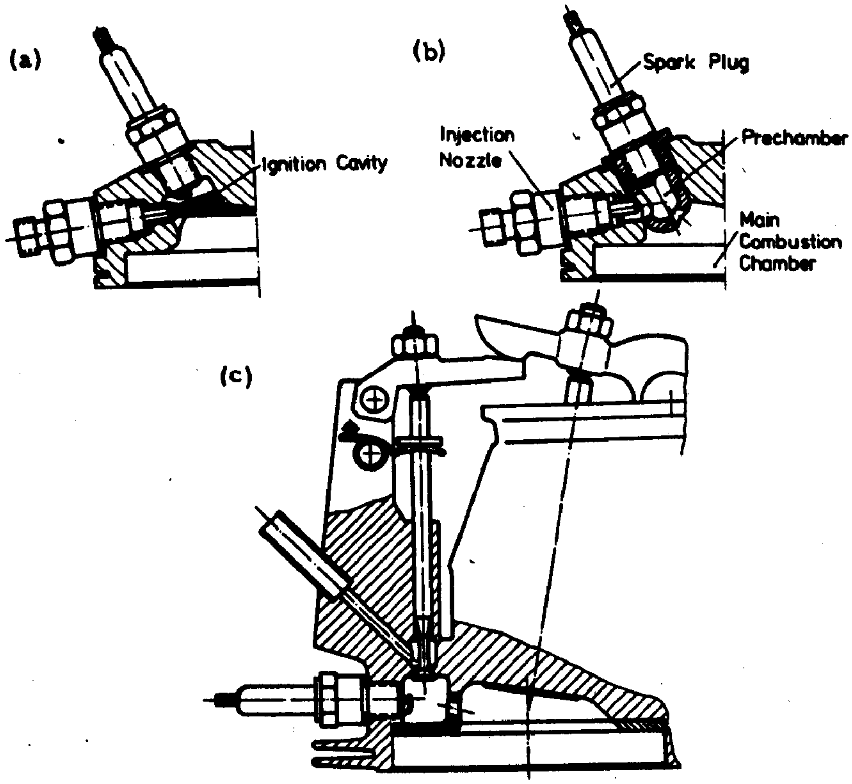


Figure 4-11. Volkswagen prechamber configurations (Ref. 4-14)

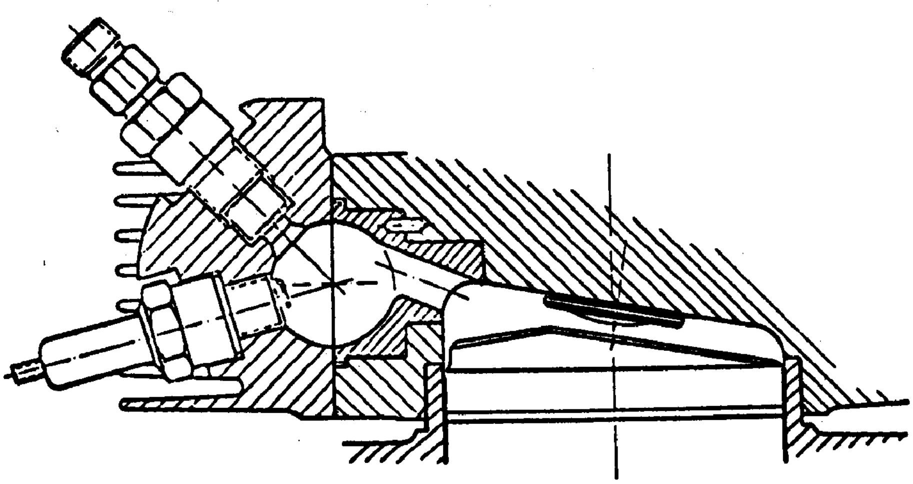


Figure 4-12. Volkswagen spherical valveless prechamber (Ref. 4-16)

This passage was located tangentially to the main chamber, to produce an air swirl for improved mixing. Fuel was injected into the prechamber by a medium-pressure gasoline pump (injection pressures ~ 350 psi), to form a fuel-rich mixture. The start of injection was varied between 50 and 100-deg BTDC with the end of injection kept constant. Surprisingly enough, it was found that injection with low delivery pressure provided the best results with regard to HC emission, probably because of reduced spray penetration and wall wetting. The lean main chamber mixture was formed by fuel injection into the intake port and load control was achieved by lean mixture control. With this concept, the operating range of the engine was extended to fuel-air equivalence ratios of about 0. 3. Intake air throttling was used at idle and up to one-half load, mainly to reduce the HC and CO emission levels.

The third-generation PCI engine designed by Volkswagen was applied to a four-cylinder air-cooled 1. 6-liter engine, as shown in Figure 4-13 (Ref. 4-15). It differed from the second-generation engine primarily in the split design of the prechamber and the smooth flow passage into the main chamber which was utilized to assure low turbulence in the main chamber and low component temperatures. Another improvement consisted of the use of a special medium-pressure fuel injection pump capable of delivering separate fuel charges to the engine prechambers and main chambers. In this design, fuel injection into the prechamber was terminated at 70 deg before top dead-center. The load was controlled by varying the quanity of fuel injected into the intake ports and by air throttling for loads between zero and half-load. During engine warmup, additional fuel was injected into the intake manifold.

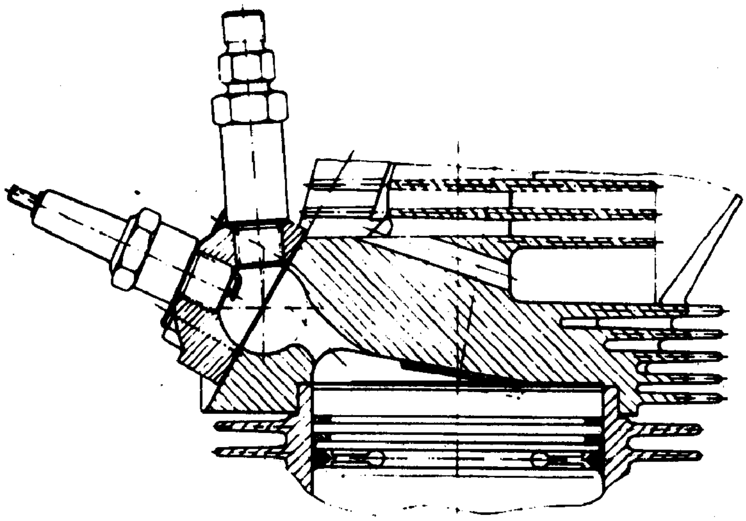


Figure 4-13. Third-generation PCI engine with spherical, valveless prechamber and steel cap (Ref. 4-15)

4.1.4.2 Emission Characteristics

HC and NO emissions from an experimental single-cylinder engine (24 CID and 8.5 compression ratio) with spherical prechamber are shown in Figure 4-14 (Ref. 4-16). The engine was operated on regular gasoline at 2000 rpm and full-throttle, and the size and shape of the prechamber outlet passage was varied. In the high-load regime, the large passage gave the lowest NO emissions, but HC was high. It appears that an intermediate-size passage would be the best compromise between HC and NO.

Tests conducted with variable load, variable ignition timing, and variable fraction of fuel injected into the prechamber resulted in rather high emission levels. Some improvement in the emissions was obtained at part-throttle, particularly in the brake

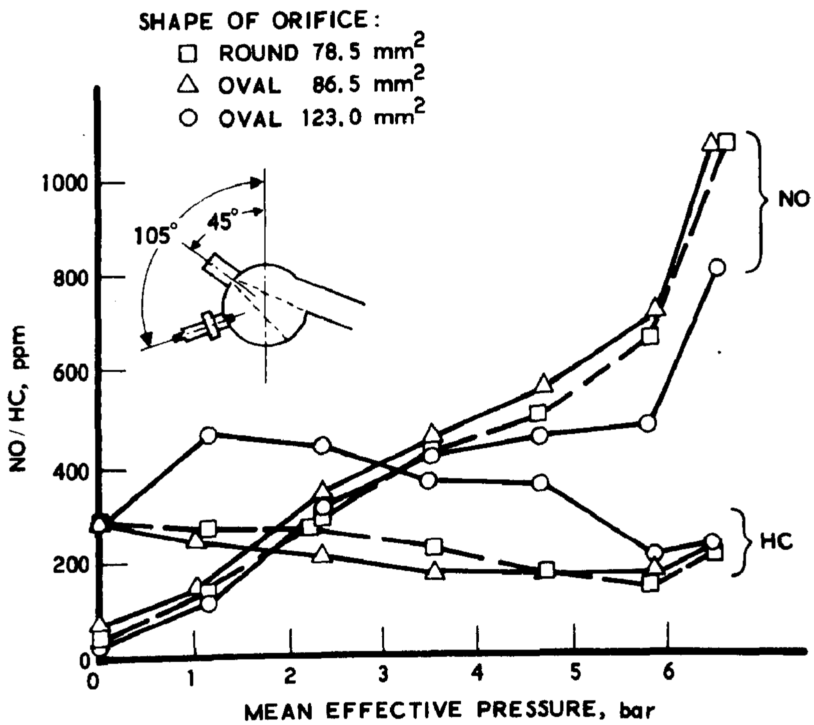


Figure 4-14. Emission characteristics, Volkswagen single-cylinder prechamber engine; 2000 rpm; unthrottled (Ref. 4-16)

mean effective pressure range between 16 and 33 psi, as shown in Figure 4-15 (Ref. 4-16). It appears, however, that a thermal or catalytic reactor would be necessary to meet future HC and CO emission standards. The prechamber was particularly effective in reducing NO_x by about 75 percent, relative to standard production engines (Ref. 4-14).

Additional emission test data from the third-generation PCI, four-cylinder engine are presented in Ref. 4-15. In these tests, the engine was operated on the engine dynamometer and maps of emissions, exhaust gas temperature, and specific fuel consumption were obtained as a function of engine speed and load. Also, tests were performed on the chassis dynamometer using a Volkswagen Beetle as the test vehicle. Results from these tests, conducted in accordance with the 1975 Federal Test Procedure, are shown in Table 4-17.

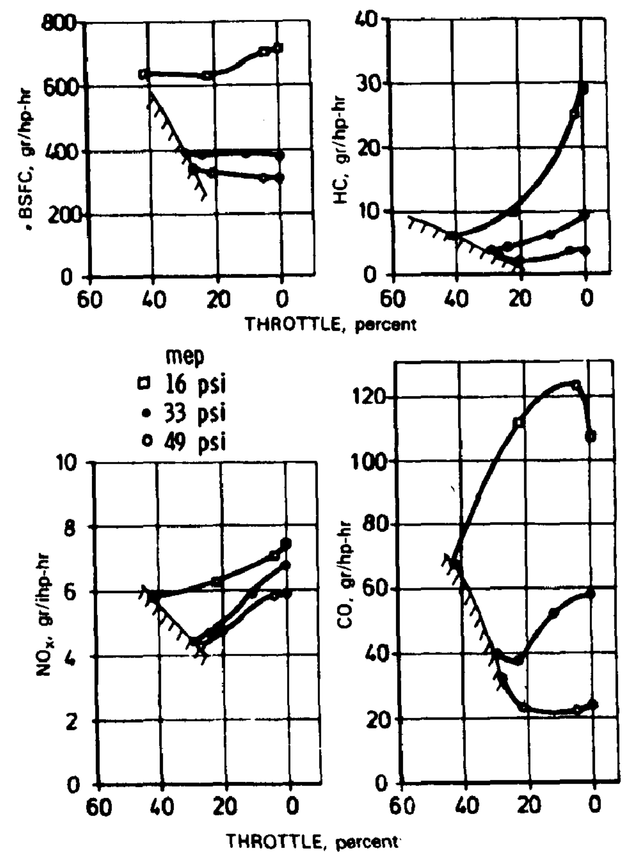


Figure 4-15. Effect of intake air throttling on emissions and fuel consumption; Volkswagen prechamber engine (Ref. 4-16)

As indicated, NO_x is below the 1977 federal emission standard, while CO and particularly HC are considerably higher than the 1977 standards.

Table 4-17. EMISSIONS OF VW BEETLE WITH PCI PRECHAMBER; 1975 FTP; 2250-lb INERTIA WEIGHT; FOUR-CYLINDER, 1.6-LITER ENGINE (Ref. 4-15)

Emission species	gr/mile	Fuel economy, mpg
HC	2.1 - 2.5	22 - 26
CO	4.4 - 8.0	
NO_x	0.75 - 0.96	

4.1.4.3 Fuel Consumption Characteristics

Specific fuel consumption data published by Volkswagen for its single-cylinder engine at full-throttle are shown in Figure 4-16 as a function of load, ignition timing, and air-fuel equivalence ratio (Ref. 4-16). Selected part-throttle data from this engine are shown in Figure 4-15. Although no comparative data are available for a standard VW engine, the values presented are comparable to the fuel consumption of similar-size automotive engines. This was further confirmed on road tests carried out on the VW Beetle equipped with the third-generation PCI engine. The fuel economy of this vehicle as measured in accordance with the DIN 70030 test method (half payload, 75 percent of maximum speed, and 10 percent margin) was 24.5 mpg, compared with 25.6 mpg for the standard Beetle (Ref. 4-15).

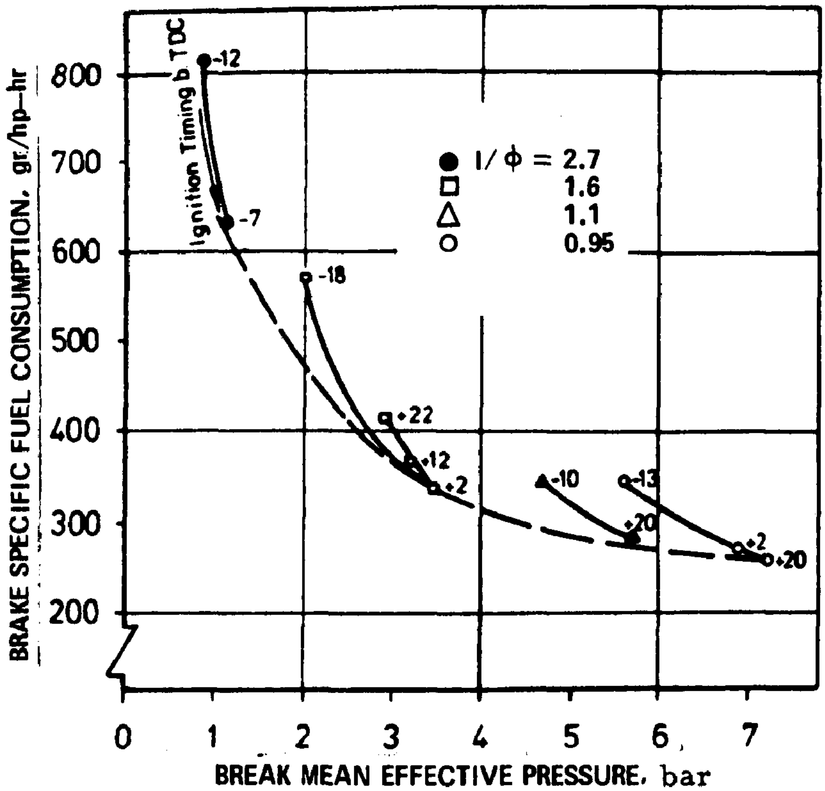


Figure 4-16. Brake specific fuel consumption of VW prechamber engine; full-throttle; 2000 rpm (Ref. 4-16)

4.1.4.4 Vehicle Performance Characteristics

Limited road performance data have been published on the PCI engine equipped VW Beetle. The similarity of the torque curves of this engine to the standard engine, and the small difference in maximum power (48 bhp for the standard engine versus 50 bhp for the PCI) would indicate similar road performance. Test data indicate 0 to 62 mph acceleration times of 20.8 sec for the PCI vehicle and 20.5 sec for the standard vehicle (Ref. 4-15).

4.1.4.5 Potential Problem Areas

The relatively high level of HC and CO encountered in the development of the Volkswagen prechamber engines indicates the need for a thermal or catalytic reactor. The level of NO_x achieved appears to be satisfactory to meet the 1977 federal emission standard (2 gr/mile), although it fell short of the ultimate goal of 0.4 gr/mile.

The use of a plunger-type injection pump and separate injectors to meter the small quantities of fuel to the prechamber may prove to be troublesome with respect to production and achievement of accurate operation in actual vehicle installations.

No data are available on the mechanical durability of the VW prechamber engine concepts and the level of emission control degradation that might occur with mileage accumulation.

4.1.4.6 Current and Projected Status

The Volkswagen prechamber concept is intended for use in new engine configurations. It is not considered to be applicable as a retrofit device for in-use vehicles because of the requirement of a modified cylinder head, fuel injection system, and other emission control devices.

Future work at Volkswagen is scheduled to include modification of the exhaust system and incorporation of a lean thermal reactor for HC and CO reduction.

4.2 OTHER MANUFACTURERS

4.2.1 Combustion Control

Combustion Control, a subsidiary of Systron Donner Corporation, Berkeley, California has been engaged in the development of a prechamber combustion system for potential application as a retrofit unit for existing automobile engines since mid-1971. These efforts are based on the prechamber concept patented by Morghen in 1968 (Ref. 4-17) and on the results of preliminary development work conducted during the 1965 to 1968 time period by the Azure Blue Company, El Dorado, California.

4.2.1.1 Engine/Vehicle Description

The conceptual design of the Systron Donner prechamber configuration is identical to that previously patented by Morghen, et al. (Ref. 4-17). Initial testing of the concept was conducted by the Azure Blue Company which had been formed by the inventors for the purpose of developing and marketing the concept for retrofit applications. As described in the patent disclosure and illustrated in Figure 4-17, the Morghen concept consists of a small prechamber assembly which is mounted in the spark plug well of each engine cylinder. The prechamber occupies about 3 percent of the total clearance volume. According to Ref. 4-17, this size is large enough to assure good ignition of the lean main chamber charge and small enough to prevent the occurrence of excessive pressure rise rates in the main chamber. Other prechamber components include a conventional spark plug, a small intake valve, and a diverging supersonic nozzle which acts as communicating passage between the prechamber and main chamber.

During the suction stroke, a very rich air-fuel mixture ($A/F \approx 3.7$) is inducted into the prechamber through the small

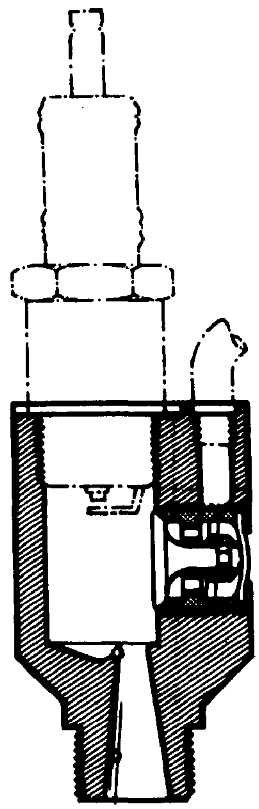


Figure 4-17. Morghen prechamber concept (Ref. 4-17)

pressure actuated intake valve. To assure proper prechamber mixture preparation under all operating conditions, a fuel vaporizer is utilized which is heated electrically or by the engine exhaust. The main chamber is supplied with a lean mixture (A/F $\approx$ 20) which is provided by the stock carburetor. During the compression stroke, a small fraction of the lean mixture from the main chamber enters the prechamber and forms an ignitable mixture (A/F $\approx$ 13.5) in the region surrounding the spark plug. To minimize the flow of unburned mixture back into the main chamber following ignition, a Borda mouthpiece is utilized which reduces the available flow area by forming a vena contracta. Since this phenomenon is not effective during the compression stroke, filling of the prechamber is not impaired as the gradual reduction in the cross-sectional area of the communicating passage inhibits flow separation and assures full utilization of the geometric flow area (Ref. 4-18).

In the prechamber engine configurations under development at Combustion Control, the metering jets of the stock carburetor were replaced by smaller jets and adjustments were made in the ignition timing to achieve constant spark advance near piston top dead-center. In addition, a vacuum relief valve was incorporated for further leaning of the mixture during periods of high manifold vacuum.

While most of the development work conducted to date was performed on a 1964 Ford Falcon automobile with manual transmission (170 CID six-cylinder engine, 8.7:1 compression ratio), a considerable amount of testing was done on a single-cylinder research engine and an automotive V-8 engine, amounting to more than 30,000 vehicle miles. Important vehicle/engine operating parameters are listed in Table 4-18 (Ref. 4-19).

Table 4-18. COMBUSTION CONTROL/FORD FALCON VEHICLE PARAMETERS (Ref. 4-19)

Parameter	Baseline	CCI system
Precombustion chamber	none	1 per cylinder
Spark plug	standard	standard
Auxiliary fuel manifold	none	1 per vehicle
Carburetor jets	standard	modified
Vacuum relief valve	none	1 per vehicle
Ignition timing	4 to 12-deg advance	3-deg constant advance
Air-to-fuel ratio (at 50 mph)	13:1-14:1	18.5:1-21:1

4.2.1.2 Vehicle Emission and Performance Characteristics

The Ford Falcon vehicle, incorporating Combustion Control prechambers and the engine modifications listed in Table 4-18 was tested on the dynamometer at a number of steady-state conditions and over the seven-mode driving cycle. Selected steady-state data are listed in Table 4-19 for different vehicle road loads (Ref. 4-19). As indicated the HC, CO, and NO_x emissions of the prechamber vehicle are substantially lower than those of the baseline (nonmodified) vehicle configuration. For example, at a steady speed of 50 mph HC was reduced by 65 percent while the reductions in CO and NO_x were about 95 percent and 85 percent, respectively. Of particular significance is the fact that effective NO_x control was achieved with this concept, accompanied by an 18 percent improvement in fuel economy.

Table 4-19. STEADY-STATE EXHAUST EMISSIONS-COMBUSTION CONTROL/FORD FALCON AUTOMOBILE (Ref. 4-19)

	Speed, mph	HC, ppm	CO, %	NO_x, ppm	Fuel economy, mpg
CCI system	20	465	<0.1	93	--
	30	400	<0.1	110	--
	40	208	<0.1	177	--
	50	125	<0.1	177	26
	60	160	<0.1	62	--
Baseline (precontrol)	Idle	700	7.6	150	--
	30	520	1.3	200	--
	40	480	2.1	700	--
	50	340	1.35	1300	22
	60	240	0.60	1775	--

Seven-mode, hot-start test data are presented in Table 4-20 (Ref. 4-19). Again, significant emission reduction was accomplished amounting to about 55 percent for HC, 93 percent for CO, and 57 percent for NO_x when operating at an overall air-fuel ratio of 18.5:1. Further NO_x reduction was achieved by operating the engine at an air-fuel ratio of 20:1 at the expense of some increase in HC and CO. The increase in HC and CO is attributed to quench effects occurring in the combustion chamber with very lean mixtures.

Table 4-20. SEVEN-MODE, HOT-CYCLE EXHAUST EMISSIONS—COMBUSTION CONTROL/FORD FALCON AUTOMOBILE (Ref. 4-19)

	Air-fuel ratio	Emissions		
		HC, ppm	CO, %	NO_x, ppm
Baseline (precontrol)	13.0:1	1350	1.62	1334
CCI system	18.5:1	624	<0.10	574
	20.0:1	854	<0.10	407

During the vehicle test program, exhaust filter samples were taken which showed a marked reduction in the amount of particulate matter emitted by the modified engine. Subsequent inspection of the engine cylinder and valves verified the cleaner burning characteristics of the prechamber engine.

Exploratory tests were conducted by Combustion Control with the prechamber equipped Ford Falcon using a variety of fuels including leaded, unleaded, and white gasoline for the main chambers and prechambers as well as propane and hydrogen for the prechambers. According to Ref. 4-19, no significant performance differences were observed in these tests and the engine ran smoothly, even with 70-octane white gas.

4.2.1.3 Driveability, Durability, and Cost

Limited driveability tests conducted by Combustion Control on the Ford Falcon vehicle indicate similar acceleration and deceleration characteristics of the vehicle with both the standard and modified engines. Also, the maximum power output of the two engines was comparable, reflecting the very small change in compression ratio resulting from the installation of the prechambers.

No mechanical failures have been encountered in the course of the test work conducted to date which includes about 20,000 miles of testing on the V-8 engine. The check valves have performed consistently and overheating of the prechamber has never been a serious problem area (Ref. 4-19).

Based on estimated figures provided informally by Combustion Control the prechamber modification cost is about $115 for a typical six-cylinder engine. This figure includes the manufacturing cost of the prechamber and the auxiliary manifold/vaporizer arrangement, replacement of the metering jets of the main carburetor, and an allowance of $25 for the actual installation of the system in the vehicle. However, additional costs due to dealer markup are not included in this estimate (Ref. 4-20).

4.2.1.4 Current and Projected Status

Current efforts by Combustion Control focus on the incorporation of its prechambers into an M-151 light-duty military vehicle under contract to the U.S. Army Tank Automotive Command. The principal objective of this work which was initiated in December 1974, is the achievement of reduced emissions and fuel consumption relative to the standard M-151 vehicle.

Additional proprietary prechamber engine development efforts are being performed in cooperation with a domestic automobile manufacturer. Preliminary test results from this work indicate reduced emission and fuel consumption levels relative to the baseline vehicle (Ref. 4-20).

4.2.2 Eaton Corporation

The Eaton Corporation has been involved in the development and testing of a three-valve prechamber engine for some time. The objective of this inhouse program was to investigate the performance characteristics of this engine and its potential as a low emission device (Ref. 4-21).

4.2.2.1 Engine Description

The Eaton prechamber was incorporated into one cylinder of a four-cylinder 2-liter Pinto engine. Other modifications incorporated in the engine included the addition of a second carburetor supplying a rich air-fuel mixture to the prechamber, and a cam-actuated prechamber intake valve. The prechamber assembly was then installed in the spark plug well of the rebuilt cylinder head. Two different prechamber sizes, 19 percent (Generation I) and 8 percent (Generation II) were fabricated and tested on the Pinto engine. The diameter of the communicating passage was varied between 0.2 and 0.5 in. Also, the effect of higher main chamber intake swirl and the addition of an insulated lean thermal exhaust reactor on engine emission and specific fuel consumption were evaluated.

4.2.2.2 Emission and Fuel Consumption Characteristics

Engine dynamometer test data from the modified Pinto engine, incorporating different Eaton prechamber configurations are presented in Table 4-21, for two speed/brake mean effective pressure settings (Ref. 4-21). For comparison, test data from the nonmodified engine are also shown in Table 4-21.

Table 4-21. EMISSIONS AND INDICATED SPECIFIC FUEL CONSUMPTION OF FIVE EATON CORPORATION PRECHAMBER ENGINE INSTALLATIONS (Ref. 4-21)

	Standard engine	Prechamber engine configuration				
		1	2	3	4	5
Engine generation	---	I	II	II	II	II
Prechamber volume, percent	---	19	8	8	8	8
Nozzle diameter, in.	---	0.5	0.2	0.3	0.3	0.3
Swirl	no	no	no	no	yes	yes
Thermal reactor	no	no	no	no	no	yes
2500 rpm; imep[a] = 78 psi						
Air-fuel ratio	17.4	19.0	20.8	22.0	22.0	23.0
Indicated specific fuel consumption, lb/ihp-hr	0.405	0.415	0.325	0.323	0.356	0.375
HC, gr/ihp-hr	0.76	0.95	2.69	2.29	1.87	0.29
CO, gr/ihp-hr	3.87	5.20	1.15	3.84	3.65	0.95
NO_x, gr/ihp-hr	6.86	1.40	1.76	0.45	0.63	0.61
2300 rpm; imep[a] = 44 psi						
Air-fuel ratio	16.3	25.0	19.8	19.8	20.0	22.8
Indicated specific fuel consumption, lb/ihp-hr	0.617	0.550	0.422	0.422	0.419	0.450
HC, gr/ihp-hr	0.38	41.50	0.84	2.18	0.72	0.41
CO, gr/ihp-hr	2.31	9.23	8.1	5.77	9.75	0.38
NO_x, gr/ihp-hr	1.54	0.49	0.79	0.73	0.10	0.20

[a]Indicated mean effective pressure

Compared to the standard engine, configuration 1 showed 70 to 80 percent reduction in NO_x, accompanied by a substantial increase in HC and CO. At the high-load point, the specific fuel consumption of prechamber engine configuration 1 was about 2 percent higher, whereas a 10 percent reduction was observed at the low-load setting. Although further improvements in NO_x were obtained with configuration 4, utilizing main chamber swirl, HC and CO remained above the levels of the standard engine while the specific fuel consumption was improved substantially at both load settings. As expected, incorporation of the lean thermal reactor (configuration 5) resulted in a reduction of the HC and CO emissions below the levels of the standard engine.

4.2.2.3 Current and Projected Status

Based on these tests, Eaton Corporation has concluded that the test objectives were met with its prechamber engine. By operating extremely lean (A/F = 23) NO_x was reduced by about 90 percent relative to the standard engine. The addition of a lean thermal reactor was advantageous in maintaining low HC and CO emissions (Ref. 4-21).

No information is available regarding future prechamber engine efforts by the Eaton Corporation.

4.2.3 Phillips Petroleum Company

Phillips Petroleum has conducted laboratory tests on a single-cylinder prechamber engine to determine the effect of various gasoline compositions on the performance of this engine type. Three different fuels were used in the program; fuel A was a regular grade, full boiling range gasoline; fuel B was a premium, full boiling range gasoline; and fuel C was a regular grade narrow cut gasoline (Ref. 4-22).

4.2.3.1 Engine Description

The engine configuration tested by Phillips consisted of a small prechamber having a volume of about 2 percent of the total clearance volume. The unit was inserted into the spark plug opening of a CFR laboratory single-cylinder engine. The prechamber was equipped with a spark plug and an intake flapper valve. Separate fuel supply systems were utilized for the prechamber and main chamber. The fuel for each of the combustion chambers was pre-mixed with air in two separate air mixing tanks. The air-fuel ratio of the prechamber mixture was about 3, while the mixture ratio in the main chamber was lean and was varied between about 15 and 25 during the test program. The prechamber inlet valve was actuated during the induction stroke by the pressure differential between the cylinder and the mixing tank. In all cases, the spark timing was adjusted for maximum torque.

4.2.3.2 Emission Characteristics

Emissions from the Phillips prechamber were measured in the laboratory using a flame ionization analyzer for HC, an infrared analyzer for CO, and electrochemical sensors for NO_x. Aldehydes were determined manually using the MBTH method.

Typical emission data from the prechamber engine and the standard engine are presented in Figure 4-18 showing HC, CO, and NO_x emissions expressed in grams per indicated horsepower-hour as a function of indicated brake mean effective pressure (Ref. 4-22). The data plotted are based on the knock-limited compression ratio and maximum power spark timing. Below an imep of 70 psi, CO from the prechamber engine was much lower than from the standard engine, reflecting the leaner mixture used in the prechamber. Conversely, above 70 psi, CO was higher than that of the standard engine. In the high-load regime, the HC emission of the prechamber engine was about 2 gr/ihp-hr which was much higher than that of the standard engine. This would indicate the need for some form of additional emission control. As expected, the NO_x emission of the prechamber engine was substantially lower than that of the standard engine over the whole range of operating conditions, particularly at loads below 70 psi. While the maximum aldehyde emission of 0.16 gr/ihp-hr obtained with the prechamber engine is relatively low, it is higher than what is normally encountered in conventional engines.

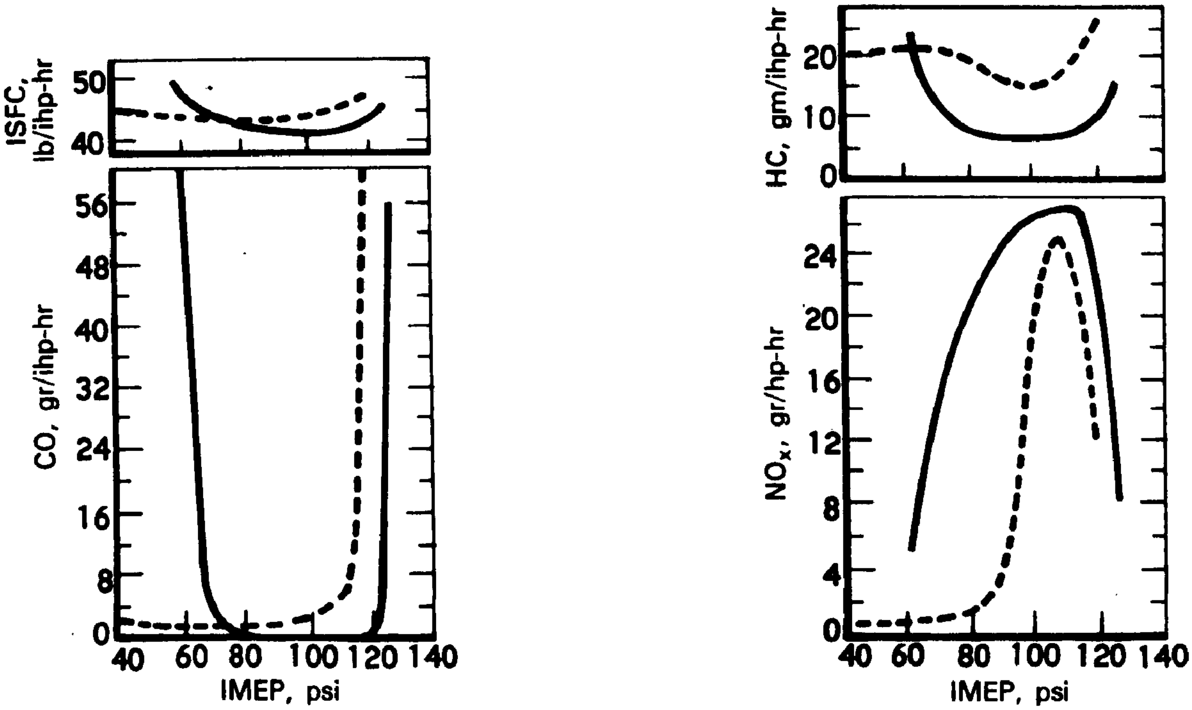
4.2.3.3 Fuel Consumption Characteristics

Relative to the standard engine, the prechamber engine shows higher indicated specific fuel consumption for indicated mean effective pressures above 70 psi and lower values in the low-load regime, as illustrated in Figure 4-18.

4.2.3.4 Engine Performance Characteristics

The principal feature of prechamber engines is their ability to operate with lean mixtures. This is illustrated in Figure 4-19, indicating an extension of the operational limit of the prechamber engine from the limiting fuel-air equivalence ratio of 0.8 of the standard engines to about 0.55. While favorable emission and fuel consumption characteristics were realized with the prechamber, the engine suffered a 5.5 percent loss in maximum power relative to the standard engine.

At high power levels, the prechamber engine was noticeably noisier than the standard engine. This was found to be due to the high pressure rise rate developed during the combustion process in the prechamber engine. This rate increased rapidly as the fuel-air equivalence ratio was increased above unity and would indicate a practical equivalence ratio limit of about 0.95 (Ref. 4-22).



Note: ——— Standard Engine (CR = 6.0)

- - - Prechamber Engine (CR = 6.46)

1000 rpm
Maximum Power Spark Timing

Figure 4-18. Phillips single-cylinder prechamber engine emissions and indicated specific fuel consumption versus indicated mean effective pressure (IMEP) (Ref. 4-22)

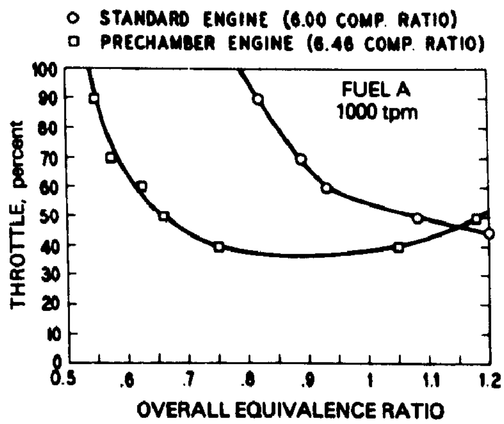


Figure 4-19. Operational limits of standard and prechamber engines (Ref. 4-22)

The prechamber engine had a lower octane requirement than the standard engine. For instance, for a research octane number (RON) of 91, the maximum knock-free compression ratio of the single-cylinder engine was 6.0:1 compared to 6.46:1 for the prechamber engine. This improvement is attributed to the rapid combustion occurring in the prechamber engine which minimizes the effect of the end gas.

4.2.3.5 Current and Projected Status

The principal problem area of the Phillips prechamber engine is related to its relatively high emissions. While some improvement would be expected from further development work, Phillips has no current plans for additional work on the engine.

4.2.4 Teledyne Continental Motors

Development of light-duty and heavy-duty automotive engines has been underway at Teledyne Continental Motors for many years, primarily under sponsorship of the U.S. Army Tank Automotive Command (USATACOM). The principal objective of these programs was the achievement of good fuel economy combined with a multi-fuel capability. As an outgrowth of these efforts, a number of lean burning spark-ignition engine concepts were studied by Teledyne Continental Motors in 1961 including both open chamber and prechamber concepts (Ref. 4-23). In 1966, a hardware program was initiated which was aimed at the development of a nonaspirated torch ignition engine concept for potential application to the L-141 military (Jeep) engine. This was followed by another hardware program involving the incorporation of the Walker Stratofire prechamber concept which is discussed in Section 4.2.6. While the Stratofire concept exhibited more desirable performance characteristics than the nonaspirated design, the program was terminated in 1968 primarily because of severe engine durability problems which were caused by overheating of the prechamber unit.

With the renewed interest in prechamber engines emerging in the 1972/73 time period, the Teledyne effort was revived in 1973 with particular attention focused on the solution of the heating problems encountered in the previous programs.

4.2.4.1 Engine Description

The prechamber engine work conducted by Teledyne Continental Motors in 1966 and 1974 under USATACOM sponsorship evolved around the Walker Stratofire concept. This device was developed by Walker Manufacturing Company (discussed in Section 4.2.6) for potential application as a retrofit unit for automobile engines. The device consists of a small prechamber, a spark plug, and a small passage connecting the prechamber and the main chamber. The first Stratofire configuration tested by Continental Motors employed a spherical flapper valve which was actuated by the pressure differential generated during the induction stroke of the piston. In an effort to eliminate the frequent valve failures and inconsistent valve action encountered with this early configuration, Continental Motors replaced the flapper valves with conical valves. The new design which was manufactured from high temperature N-155 alloy is shown in Figure 4-20 (Ref. 4-24). Although improved durability was achieved with this configuration, valve breakage could not be completely eliminated. Subsequently, additional modifications were incorporated in the prechamber design, including the application of a larger valve diameter to provide better valve guidance. Apparently the problem of valve failure was eliminated by this particular design change.

The fuel flow into the main chamber of this engine was controlled by means of a modified Zenith Model 12848 carburetor

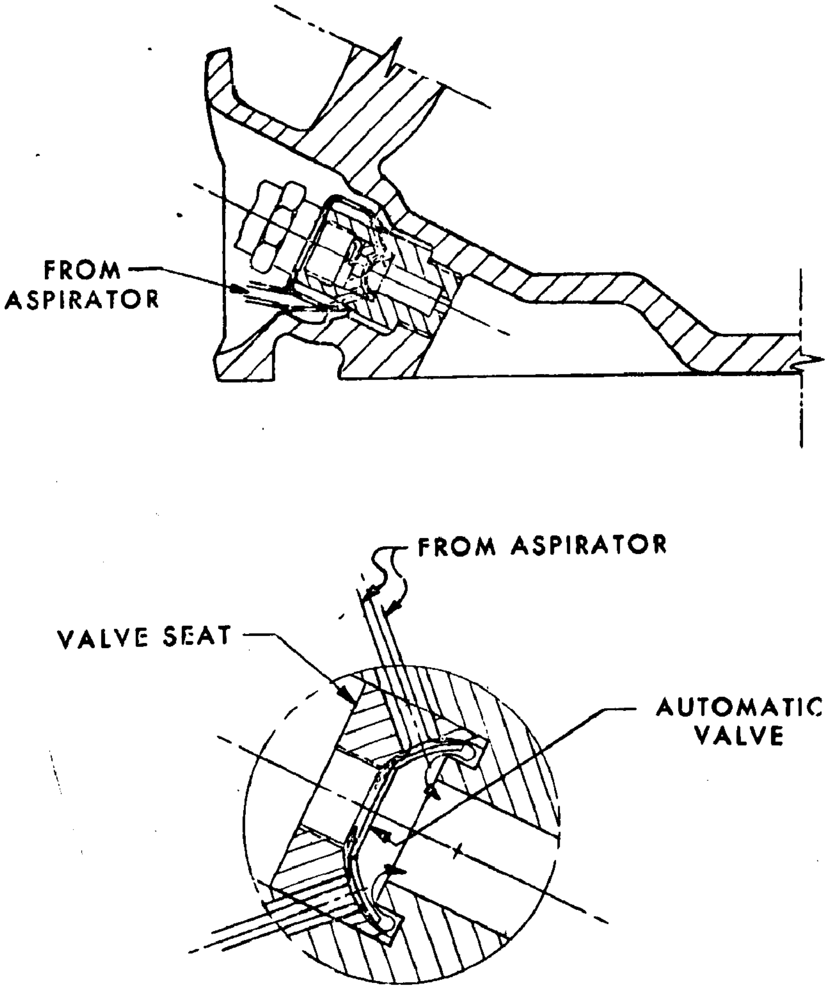


Figure 4-20. Cross section of Continental/Walker cone valve prechamber (Ref. 4-24)

whose power jet was blocked off and the main metering jet was controlled with an adjustable needle valve. The prechambers were initially operated with a simple aspirator tube fuel system which was adjusted manually at the various engine operating points to provide the desired air-fuel ratios and float levels. Optimum engine performance was obtained with prechamber air-fuel ratios between 7.5 and 8.0 and main chamber air-fuel ratios between 22 and 25.

4.2.4.2 Engine Performance Characteristics

Initially, the modified L-141 engine incorporating Walker/Continental prechambers was tested on gasoline without changing the engine head. As a result, the compression ratio of the engine was reduced from 7.5 to 7.29. According to Figure 4-21 (Ref. 4-24), the specific fuel consumption of the modified engine operated at low speed was considerably lower than that of the conventional engine, particularly in the low-load regime where the improvement amounted to as much as 28 percent. However, at higher speeds, the prechamber valve action was inconsistent, causing unsatisfactory engine operation. This problem was alleviated by increasing the compression ratio to 9.59. At this point, no serious engine knock was encountered and the brake specific fuel consumption (bsfc) of the engine was reduced between 15 and 28 percent at medium-load levels and speeds up to 2500 rpm. However the bsfc improvement was generally lower at higher engine speeds and loads. For example, at 3500 rpm and a brake mean effective pressure (bmep) of 70 psi, the observed bsfc improvement was only about 5 percent.

Additional performance tests were conducted on the L-141 prechamber engine configuration using CITE and JP-4 fuels and a number of different compression ratios ranging between 6.2 and 11.15. While a reduction of the compression ratio resulted in an increase of the knock limited bmep operating point of the engine, startup became more difficult with decreasing compression ratio. Typical bsfc versus bmep curves obtained with the various fuels are shown in Figure 4-22 (Ref. 4-24), indicating a small increase in bsfc for CITE fuel relative to gasoline.

Emission test data obtained by Continental Motors in 1974 are plotted in Figure 4-23, showing HC and NO_x specific mass emissions as a function of manifold vacuum and engine torque

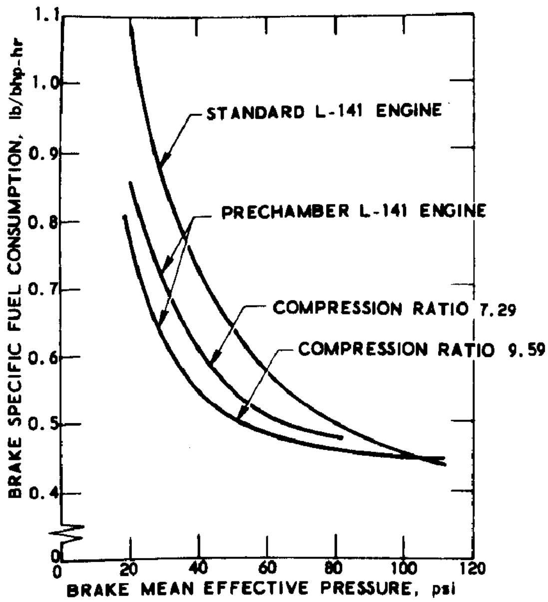


Figure 4-21. Part-load brake specific fuel consumption; Walker/Continental cone valve design; 1500 rpm; gasoline (Ref. 4-24)

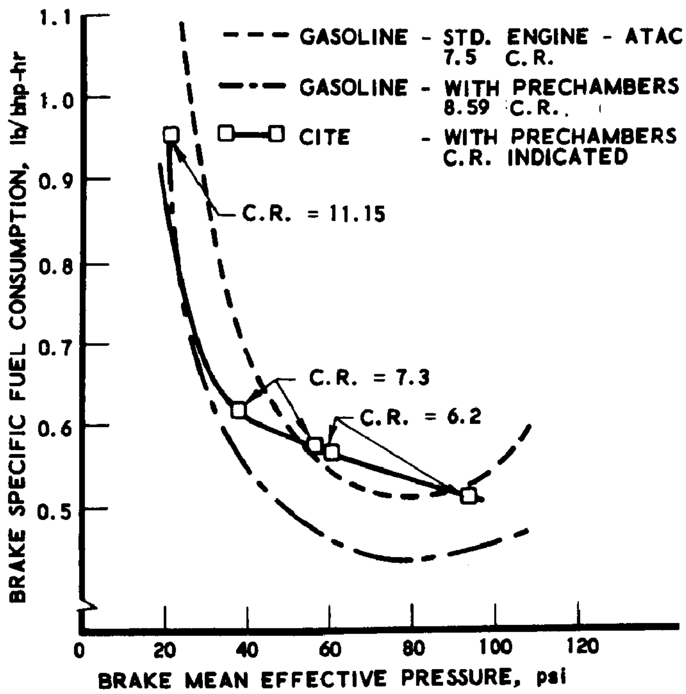


Figure 4-22. Brake specific fuel consumption versus brake mean effective pressure; standard and Walker/Continental L-141 engines; 1600 rpm (Ref. 4-24)

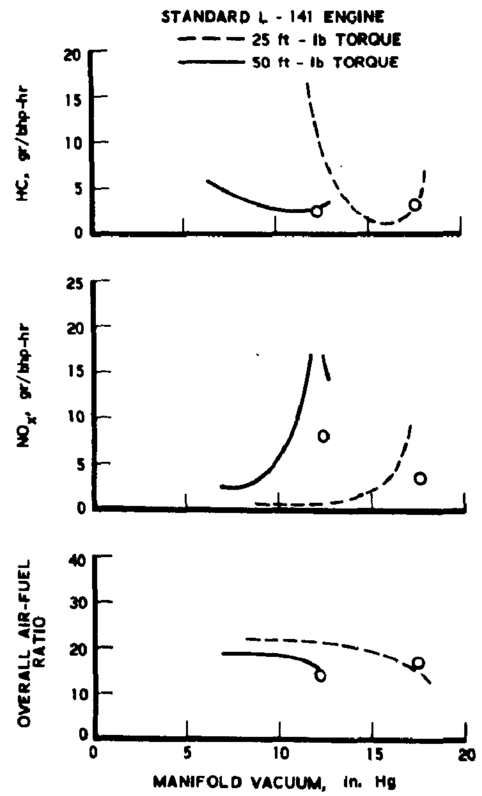


Figure 4-23. Emissions and air-fuel ratio versus manifold vacuum; Walker/Continental L-141 prechamber engine; 3 percent prechamber volume; 2000 rpm; 7.5:1 compression ratio (Ref. 4-25)

(Ref. 4-25). In this case, the engine was operated at 2000 rpm utilizing a prechamber volume of 3 percent of the total clearance volume. Also shown in the figure are discrete data points representing the nonmodified L-141 engine. As expected, NO_x decreases rapidly in the lean regime while HC assumes a minimum at air-fuel ratios slightly above stoichiometric. In these tests, the engine was operated successfully at air-fuel ratios up to 22. As shown, very low NO_x emissions were achieved, accompanied by rather high HC levels. The CO emissions, which are not shown in the figure, were very low throughout the lean operating regime evaluated in the program. Similar results were obtained at 3000 rpm and low-to-medium-load levels. However, in the high-load regime the NO_x emission of the prechamber engine was higher than that of the baseline configuration. In this test, considerable overheating of the prechamber was encountered which might have caused preignition of the air-fuel mixture accompanied by higher NO_x formation rates.

4.2.4.3 Potential Problem Areas

A number of potential problem areas were encountered by Teledyne Continental Motors during the course of its prechamber development programs. These include overheating of the prechamber assembly during operation at high loads, frequent failure of the flapper-type prechamber valve, and deposit buildup in the prechamber valve area when the engine was operated on combat gasoline (MIL-G-46015), which had a sulfur content of 0.10 to 0.15 percent. In this case, lead sulfate was formed during the combustion process and was deposited throughout the engine, causing malfunction of the prechamber valve. No deposit problems were encountered with CITE fuel, which is lead-free.

4.2.4.4 Current and Future Status

While some operational difficulties remain to be resolved, the Continental/Walker cone valve prechamber concept shows some potential for improving the specific fuel consumption of automotive spark-ignition engines, particularly in the part-load regime. However, Continental Motors feels that the concept might not be readily applicable as a retrofit device because of many engine adjustments that would be required to achieve acceptable engine/vehicle driveability. Further improvements in fuel economy and durability might be realized by means of certain design modifications, including optimization of the prechamber geometry and incorporation of electronic manifold fuel injection.

The results of the most recent prechamber effort conducted by Continental Motors under contract to the USATACOM is scheduled for publication in the near future (Refs. 4-26 and 4-27).

4.2.5 Thermo Electron Corporation (Clawson Concept)

The development of the Clawson prechamber concept (Ref. 4-28) was started at the Dynatech Corporation, Cambridge, Massachusetts, and is now being pursued under Clawson's direction by the Thermo Electron Corporation, Waltham, Massachusetts.

4.2.5.1 Engine Description

The Clawson prechamber concept is illustrated in Figure 4-24 (Ref. 4-28). The prechamber is spherical in shape, occupying about 15 percent of the total clearance volume and is connected to the main chamber of the engine through a flow orifice. The geometry of the prechamber and main chamber is an important design parameter, and departure from the optimum configuration results in some loss in engine efficiency (Ref. 4-28). Combustion air is supplied to the prechamber through the connecting orifice during the compression stroke of the piston, and the turbulence created by the inrushing air flow is utilized to provide a uniform air-fuel mixture. Since the prechamber is unscavenged, some of the combustion products are retained in the prechamber during the exhaust stroke which has some effect on the operation of the engine. The prechamber operates with rich air-fuel mixtures, while the main chamber operates lean. Separate fuel supply circuits are utilized for the prechamber and main chamber, and both fuel injection and carburetion systems have been evaluated to date.

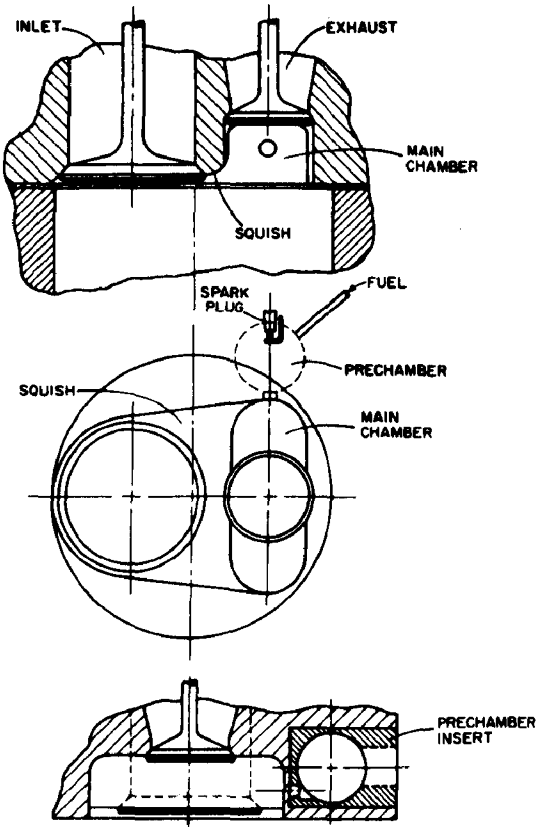


Figure 4-24. Clawson prechamber design (Ref. 4-28)

The Clawson prechamber concept has been tested in a single-cylinder CFR engine using gasoline as the fuel. Other development work was performed on a two-stroke SAAB vehicle using JP-4 fuel, and a four-stroke L-141 military light-duty engine modified by Ford for PROCO operation and installed in a Ford Capri automobile. Currently, Thermo Electron is considering the application of the Clawson concept to its line of Palmer Crusader marine engines.

Pertinent data on the modified SAAB and Ford Capri vehicles are listed in Table 4-22 (Ref. 4-28). The compression ratio of the SAAB engine was increased from the standard value of 8.5 to 11.0, and fuel injection was utilized for both the prechamber and main chamber. The main chamber throttle valve remained open at all engine operating conditions and the load was controlled by varying the quantity of fuel injected. An Engelhard catalyst was installed in the exhaust system for HC and CO control.

Table 4-22. CLAWSON PRECHAMBER ENGINE/VEHICLE CONFIGURATIONS (Ref. 4-28)

Vehicle	SAAB	Ford Capri
Model year	1967	1971
Transmission	4-speed manual	Automatic
Compression ratio	11:1	11:1
Main chamber fueling	Bosch fuel injection	Carburetion
Prechamber fueling	Bosch fuel injection	Gear pump/dribble orifice
Ignition timing	4 to 5-deg BTDC	4-deg BTDC
Catalyst	Engelhard	None
EGR	No	20 percent at idle; 5 percent at 70 psi bmep
Fuel	JP-4	100-octane lead-free gasoline

A number of additional modifications were incorporated in the L-141/PROCO engine used in the Capri vehicle, including larger intake and exhaust valves. In the engine, the fuel to the main chamber was provided by the standard carburetor while the prechamber was fueled by a separate low pressure gear pump and check valve/dribble orifice arrangement. Exhaust gas recirculation was employed at rates varying between 20 percent at idle and 5 percent near full load.

4.2.5.2 Emission Characteristics

The SAAB vehicle was tested by the EPA in accordance with the 1972 and 1975 Federal Test Procedures using a 2250-lb inertia weight setting. In addition, the engine was tested at a number of steady-state conditions (Ref. 4-29). As indicated in Table 4-23, the NO_x emissions, as determined over the Federal Driving Cycle, were below the 1978 federal emission standards of 0.4 gr/mile, while CO was slightly higher than the 1977 standard of 3.4 gr/mile. However, HC was considerably above the 1977 standard of 0.4 gr/mile. The high level of HC is attributed to the presence of heavy hydrocarbons in the JP-4 fuel and the contamination of oil from the crankcase scavenged two-stroke engine. It is anticipated that the use of gasoline instead of JP-4, combined with fan scavenging of the two-stroke engine, would make the converter more effective (Ref. 4-28).

Table 4-23. SAAB/CLAWSON PRECHAMBER VEHICLE EMISSIONS; 2250-lb INERTIA WEIGHT (Ref. 4-29)

Federal Test Procedure	Emissions, gr/mile			
	HC[a]	CO	NO_x	CO_2
1972 (hot start)	5.6	3.3	0.3	429
1975	6.4	3.6	0.3	430

[a]About 30 percent too low because of condensation of the heavier HC species in the CVS bag.

Data from constant vehicle speed tests conducted on the SAAB vehicle with Clawson prechambers are listed in Table 4-24 (Ref. 4-29). During these tests, a flame ionization detector with heated sample lines was used to prevent condensation of the heavy hydrocarbon species. Test data from two conventional vehicles are included in this table for comparison. As indicated, the NO_x and CO emissions from the prechamber engine are substantially lower than for the conventional vehicles. Conversely, HC is higher for the prechamber engine.

Table 4-24. STEADY-STATE EMISSIONS FROM SAAB/CLAWSON PRECHAMBER VEHICLE AND CONVENTIONAL VEHICLES; HOT-START BAG PROCEDURE (Ref. 4-29)

	SAAB/Clawson prechamber				1971 Ford (351 CID)			1970 Datsun (97 CID)		
	Emissions, gr/mile				Emissions, gr/mile			Emissions, gr/mile		
	HC	CO	NO_x	HC (hot)[a]	HC	CO	NO_x	HC	CO	NO_x
20 mph	6.9	1.1	0.3	9.6	1.2	2.9	2.4	1.9	20.3	2.5
30 mph	2.8	0.8	0.1	4.4	1.2	2.3	3.1	1.7	11.2	3.3
40 mph	2.3	1.0	0.1	4.1	1.2	2.5	7.1	1.6	6.0	4.2
50 mph	2.0	1.0	0.2	2.7	1.2	4.2	9.0	1.8	2.3	7.6

[a]Measured with heated flame ionization detector setup.

Steady-state emission maps from the modified L-141/Clawson prechamber are presented in Figures 4-25 through 4-27 showing HC, CO, and NO_x specific mass emissions, expressed in terms of gr/bhp-hr, as a function of brake mean effective pressure and engine speed (Ref. 4-28). While no factors are available to convert these emissions into gr/mile units, it appears that the emission levels would be quite low.

Recent unpublished test data provided by Thermo Electron for the Capri vehicle incorporating the L-141/Clawson

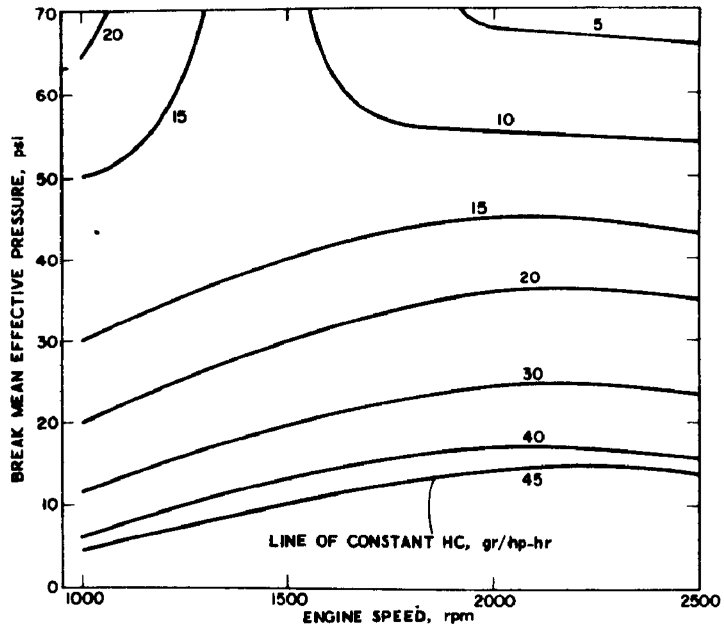


Figure 4-25. HC specific mass emission of L-141/Clawson prechamber engine; carbureted version (Ref. 4-28)

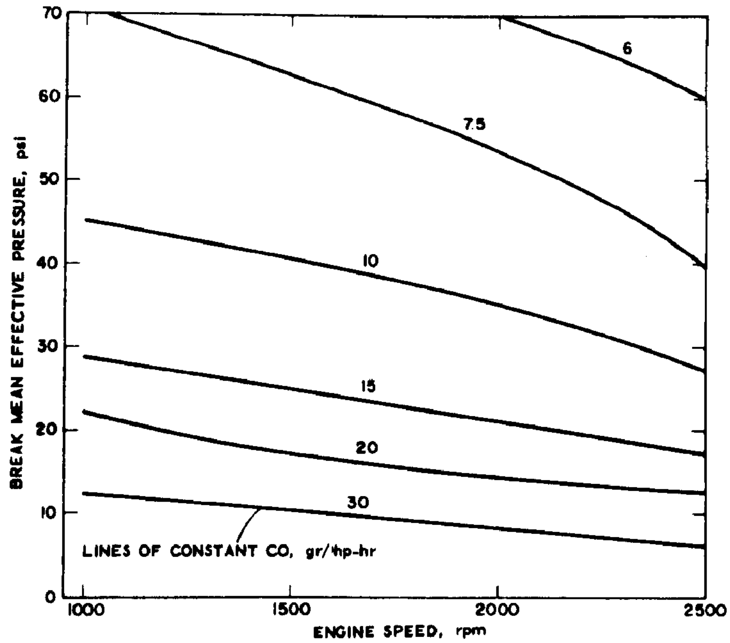


Figure 4-26. CO specific mass emission of L-141/Clawson prechamber engine; carbureted version (Ref. 4-28)

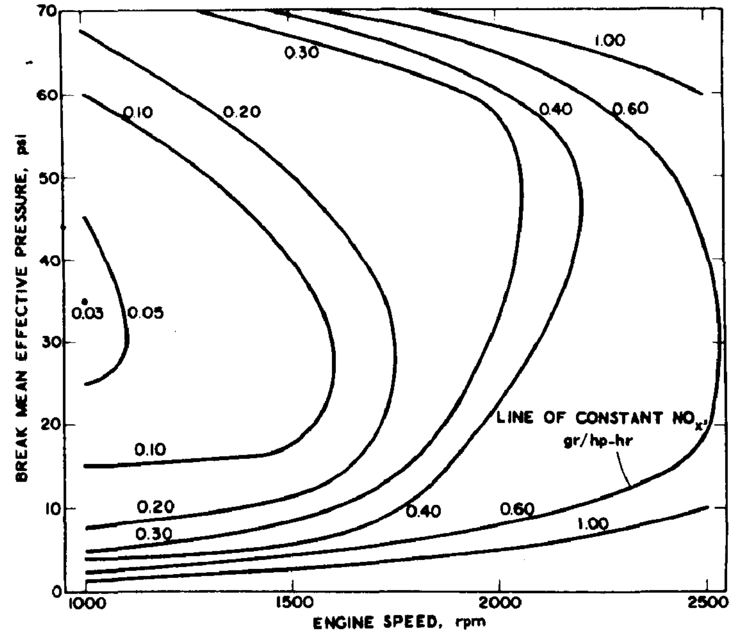


Figure 4-27. NO_x specific mass emission of L-141/Clawson prechamber engine; carbureted version (Ref. 4-28)

prechamber engine, plus an exhaust catalyst for HC and CO control, indicate 0.7 gr/mile, NO_x, 1 gr/mile CO, and 1 gr/mile HC when tested over the 1975 Federal Driving Cycle. While CO is below the 1978 standard, HC and NO_x are considerably above the standards for 1978.

4.2.5.3 Fuel Consumption Characteristics

The specific fuel consumption of the L-141/Clawson prechamber engine is presented in Figure 4-28 as a function of brake mean effective pressure and engine speed (Ref. 4-28). Selected data points from this figure are compared in Table 4-25 with the corresponding specific fuel consumption data reported by Teledyne Continental Motors (Ref. 4-22) for a standard nonmodified L-141 engine.

As indicated in Table 4-25, the specific fuel consumption of the prechamber engine is about 10 to 25 percent lower than that

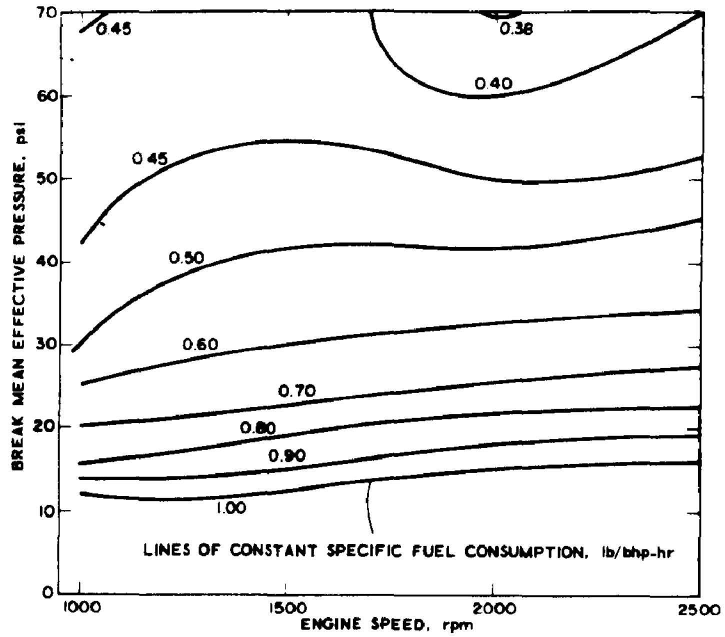


Figure 4-28. Brake specific fuel consumption of the L-141/ Clawson prechamber engine (Ref. 4-28)

Table 4-25. SPECIFIC FUEL CONSUMPTION COMPARISON (Ref. 4-28)

Speed, rpm	bmep, psi	Specific fuel consumption, lb/bhp-hr	
		Standard L-141 engine[a]	L-141/prechamber engine[b]
1500	20	1.05	0.75
	70	0.53	0.45
2000	20	1.05	0.80
	70	0.52	0.38
2500	20	1.08	0.88
	70	0.47	0.40

[a]Compression ratio 9.59:1

[b]Compression ratio 11.0:1

of the conventional engine. However, some of this improvement is attributed to the higher compression ratio of the prechamber engine.

More recent data provided by Thermo Electron from its prechamber engine-equipped Capri vehicle indicate a fuel economy of 22 mpg over the Federal Driving Cycle, which represents an improvement of about 18 percent relative to the EPA certification data for the standard Capri (2000-cc engine and automatic transmission). It should be noted, however, that the power output of the standard Capri engine is somewhat higher than that of the modified L-141 engine.

According to Thermo Electron, the specific fuel consumption of the CFR engine with Clawson prechamber is about 10 to 20 percent lower than for the standard CFR engine (Ref. 4-28).

4.2.5.4 Materials and Manufacturing

Incorporation of Clawson prechambers into automotive engines would require a number of engine modifications including a new cylinder head and a separate fuel supply system for the prechamber. While no new materials and manufacturing techniques would be required, it appears that the Clawson concept would not be economically feasible as a retrofit device for in-use vehicles.

4.2.5.5 Vehicle Performance Characteristics

Limited data are available on the road characteristics of cars equipped with Clawson prechambers. Based on the SAAB vehicle tests, it is concluded that a loss in maximum power was encountered as a result of lean mixture operation. While the magnitude of the power loss has not been quantized, the rate of acceleration of the vehicle at full-throttle was definitely lower than for the standard configuration. It appears that some improvement in the emissions and fuel economy might be obtained with an oversized prechamber engine operating in the 2000 to 2500-rpm range.

The Clawson prechamber engine tends to be noisier than the standard engine, particularly at idle and low loads. This is attributed to the more rapid pressure rise rate obtained with prechamber ignition.

4.2.5.6 Potential Problem Areas

The main problem of the Clawson prechamber concept is related to the high HC emission as evidenced in the SAAB vehicle and emphasized even more in the converted L-141 engine. The high HC level in the L-141 engine was caused mainly by poor fuel atomization in the dribble-type prechamber injector (Ref. 4-28). It appears that a more sophisticated prechamber fuel injection system would be required to reduce HC and CO. The solution to this problem requires a compromise between (1) adequate fuel atomization in the prechamber, (2) adequate air-fuel mixing in the main chamber, and (3) acceptable cost of the fuel delivery system.

Because of the inherent loss in power output capability of this prechamber engine relative to conventional engines, the displacement of the prechamber engine might have to be increased, particularly in the case of small cars where a power loss might not be tolerable.

4.2.5.7 Current and Projected Status

Present prechamber work at Thermo Electron is mainly focused on the converted L-141 engine. Obviously, improvements in the fuel system and air-fuel mixing mechanism are necessary to meet the 1978 federal emission standards. To accomplish these goals, new fuel and air system components will be evaluated including auxiliary fuel pumps, injectors, and air bleed arrangements.

Based on the work conducted on the prechamber equipped SAAB and L-141 engines, it is concluded that these engines are more tolerant to low-octane fuels than conventional engines. Future efforts are scheduled to include an evaluation of the effects of methanol and gasoline/methanol mixtures on prechamber engine performance. Also, Thermo Electron intends to utilize the Clawson concept in its Palmer Crusader engines for the purpose of acquiring additional operational experience and cost data.

While introduction of the prechamber into new engines would carry only a small cost penalty, modification of existing engines would be very costly and is not considered to be economically feasible. Application of the Clawson prechamber concept by the automotive industry would depend on several factors including (1) demonstrated ability to meet the 1978 emission standards at high mileage, (2) accumulation of satisfactory operational experience on a number of engines, and (3) licensing agreements with at least one car manufacturer.

4.2.6 Walker Manufacturing Company

During the 1960/61 time period, Walker Manufacturing Company, Racine, Wisconsin, conducted an inhouse program which was aimed at the development of an efficient low pollution concept that would be applicable as a retrofit unit for in-use automobile engines. In the course of this program, a number of prechamber engine configurations were built and tested on the engine dynamometer and vehicle chassis dynamometer. Although some improvement in specific fuel consumption was accomplished, the program was eventually cancelled by Walker, primarily because none of the designs evaluated was capable of meeting the emission goals established by Walker management.

The development of the Walker prechamber engine concept was revived by Teledyne Continental Motors in 1966 and again in 1974 as part of the overall engine improvement efforts pursued by the U.S. Army Tank Automotive Command. The status of the Teledyne Continental Motors program is discussed in Section 4.2.4.

4.2.6.1 Engine Description

The principal design objective of the Walker Stratofire prechamber engine concept (Refs. 4-30 and 4-31) was the achievement of lower emissions and improved fuel economy relative to equivalent conventional automotive spark-ignition engines. Since the concept was intended to be utilized in retrofit applications, low system cost, ease of installation, and minimum loss in engine power output capability were major design considerations. Therefore, fuel injection and cam-actuated prechamber intake valves were excluded from the list of potential system components. Furthermore, the size of the prechamber was limited to less than 10 percent of the total clearance volume to minimize the reduction in compression ratio caused by the addition of the prechamber.

The original Stratofire unit (configuration No. 1) tested by Walker is illustrated in Figure 4-29 (Ref. 4-32). This device, which is inserted into the spark plug holes of the engine, consists of a small upper tube containing the conventional spark plug and intake valve, a larger intermediate chamber and a cylindrical passage connecting the prechamber and main chamber. The prechamber intake valve is biased into seating by a cantilevered spring and is actuated by the suction pressure created in the engine cylinder during the induction stroke. Approximately 5 percent of the fuel is supplied to the prechamber by means of an auxiliary carburetor. This carburetor was adjusted to handle the total engine fuel at idle and provides a near-stoichiometric mixture at all other load conditions. A lean mixture was inducted into the main chamber through the stock carburetor which was fitted with a smaller main jet. Apparently good mixture ratio control was achieved over the whole range of operating conditions tested by Walker (Ref. 4-32).

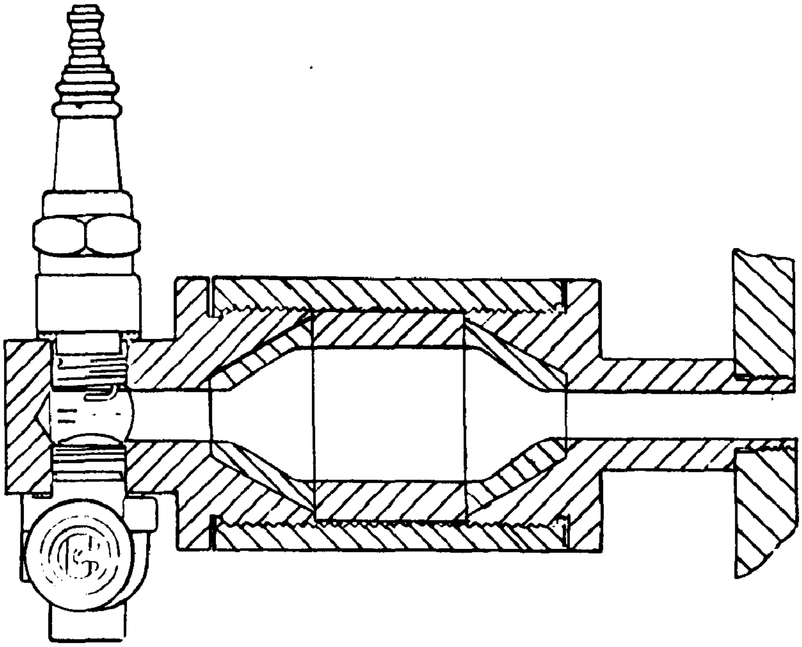


Figure 4-29. Walker Stratofire configuration No. 1 (Ref. 4-32)

During the compression stroke, a small fraction of the lean mixture inducted into the main chamber enters the prechamber through the communicating passage. As a result, the prechamber charge is stratified, forming a near-stoichiometric mixture zone in the spark plug region and a lean zone in the vicinity of the communicating passage. Upon ignition, a high velocity jet of hot gases is discharged from the prechamber into the main chamber to provide multiple ignition sources for the lean main chamber mixture. At idle, the overall engine air-fuel ratio was adjusted to about 20:1. With increasing load, the main chamber throttle was gradually opened and the air-fuel ratio was then reduced to values below stoichiometric for full-load operation.

Stratofire engine configuration No. 2 is shown schematically in Figure 4-30 (Ref. 4-32). In this design, the combustion chamber was cylindrical (0.375-in. diameter; 2.5-in. length) and smaller in volume than in the case of configuration No. 1. The flapper valve and spark plug were installed at the top-end of the unit, while a conical orifice was used as the communicating passage. An orifice diameter of 0.17-in. appeared to be the optimum. The cooling problem encountered in the first configuration was alleviated by incorporating a set of cooling fins around the unit.

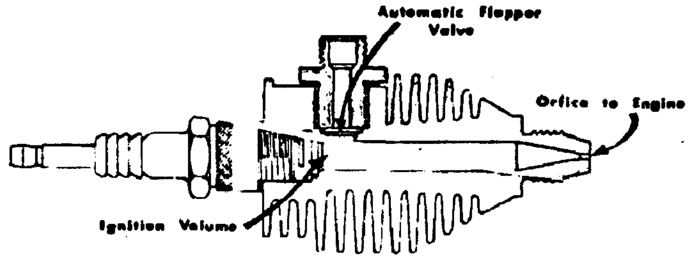


Figure 4-30. Walker Stratofire configuration No. 2 (Ref. 4-32)

A number of other design modifications were considered by Walker Manufacturing Company. In particular, the Mark III configuration was extensively tested in a 1960 Corvair vehicle. This particular configuration, shown in Figure 4-31, employed a very small prechamber, which was regeneratively cooled using the prechamber air-fuel mixture as the heat sink. The unit incorporated a small lip which was utilized to delay the evaporation of fuel droplets in the vicinity of the spark plug until the start of the compression stroke. This approach performed quite well and resulted in a small improvement in specific fuel consumption relative to the other configurations tested.

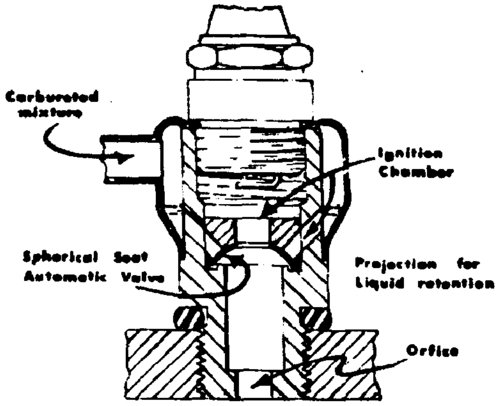


Figure 4-31. Walker Stratofire Mark III configuration (Ref. 4-32)

4.2.6.2 Engine Emission and Performance Characteristics

The Stratofire prechamber configuration No. 1 was tested by Walker in a single-cylinder, air-cooled, spark-ignition engine. Apparently the engine ran quite well and the brake specific fuel consumption (bsfc) at idle and low loads was of the order of 20 percent better than that of the nonmodified engine. At higher engine loads, the fuel consumption was also lower, but difficulties were encountered in providing adequate prechamber cooling (Ref. 4-32).

Steady-state test data from a Corvair engine incorporating prechamber configuration No. 2 are presented in Figure 4-32, showing the brake specific fuel consumption of the modified and standard engines at two power levels (4 bhp and 8 bhp) as a function of manifold vacuum (Ref. 4-32). In these tests, the engine speed and load remained constant while the air-fuel ratio and main chamber throttle position were varied. As indicated, the modified engine showed slightly lower bsfc at the light-load condition and somewhat higher bsfc at higher loads.

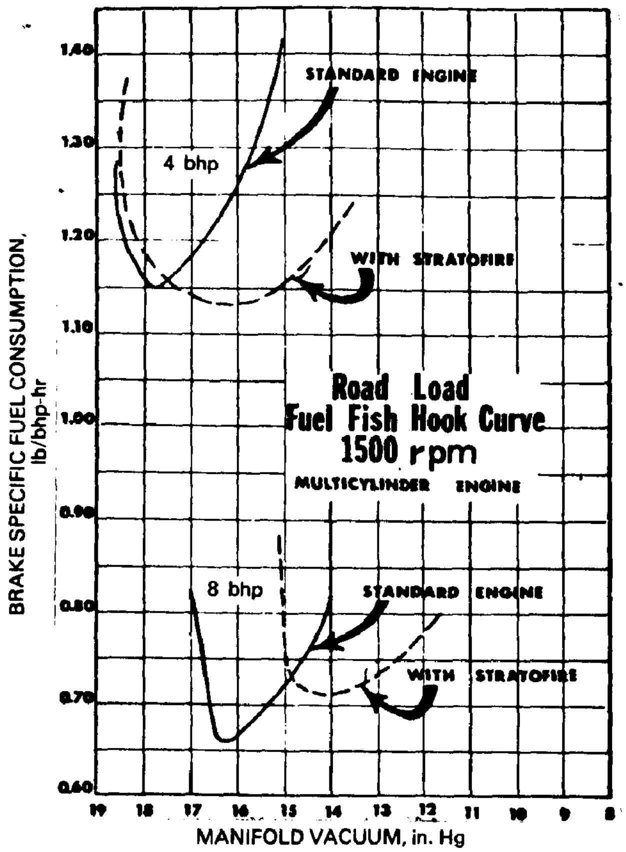


Figure 4-32. Brake specific fuel consumption versus manifold vacuum; Walker Stratofire configuration No. 2, Corvair installation (Ref. 4-32)

A limited number of HC measurements were made with the Corvair engine employing prechamber configuration No. 2. These data indicate a 30 to 40 percent reduction in HC relative to the non-modified Corvair (Ref. 4-32). Installation of the Stratofire units was simple, and the engine exhibited good starting characteristics.

4.2.6.3 Vehicle Emission and Performance Characteristics

Stratofire configurations No. 2 and Mark III were tested in the Corvair vehicle. In all cases, vehicle acceleration was good and operation with disconnected vacuum spark advance resulted in optimum vehicle driveability, except, perhaps, during periods of high acceleration when diesel-like engine knock was experienced. Although a series of different prechamber orifice geometries were utilized in this phase of the program, the inability of achieving significant improvements in specific fuel consumption was a continuing disappointment to Walker (Ref. 4-32).

Subsequently, the Corvair vehicle was equipped with Mark III units and was road tested over about 1000 miles. Apparently, the fuel economy of the vehicle was increased by an unspecified amount over the baseline Corvair. At constant vehicle speed of 65 mph, the modified Corvair incorporating Mark III prechambers had a fuel economy of 29.4 mpg, which represents a 35 percent improvement over the standard car. While the observed increase in gasoline mileage was encouraging, the emissions of the modified Corvair at the conclusion of the test were essentially unchanged from the baseline level. The poor emission performance of the concept was attributed to a loss of engine tune due to mileage accumulation and this contributed to the cancellation of the program.

4.2.6.4 Potential Problem Areas

The principal problem areas encountered with the Walker Stratofire prechamber concept are related to excessive prechamber heating (observed in several configurations) and the inability of the systems to produce sufficient improvement in specific fuel consumption and exhaust emissions. Other potential problem areas to be resolved include the achievement of adequate system durability and reliability, particularly under high-load operation of the engine.

4.2.6.5 Current and Projected Status

Walker has expended a sizable development effort on its Stratofire prechamber engine concept which was originally configured for use as a low emission retrofit device for automobile engines. While some improvement in fuel economy was observed under certain engine/vehicle operating conditions, the disappointing emission performance of the concept was primarily responsible for the termination of the program by Walker in 1965.

Subsequently, Teledyne Continental Motors entered into an agreement with Walker, covering the utilization of the Stratofire concept and modifications thereof in its prechamber engine development work conducted in 1968 and 1974 under contract to the U.S. Army Tank Automotive Command.

4.3 RESEARCH ORGANIZATIONS AND UNIVERSITIES

4.3.1 The Aerospace Corporation

The prechamber work conducted by The Aerospace Corporation consisted of laboratory tests using a single-cylinder engine. The objectives of these efforts were to explore the prospects of emission reduction in spark-ignition engines by means of two-stage combustion (Ref. 4-33).

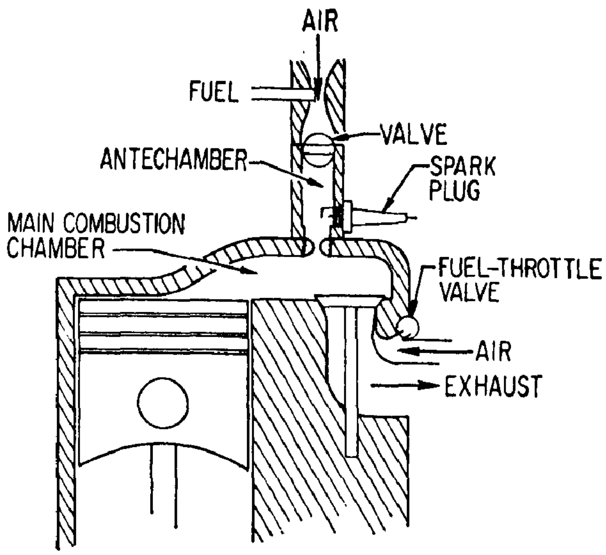


Figure 4-33. Aerospace prechamber engine schematic (Ref. 4-33)

4.3.1.1 Engine Description

A single-cylinder Wisconsin Model No. AGND engine was modified to include a small prechamber as shown in Figure 4-33 (Ref. 4-33). The volume of the prechamber was about 8 percent of the total clearance volume, and the spark plug was fitted into the prechamber wall. The compression ratio of the engine was increased from the original value of 5.3 to 6.7. The main combustion chamber was fuel-fed from a standard carburetor while the prechamber incorporated a separate fuel feed through a mixing venturi and a third valve which was operated by the pressure differential generated during the suction stroke.

The air-fuel ratio of the prechamber was varied between 100 and 40 corresponding to an overall mixture ratio range between 29 and 17, with most measurements taken between 20 and 22.

Isooctane was used in the main chamber while the prechamber was fueled with propane. The base engine was run with isooctane for comparison.

4.3.1.2 Emission Characteristics

In the test program, NO which constitutes 98 percent of NO_x was measured with a Beckman spectrophotometer. The hydrocarbons were continuously monitored in a modified flame ionization detector and were further analyzed by means of a silicon oil column attached to a flame detector for the purpose of separating olefins and alkanes. Carbon monoxide, carbon dioxide, and oxygen were measured in a gas chromatograph (Ref. 4-33).

Selected test results obtained at 1800 rpm are shown in Figure 4-34 (Ref. 4-33). As indicated, the NO_x emission of the prechamber engine is about 50 percent of that of the standard engine. While CO was significantly lowered, the reduction in HC was moderate which was attributed primarily to a leak in the third valve. Similar trends were observed at other engine speeds.

4.3.1.3 Fuel Consumption and Performance Characteristics

While specific fuel consumption data are not included in Ref. 4-33, it appears that the fuel consumption of the prechamber engine was comparable to that of the standard engine.

The Aerospace prechamber engine was capable of operation at much leaner mixtures than the standard engine which

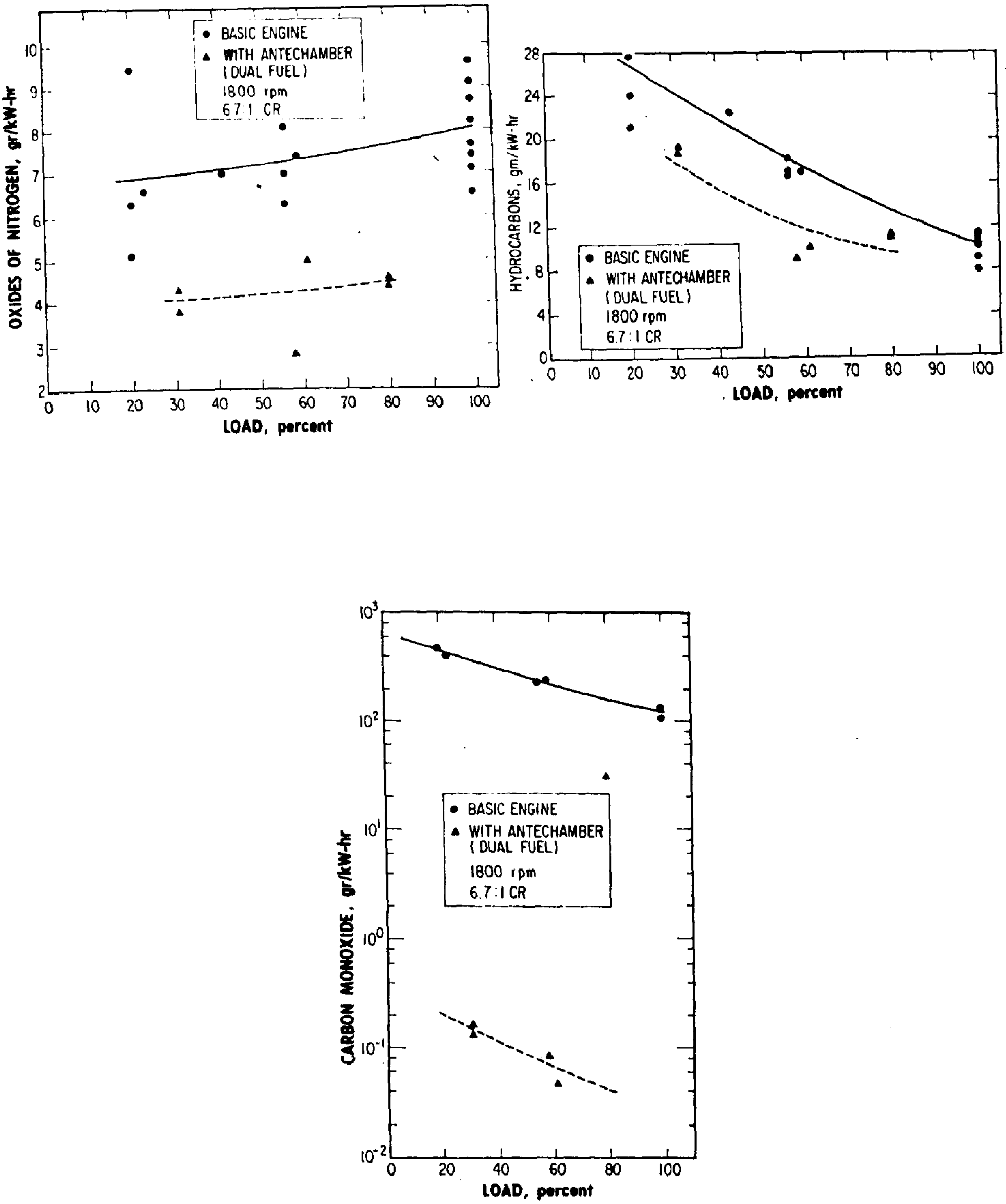


Figure 4-34. Emissions versus engine load; Aerospace prechamber engine (Ref. 4-33)

was limited to air-fuel ratios of about 14 to 15. However, at these lean mixtures the maximum power output capability of the prechamber engine was degraded by about 20 percent relative to the standard engine. During engine cold-start, no choke action was required which is desirable to minimize HC.

Additional testing was performed with isooctane inducted into the prechamber. The engine ran satisfactorily on gasoline and the power loss was reduced to 10 percent.

4.3.1.4 Potential Problem Areas

The principal problem encountered in the test program was related to leakage of the third valve. In addition, valve durability problems were encountered on this particular prechamber configuration.

4.3.1.5 Current and Projected Status

There are no plans for additional work on this concept.

4.3.2 California State University

During the past several years, a research program has been in progress at California State University, Sacramento, to demonstrate the feasibility of the Morghen prechamber concept (Ref. 4-17) as a low emission retrofit device for potential application in new and in-use automotive engines (Ref. 4-34). As part of this effort, California State University has incorporated prechambers into two Ford Falcon vehicles and has participated in the Reduced Emissions Devices (R.E.D.) Rallies conducted in 1972 between Richmond, California and Los Angeles, California (Ref. 4-35), and in 1974 between Davis, California and Los Angeles, California (Ref. 4-36). The principal objective of these rallies was to demonstrate the low emission potential of various engine modifications and concepts. Other important aspects included the achievement of good vehicle/engine performance and improved fuel economy relative to conventional automobiles, with some emphasis on retrofit devices.

4.3.2.1 Engine/Vehicle Description

The particular design utilized in this program is based on the concept patented by Morghen, et al. (Ref. 4-17). As previously shown in Figure 4-17, it consists of a small cylindrical combustion chamber, a spark plug, a pressure-actuated auxiliary valve, and a divergent supersonic nozzle which serves as a communicating passage between the prechamber and the main combustion chamber of the engine. The prechamber units are screwed into the spark plug hole of the engine cylinders. The volume of the prechamber is of the order of 3 percent of the total clearance volume. Under all operating conditions, the prechamber is supplied with a rich air-fuel mixture which is ignited near piston top dead-center to provide a multiple-ignition source for the lean mixture inducted into the main chamber.

The prechamber engine system utilized in the 1974 R.E.D. rally is shown schematically in Figure 4-35 (Ref. 4-36). In this configuration, the standard carburetor of the engine was replaced by a low pressure fuel injection system with digital electronic controls to assure precise fuel metering under all engine operating conditions. A small portion of the fuel was diverted into the electrically heated fuel vaporizer. The vaporized fuel was then mixed with preheated air and admitted to the individual prechambers during the intake stroke of the pistons.

In principle, the prechamber system utilized in the 1972 R.E.D. rally is similar to the 1974 system, except that the standard carburetor was used in place of the fuel injection system. The carburetor was leaned out to provide an air-fuel ratio of 20:1 and the accelerator pump was removed. In addition, the spark advance was disconnected and a constant spark timing of 4 deg before top

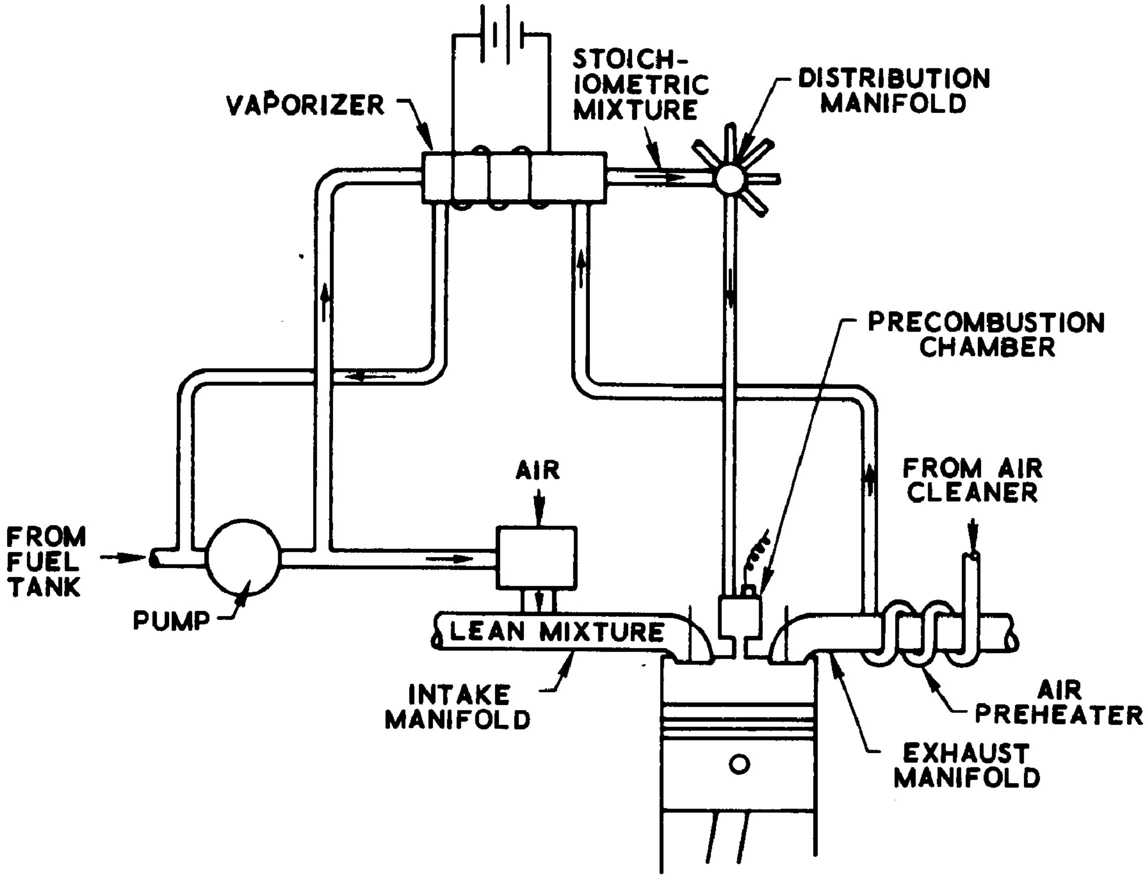


Figure 4-35. Prechamber engine system schematic - California State University (Ref. 4-36)

dead-center was used throughout the operating range of the engine. Pertinent engine and vehicle data are presented in Table 4-26.

Table 4-26. VEHICLE DESCRIPTION - CALIFORNIA STATE UNIVERSITY R.E.D. RALLY ENTRIES (Refs. 4-35 and 4-36)

	1972 Entry	1974 Entry
Make	1962 Ford Falcon	1964 Ford Falcon
Compression ratio	9:1	8.7:1
Rear axle ratio	3.1:1	3.2:1
Main fuel supply	Carburetion	Fuel injection
Displacement, CID	170	
No. of cylinders	6	
Transmission	3-speed manual	
Stock emission control	PCV - system	
Fuel	Gasoline	

4.3.2.2 Vehicle Emissions and Fuel Economy

Vehicle emission and fuel economy data taken during the 1972 and 1974 R.E.D. rallies in accordance with the 1972 Federal Test procedure (CVS) and the seven-mode cycle are listed in Table 4-27 (Refs. 4-35 and 4-36). As indicated, the observed HC emissions were quite high, while CO and NO_x were rather low in the 1972 entry (1962 Falcon). Conversely, very high CO and moderately high NO_x levels were encountered in the 1974 entry (1964 Falcon) indicating air-fuel mixture distribution and/or combustion problems.

The measured fuel economy of the 1964 Falcon with prechambers was 23.4 mpg which was slightly better than that of the nonmodified vehicle.

Table 4-27. EMISSIONS AND FUEL ECONOMY OF THE CALIFORNIA STATE UNIVERSITY PRE-CHAMBER ENGINE EQUIPPED FALCON AUTOMOBILES - R.E.D. RALLY (Refs. 4-35 and 4-36)

Test procedure	1972 Entry[a]				1974 Entry[a]			
	HC	CO	NO_x	Fuel economy, mpg	HC	CO	NO_x	Fuel economy, mpg
1972 CVS	14.2	6.0	1.4	-	11.5	>70	2.6	23.4
Seven mode	10.3	3.2	0.5	-	-	-	-	-
Short cycle	-	-	-	-	5.1	57.3	2.5	-

[a]Emissions in grams per mile

4.3.2.3 Manufacturing and Cost

The prechamber utilized by California State University was manufactured from stainless steel, except for the auxiliary valve which was fabricated from Inconel 718.

The estimated production cost of the Falcon system is $50, exclusive of the fuel injection pump, manifold injectors, and controls (Ref. 4-36). Apparently, 1.5 man-hours were required for the installation of the system in the vehicle.

4.3.2.4 Current and Future Status

Current efforts at California State University are aimed at improving the durability and performance characteristics of the basic Morghen prechamber concept in the Ford Falcon installations. Recent tests conducted on a prechamber equipped CFR engine have indicated that a modified convergent prechamber nozzle might improve the combustion characteristics of the system relative to the divergent nozzle used in the previous installations (Ref. 4-34).

Future projects are scheduled to include the incorporation of prechambers into a Datsun vehicle and operation of this vehicle on methanol and gasoline/methanol mixtures.

4.3.3 Cornell University

The Cornell spark plug concept is based on the observation that NO_x decomposes rapidly (in milliseconds) in the presence of unburned HC at temperatures above 2600°R providing there is a large excess of NO_x molecules relative to O_2 molecules. This phenomenon is attributed to the greater affinity of carbon and hydrogen to oxygen than to nitrogen. Concersely, the Zeldovich reaction mechanism, which is generally applied in NO_x formation models, requires residence times of the order of hundreds of seconds to achieve a substantial reduction in NO_x.

This phenomenon was first observed by Cornell in experiments in which the exhaust from a standard 1973 Pontiac engine was passed through a carbon steel pipe heated to 2600°R. The duration of the gas flow through the pipe was approximately 10 msec and in that time NO_x was reduced from the original level of 3000 ppm to the equilibrium concentration of less than 10 ppm. Apparently, neither the heated pipe walls nor the unburned hydrocarbons acted as catalyst in the process.

In the practical application of this phenomenon, a cavity around the spark plug was formed to contain a small amount of unburned hydrocarbons as illustrated in Figure 4-36 (Ref. 4-37). Most of the NO_x is probably formed in the mixture around the spark plug which burns first and is further heated by compression resulting from the combustion of the remaining charge. In the Cornell spark plug, a small cavity is formed around the spark plug, separated from the combustion chamber by a steel end-piece incorporating a small perforated plate. A fuel-rich mixture enters the cavity during the compression stroke, and remains unburned during the combustion phase because of the cool walls of the cavity. During the expansion stroke, the unburned hydrocarbons are then ejected into the high NO_x zone near the spark plug, thus promoting the reduction of NO_x (Ref. 4-37).

Based on the test data from a number of different plug designs, it appears that a cavity volume of about 0.5 to 1 percent of the combustion chamber volume is sufficient to achieve good effectiveness. It is important, however, to keep the cavity as cool as possible to prevent premature decomposition of the fuel in the cavity, and to operate the engine at air-fuel equivalence ratios larger than 1.08.

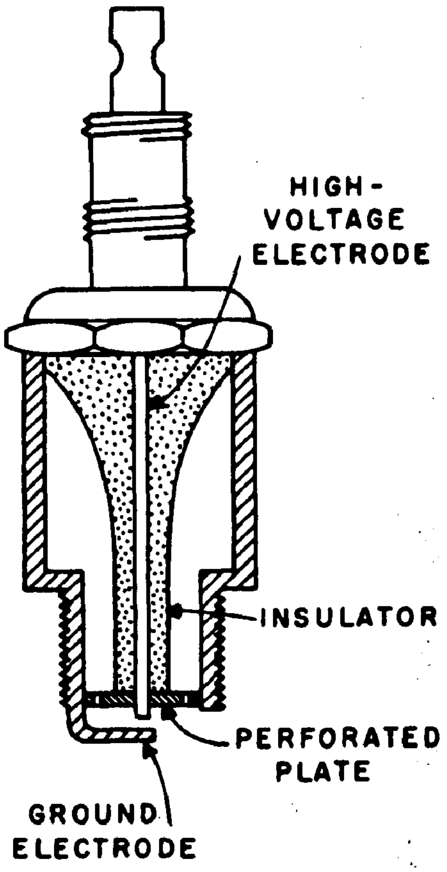


Figure 4-36. Cornell spark plug (Ref. 4-37)

4.3.3.2 Emission Characteristics

While The Cornell spark plug reduces NO_x, it has no effect on HC and CO. Therefore other techniques or devices such as catalytic converters or thermal reactors would be required to reduce the inherently high HC and CO emissions associated with rich mixture operation of the engine.

NO_x emission tests with the Cornell spark plug were conducted on a variety of engines. Initially, a single-cylinder Briggs and Stratton engine was tested in 1970 and the results from this work are presented in Ref. 4-38. In this case, the cylinder head was modified by drilling a hole to accept a small cavity of approximately 1 percent of the cylinder clearance volume. In the various tests carried out, substantial NO_x reduction was achieved and a number of criteria were established for the cavity design. These include (1) the requirement of adequate mixing in the cavity to obtain a non-flammable mixture; (2) an initially rich mixture in the cylinder to maintain a low oxygen environment; and (3) a cool cavity wall to prevent premature decomposition of the hydrocarbons. In the final configuration of the single-cylinder test, the cavity was incorporated within the spark plug, as shown in Figure 4-36.

Tests carried out on a 1967 six-cylinder Dodge Dart (compression ratio 8.4) at a constant speed of 35 mph have shown a reduction in NO_x from 3500 ppm with standard plugs to 400 ppm with the Cornell plugs (Ref. 4-37). Similar tests, carried out on a four-cylinder 1971 Volvo vehicle have shown less reduction in NO_x. This was attributed to the higher engine compression ratio of 10:1, which might be the cause of some hydrocarbon decomposition in the plug cavity before the beginning of the expansion stroke.

Another series of tests was conducted on a 1971 American Motors Matador which was equipped with a six-cylinder, 232 CID engine and a standard transmission. These tests which were performed by the New York State Vehicle Emission Laboratory at Albany, New York in accordance with the CVS procedure (no cold soak) included measurements on (1) the standard engine incorporating a proprietary mixture stratification system (TPDR) developed by Cornell and (2) measurements on this engine including TPDR and the Cornell spark plug. The emissions from the first two bags indicate that most of the observed reduction of NO_x was achieved with TPDR, while an additional 10 percent reduction was due to the Cornell spark plug. This is not unexpected since the concentration of NO_x around the spark plug is reduced with the TPDR system. Therefore, the benefit derived in this case from the Cornell spark plug consists mainly of a reduction in the NO_x peaks occuring during the cycle.

The composite emissions of the standard and modified Matador vehicles are listed in Table 4-28, indicating a substantial reduction in CO and NO_x, accompanied by an increase in HC (Ref. 4-39). To date, no explanation has been found for the observed rise in HC.

Table 4-28. 1971 AMERICAN MOTORS MATADOR PERFORMANCE COMPARISON; 1972 FTP; 12-hr COLD SOAK OMITTED (Ref. 4-39)

Emissions and fuel economy	Vehicle configuration	
	Standard vehicle	Modified vehicle (TPDR and Cornell spark plug)
HC, gr/mile	1.89	7.62
CO, gr/mile	66.39	6.96
NO_x,	4.76	1.85
Fuel economy, mpg	15.73	17.70

Finally, emission tests were conducted on a 1973 Dodge Dart vehicle equipped with a 225 CID slant six-cylinder engine and standard transmission using standard plugs and Cornell spark plugs. The standard exhaust gas recirculation system was removed and the crankshaft was replaced by an older version employing reduced valve overlap. Test data from this engine covering the first nine minutes of the CVS cycle (hot transient section) indicate a reduction in NO_x from the original level of 530 ppm, to about 340 ppm with the Cornell spark plugs. Unfortunately HC and CO data are not available from these tests.

4.3.3.3 Fuel Consumption Characteristics

The fuel consumption of an engine equipped with the Cornell spark plug should be comparable to that of a standard engine providing no other changes are made. The improvements in fuel consumption shown in Table 4-28 are attributed to the air stratification system (TPDR) which increases the manifold pressure, thus reducing the pumping losses.

On the other hand, if the application of the Cornell spark plug could be combined with reduced exhaust gas recirculation rates, some improvement in fuel consumption might be expected.

4.3.3.4 Vehicle Performance Characteristics

While no published data are available on the operation of the vehicles with the Cornell plugs, verbal communication has indicated comparable performance relative to a corresponding standard vehicle. This is to be expected since the carburetion, ignition, and combustion processes are virtually unaffected by the Cornell plug. No accurate data are available on the durability of the Cornell plugs in actual vehicle operation. However, no degradation was observed during the tests conducted to data (3500 miles).

4.3.3.5 Potential Problem Areas

The application of the Cornell spark plug to an automotive engine results in some reduction in NO_x. However, the device does not improve HC and CO, and other control techniques would be required for effective control of these species.

The NO_x reduction achieved with the Cornell plugs alone will not be sufficient to meet the 1978 standards. The efficiency of the device depends on several factors including the amount of hydrocarbons present in the plug cavity at the beginning of the expansion stroke, the concentration of NO_x around the spark plug, and the combustion chamber mixture ratio and temperature. Thus, the amount of NO_x reduction achieved is expected to vary for different engine designs and a certain amount of "tailoring" would be necessary to obtain the best results.

Other potential problem areas are related to the durability of the Cornell plug under actual engine/vehicle operating conditions and the tendency of plugging of the small holes or slots in the plug end cover.

4.3.3.6 Current and Projected Status

Present work at Cornell University on the Cornell spark plug is primarily concerned with the evaluation of the NO_x emission reduction potential for various engine/vehicle combinations. Future plans consist of durability testing of a selected plug configuration under normal vehicle operating conditions. In these tests, the mechanical and electrical characteristics of the plugs will be monitored at 5000-mile intervals.

Discussions are in progress with certain manufacturers of automotive engines regarding the incorporation of Cornell spark plugs in a vehicle test fleet.

In principle, The Cornell spark plug is a very simple device that could be incorporated into existing engines in place of the standard plugs. However, the related reduction in NO_x is expected to vary significantly for various engine designs. Therefore, its most probable application might be in conjunction with other NO_x control techniques, such as EGR, or reducing catalysts. In this case, the performance of the engine might be improved by permitting a reduction of the EGR flow rate, and the durability of the reduction catalyst might be extended. In any case, the future prospects of the device are predicated upon its acceptance by the automotive industry as a viable add-on to advanced emission control systems.

4.3.4 Stanford University (Heintz Concept)

The Ram Straticharge engine concept was patented by Heintz in 1958 for potential incorporation into new and existing four-stroke automotive spark-ignition engines (Ref. 4-40). In 1961 and

1963, two additional prechamber engine patents were issued to Heintz covering a modified version of his first four-stroke engine configuration (Ref. 4-41) and the application of the basic Ram Straticharge concept to two-stroke engines (Ref. 4-42).

Stanford University's involvement in prechamber engine work commenced in 1958 under the direction of Professor A. L. London with the start of a research program structured to demonstrate the exhaust emission and performance potential of the Heintz Ram Straticharge concept. The test engine utilized in that program was a modification of a 1957 Chrysler 392 CID, V-8 stock engine. The principal objective of the modification was to reduce the exhaust HC emissions of the engine to about 1 percent of the total fuel flow, which corresponds to a rollback of the HC emissions of about 80 percent relative to precontrolled automotive engines. Other design objectives included the demonstration of (1) efficient engine operation on a variety of distillate fuels and (2) better part-load fuel economy relative to conventional spark-ignition engines. Although these goals were eventually achieved, research on the engine was terminated in 1964 primarily because of a lack of interest on the part of the automotive industry regarding further development of the concept.

The development of two-stroke Ram Straticharge engines was jointly pursued by Heintz and Stanford University during the 1964-1970 time period. These efforts involved the manufacture and testing of a V-4 blower scavenged, 108.6 CID engine as well as adaptation of the Ram Straticharge concept to a stock DKW AU1000S, 58 CID automotive engine. Interest in the two-stroke engine was stimulated by the inherent advantages of this engine type relative to four-stroke engines, including higher power density, lower cost, and lower complexity.

With the advent of the prechamber engine-powered Honda CVCC engine in 1973, the development of the Heintz four-stroke Ram

Straticharge concept was resumed. The new program which is a joint effort between Heintz, Stanford University, and American Motors, is in progress and involves the adaptation of different Ram Straticharge modifications to a 1974 American Motors 232 CID Hornet engine (Refs. 4-43 and 4-44). To date, performance testing has been conducted on four different configurations (Mods. I through IV).

Pertinent test data from the various four-stroke and two-stroke engines are discussed in Subsections 4.3.4.1 and 4.3.4.2, respectively.

4.3.4.1 Four-Stroke Engines

4.3.4.1.1 Engine Description

The Heintz Ram Straticharge concept as incorporated in the 1957 Chrysler engine is shown in Figure 4-37 (Refs. 4-45 and 4-46). In this arrangement, the original spark plug wells were re-bored to accommodate the prechamber assemblies. Each prechamber has a laterally-disposed shrouded intake valve which is activated by the main intake valve train through a separate rocker arm. In general, the prechamber valve operates synchronously with the main chamber intake valve, although under certain circumstances the timing of its opening and closing might differ from the main valve.

A prechamber volume of 1.36 cu in. was selected for this engine representing about 18.6 percent of the total clearance volume. Since no other adjustments were made to the cylinder head, incorporation of the prechambers resulted in a reduction of the engine compression ratio from 9.25 to 7.7. As shown in Figure 4-37, the prechamber has a concentric trough near the top to prevent liquid fuel from entering the main combustion chamber. Also shown in the figure is the 10-mm Champion UY-6 spark plug located at the top of the prechamber. The prechamber is connected to the main chamber through a set of six small holes (0.0968-in. diameter) drilled into

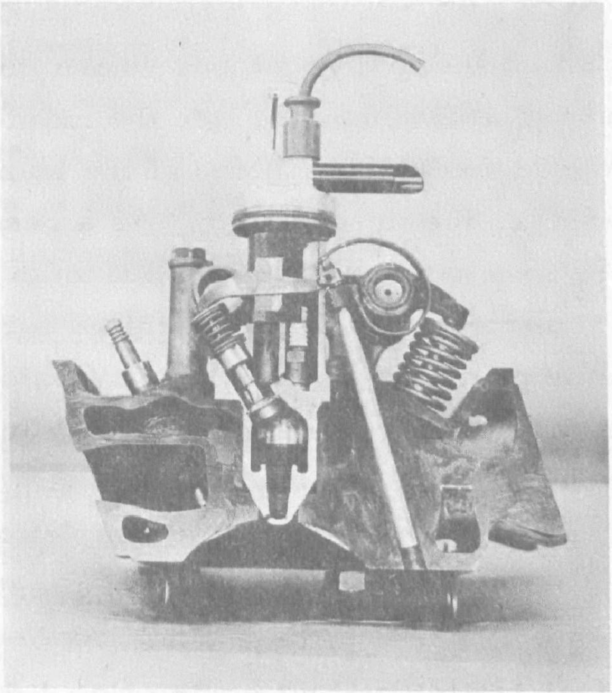


Figure 4-37. Heintz Ram Straticharge modification of 1957 Chrysler V-8, 392 CID engine (Ref. 4-45)

the tip of each unit. A constant flow fuel injection system is utilized instead of carburetion to reduce the throttling losses of the engine, particularly under part-load operating conditions. This system has separate injectors for the prechambers and the main chambers.

In the low-load and low-speed regime (loads up to 40 percent of maximum load and speeds up to 2500 rpm), all fuel is supplied to the prechamber injectors while the main chamber is operated on air alone. The engine was designed to operate unthrottled except, perhaps, for very low-load conditions where some throttling might be beneficial to minimize HC. Additional fuel is injected into the main chamber intake manifold during periods of high power demand. While the overall air-fuel ratio is varied with load, it is always adjusted lean.

As the prechamber and main chamber intake valves open, a fuel rich mixture is drawn into the prechamber. A portion of this fuel passes into the main chamber where mixing takes place with the air or lean mixture inducted into the main chamber. During the following compression stroke, some of the lean mixture from the main chamber reenters the prechamber. As a result, the initially rich mixture in the prechamber is leaned out to an air-fuel ratio between about 10:1 and 12:1 which represents a very desirable range from ignition, flame propagation, and NO_x emission points of view. Upon ignition, combustion proceeds very rapidly in the prechamber thus forcing hot combustion products into the main chamber where the combustion process is then carried to its completion.

The Heintz-Chrysler engine was tested with different distillate fuels including 83/91 military fuel, 68 octane naphtha, 100/130 aviation fuel, JP-4, and 50-octane No. 1 diesel fuel.

Testing of Modification I of the Hornet engine, which combines a standard Hornet engine block with a specially-cast head containing the prechamber (15 percent of the total clearance volume) was initiated at Stanford University in the spring of 1974 (Ref. 4-43 and 4-44). In these tests, the compression ratio of the engine was varied between 8.4 and 9.7 using a variety of prechamber manifolding schemes and spark plug locations, as well as prechamber carburetion. However, the operating characteristics of this engine proved to be very unsatisfactory. In the Modification II engine, the exhaust valves were recessed to facilitate unobstructed flow from the prechamber into the main chamber. The compression ratio was reduced to 7.9 and a custom-made camshaft was added. Modifications III and IV are similar to Modification II except for the prechamber nozzle size which was progressively reduced.

4.3.4.1.2 <u>Manufacturing Considerations</u>

While the questions of mass producibility and cost of the Ram Straticharge engine concept have not yet been accurately evaluated by the inventor, he has indicated that incorporation of this concept into existing engine designs would be possible at a fraction of the investment cost of a new engine design. Conversely, retrofitting of existing engines in the field would not appear to be feasible because of excessive cost and complexity.

4.3.4.1.3 Emission Characteristics

HC, CO, and NO_x emission data taken by Borns are presented in Figure 4-38 in terms of an emission index which is defined as the ratio of the mass of pollutant species emitted by the engine to the total engine fuel flow rate (Ref. 4-45). In these tests, the engine was operated on the dynamometer at a constant speed of 2000 rpm and a brake mean effective pressure of 56.9 psi. Other tests were conducted at 36.8 psi, 26.3 psi, and 17.5 psi. The air-fuel ratio was varied from less than 15 to 30, and the spark plug timing was always adjusted to the point of minimum advance for best torque (MBT). HC was measured by means of heated FID, while NDIR and chemiluminescence were used to measure CO and NO_x, respectively.

Similar to conventional spark-ignition engines, the NO_x emission of the Ram Straticharge engine decreases rapidly with increasing air-fuel ratio and decreasing bmep. At the 56.9-psi load condition, CO and HC have a minimum at air-fuel ratios of about 18 and 21, respectively. Both species increase rapidly as the air-fuel ratio is further increased. This rise is attributed to quenching of the HC and CO oxidation reactions which becomes more evident in leaner mixtures. At a given air-fuel ratio, both HC and CO tend to increase with decreasing load as a result of a reduction in the main chamber turbulence level caused by the low air flow rates associated with low power operation. Similar trends were observed by Morgan (Ref. 4-46).

The effect of spark retard on the emission characteristics of the Ram Straticharge engine was investigated by Borns for the

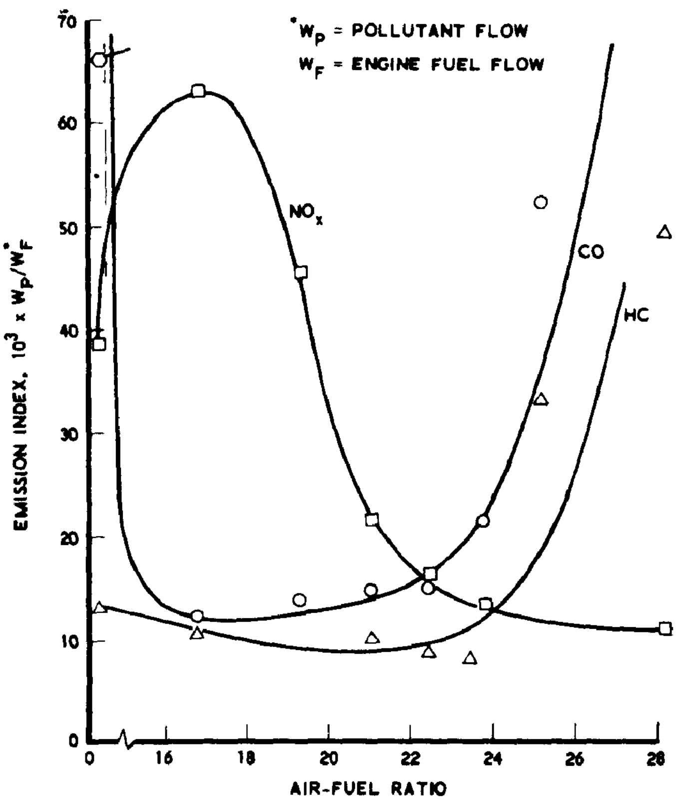


Figure 4-38. HC, CO, and NO_x emission index versus air-fuel ratio; four-stroke Heintz Ram Straticharge/ Chrysler engine; 56.9 psi; 2000 rpm; MBT spark advance (Ref. 4-45)

26.3-psi load condition. As expected, retarding the spark by 5 deg from the MBT setting resulted in substantially lower HC emission, at the expense of a 5 percent increase in specific fuel consumption. With retarded spark, the NO_x emissions were lower for air-fuel ratios below 21, while some increase was observed at higher mixture ratios. CO was only slightly affected by spark timing.

Comparison of the data published by Borns (Ref. 4-45) indicates that some emission control was achieved in the Ram Straticharge engine by operating the engine at air-fuel ratios between about 18 under light-load conditions and about 23 for medium loads. This is further illustrated in Table 4-29 which includes "optimum" HC, CO, and NO_x emission indexes for the four load-settings investigated by Borns. Also listed in the table are gram/mile values which were computed from the corresponding emission indexes by assuming an engine fuel economy of 14 mpg.

The HC, CO, and NO_x mass emissions of the Hornet Mod IV engine expressed in terms of gr/bhp-hr, are presented in Figures 4-39 through 4-41 as a function of air-fuel ratio and spark timing (Ref. 4-44). Again, minimum HC and CO emissions were obtained for air-fuel ratios between 17 and 19, while NO_x decreased steadily with increasing air-fuel ratio.

Table 4-29. RAM STRATICHARGE/CHRYSLER ENGINE EMISSIONS AT STEADY-STATE; MBT SPARK ADVANCE; 2000 rpm (Ref. 4-45)

bmep, psi	Air-fuel ratio	Emission index[a]			Computed emissions, gr/mile		
		HC	CO	NO_x	HC	CO	NO_x
56.9	23	10^{-2}	1.8×10^{-2}	1.4×10^{-2}	2.00	3.6	2.8
36.8	22	1.6×10^{-2}	2.4×10^{-2}	1.5×10^{-2}	3.6	4.8	3.0
26.3	20	1.9×10^{-2}	1.8×10^{-2}	1.3×10^{-2}	3.8	3.6	2.6
17.5	18	1.5×10^{-2}	1.6×10^{-2}	0.9×10^{-2}	3.0	3.2	1.8

[a]Ratio of pollutant flow to engine fuel flow

4.3.4.1.4 <u>Specific Fuel Consumption Characteristics</u>

Measured specific fuel consumption data taken by Borns at a constant engine speed of 2000 rpm are plotted in Figure 4-42 as a function of air-fuel ratio and bmep (Ref. 4-45). As indicated,

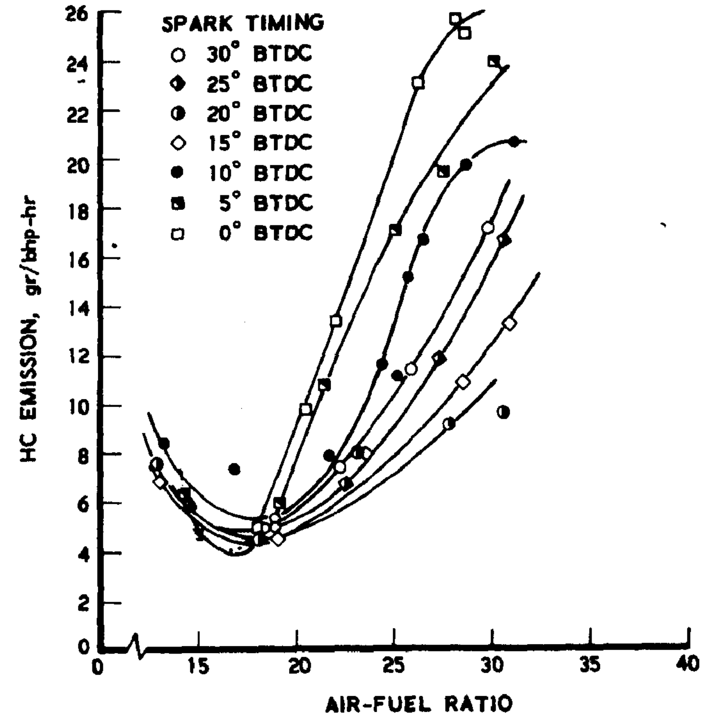


Figure 4-39. HC emission versus air-fuel ratio; Heintz/Hornet engine Mod. IV; 2000 rpm; 39.1-psi bmep (Ref. 4-44)

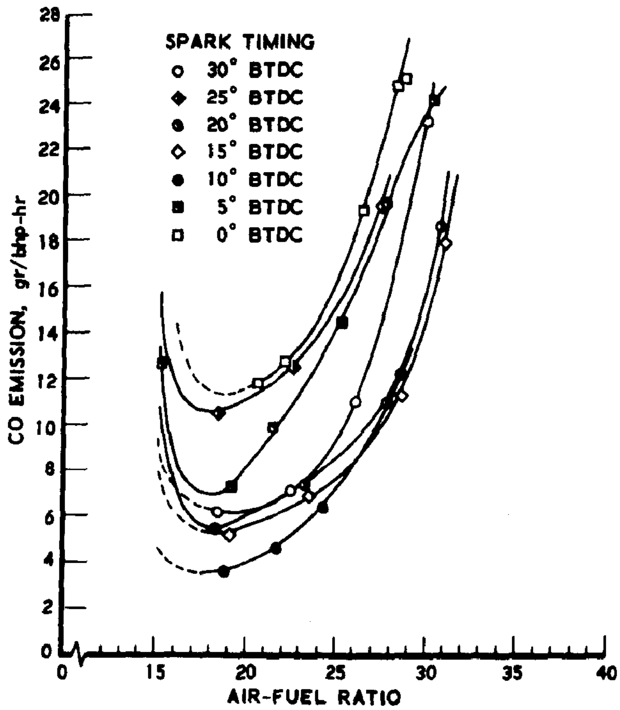


Figure 4-40. CO emission versus air-fuel ratio; Heintz/Hornet engine Mod. IV; 2000 rpm; 39.1-psi bmep (Ref. 4-44)

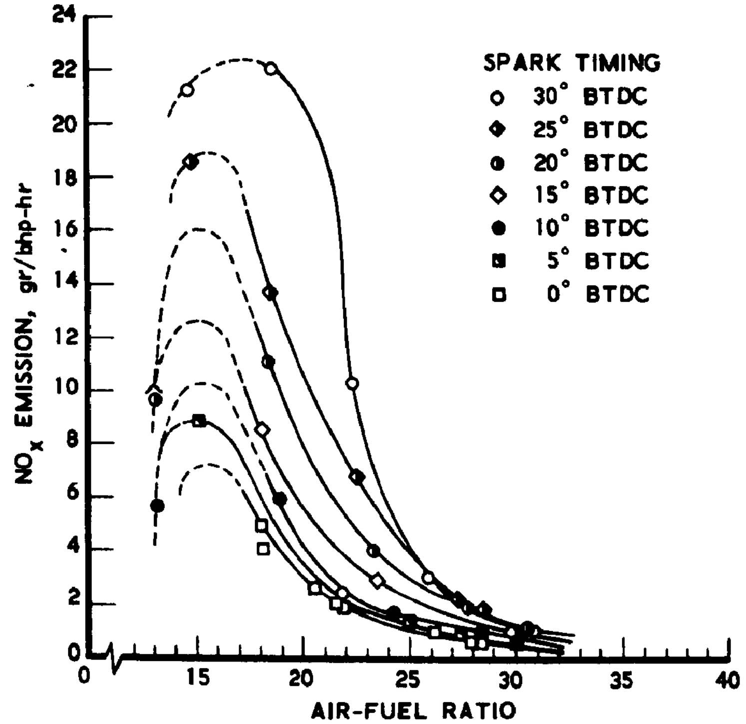


Figure 4-41. NO_x emission versus air-fuel ratio; Heintz/Hornet engine Mod. IV; 2000 rpm; 39.1-psi bmep (Ref. 4-44)

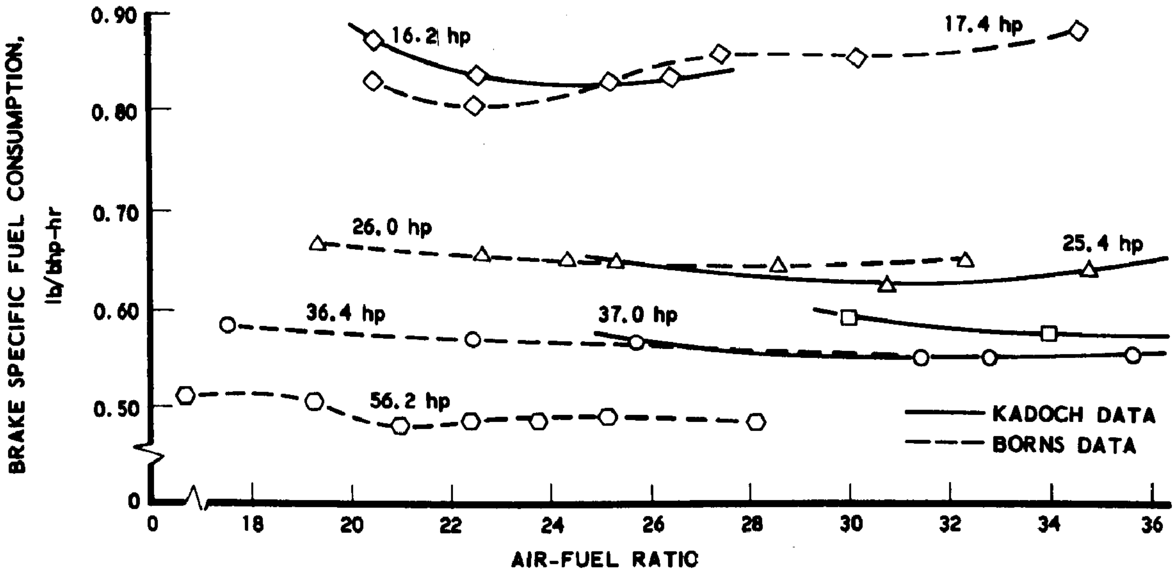


Figure 4-42. Brake specific fuel consumption versus air-fuel ratio; Heintz Ram Straticharge/Chrysler engine (Ref. 4-45)

Borns' data are in good agreement with earlier data obtained by Kadoch (Ref. 4-47). As expected from conventional engine maps and theoretical considerations, the specific fuel consumption of the Ram Straticharge/Chrysler engine increases with decreasing load and decreases slightly with increasing air-fuel ratio.

Specific fuel consumption data taken on the modified Chrysler engine with five different distillute fuels indicate that the Ram Straticharge engine configuration is superior to the stock engine at very light loads regardless of the fuel type utilized. This is attributed to lower throttling losses and better mixing in the Ram engine. However, at higher loads, the fuel consumption of the stock engine is lower than that of the prechamber engine except when 83/91 military fuel is used. It should be noted, however, that the compression ratio of the Ram Straticharge engine was only 7.78, compared with 9.25 for the stock engine.

The minimum specific fuel consumption of the Hornet Mod. II engine was comparable to that of the stock engine, but higher than for the Chrysler engine discussed above. At part load, the engine operated satisfactorily at air-fuel ratios exceeding 28. Successful idling was achieved at an air-fuel ratio of 60. The principal feature of the Mod. III engine is its very flat brake specific fuel consumption (bsfc) versus air-fuel ratio characteristic, showing a slight reduction in bsfc as the air-fuel ratio was increased from 15 to about 27. The specific fuel consumption of the Mod IV engine is plotted in Figure 4-43 as a function of air-fuel ratio and spark timing. Relative to Mod. III, this engine shows bsfc improvements of the order of 10 percent (Ref. 4-44).

4.3.4.1.5 Potential Problem Areas

Based on the test work conducted on the Ram Straticharge/Chrysler engine, Borns has identified a number of potential

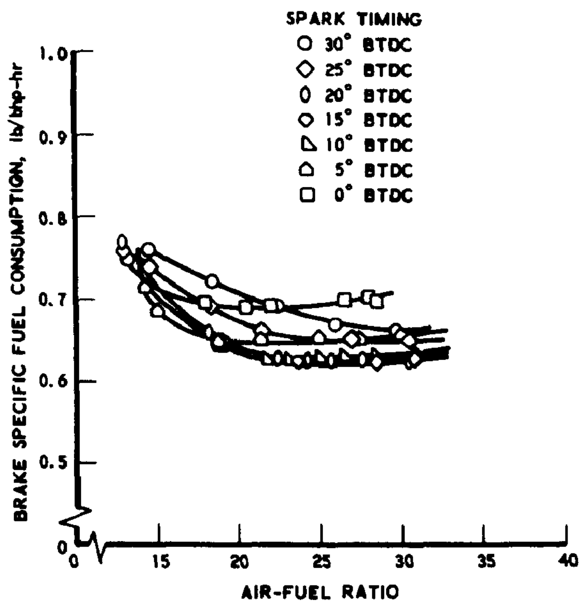


Figure 4-43. Brake specific fuel consumption versus air-fuel ratio; Heintz/Hornet engine-Mod. IV; 2000 rpm; 39.1-psi bmep (Ref. 4-44)

problem areas which are inherently related to the particular prechamber design utilized in his program. For example, the placement of the prechamber assembly between the engine main intake and exhaust valves generated high temperature gradients in that area which might cause cylinder head and intake valve warpage. Also, because of the close proximity of the prechamber to the exhaust valve, some of the partially burned residual gas exiting from the prechamber toward the end of the expansion stroke might be discharged through the exhaust valve before further oxidation could take place, thus creating a potential emission problem.

Additional difficulties have surfaced during light-load operation of the modified Chrysler engine. In this case, the low turbulence level, which is characteristic of hemispherical combustion chambers, inhibits complete mixing and combustion of the partially burned gases emanating from the prechamber. Moreover, the selected prechamber volume was apparently too large, resulting in quenching of the reactions at light loads and excessive HC and CO emissions. Overheating of the prechamber is another potential problem area, causing fuel preignition and damage to the prechamber intake valve and nozzles.

Starting of the engine on No. 1 diesel fuel proved to be impossible unless a small amount (about 2 cc) of gasoline was used as primer.

4.3.4.1.6 Current and Projected Status

The Heintz Ram Straticharge concept, when incorporated in a Chrysler V-8 engine, has been successfully operated on a variety of distillate fuels, and has achieved respectable performance levels, particularly under part-load operating conditions. While low CO and NO_x emissions have been achieved at air-fuel ratios between about 22-26, the high HC emission of the engine represents a serious problem area, particularly at high air-fuel ratios.

Although current plans do not include the optimization of the prechamber size and shape, it appears that some benefit in terms of lower bsfc and HC emission could be derived from such a program. Also, incorporation of a turbocharger might prove to be beneficial from a bsfc and emissions point of view.

Testing of the Ram Straticharge/Hornet engine is in progress at Stanford University, and the work is projected to continue through 1975. The objective of future research is to examine the effect of many parameters on combustion performance, including the prechamber nozzle geometry, turbulence level, prechamber wall temperature, combustion chamber shape, and supercharging (Refs. 4-43 and 4-44).

4.3.4.2 Two-Stroke Engine

4.3.4.2.1 Engine Description

The two-stroke version of the Heintz Ram Straticharge engine is illustrated in Figure 4-44 (Ref. 4-48). The engine has a displacement of 108.6 cu in. and consists of a cast iron block in the form of a 90-deg V with two cylinders in each bank. The overall compression ratio of the engine is 8.6 with a compression ratio from exhaust port closing of about 6. Important engine design parameters are presented in Ref. 4-44.

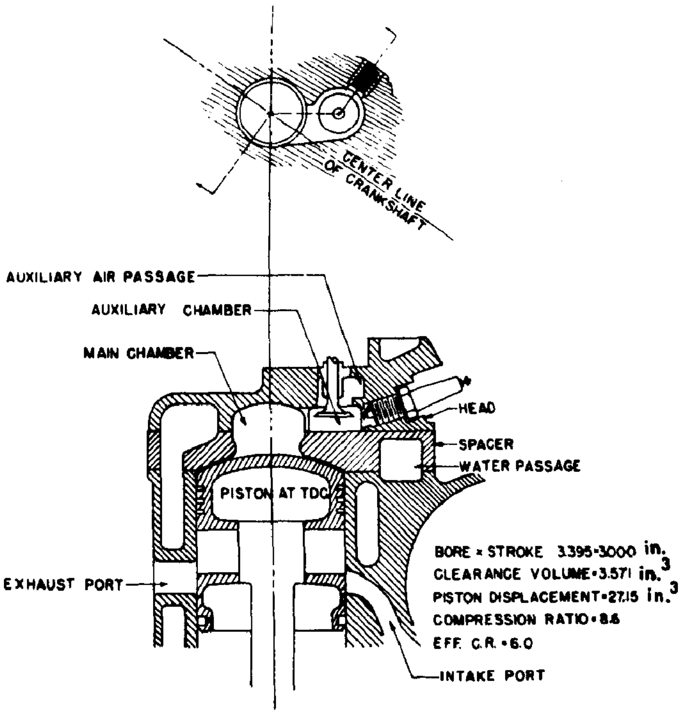


Figure 4-44. Heintz two-stroke Ram Straticharge engine (Ref. 4-48)

The cast iron cylinder head and spacer arrangement shown in Figure 4-43 was made in two pieces to simplify the casting and machining operations. Each prechamber has an intake valve which is operated by means of a rocker arm and a push rod from a single camshaft centrally located in the V of the engine.

A two-stage, belt-driven Roots blower is employed to provide the air to the prechambers and main chambers. Most of the air leaving the first blower stage enters the main combustion chamber through the intake port, while the remainder is further compressed in the second blower stage before admittance to the prechamber.

All fuel required for engine operation is injected continuously into the auxiliary air passage located upstream of the prechamber intake valve. Fuel flow control is accomplished by means of an adjustable fuel pressure line bleedoff. To maintain acceptable flame speed, the air-flow might be throttled at light-load conditions.

Charging of the prechamber and main chamber is unique in this design in that two separate streams enter during the scavenging process. The engine exhaust port starts to uncover at 99 deg after top dead-center, and the intake port begins to open about 16 deg later to admit air into the main chamber. Upon opening of the prechamber poppet valve near bottom dead-center, a flow of fuel-rich mixture enters the prechamber. Similarly to the previously discussed four-stroke engine, a portion of this mixture proceeds to the main chamber and forms a very lean mixture in that chamber. As the piston moves up, part of the main chamber mixture is forced back into the prechamber. The high degree of swirl imparted on the flow by the geometry of the chamber forces the fuel to the periphery of the prechamber and forms an ignitable air-fuel mixture in the vicinity of the spark plug. Upon ignition, the hot gases emanating from the prechamber are rapidly mixed with the swirling main chamber charge to assure rapid flame propagation and good combustion throughout the main chamber.

4.3.4.2.2 Emission Characteristics

The HC emissions of the engine are very sensitive to the selected air-fuel ratio and cam timing. In general, the data reported in Ref. 4-44 indicate rapidly increasing HC as the air-fuel ratio was increased, and a reduction in HC as engine speed increased. Conversely, little change in HC was observed over a wide range of spark timing. Over the range of brake mean effective pressures evaluated by Fandrich (20 to 80 psi), the HC emission index varied between 0.001 and 0.07, depending upon engine speed and cam dwell angle.

4.3.4.2.3 Fuel Consumption Characteristics

Specific fuel consumption data obtained by Fandrich for the Heintz two-stroke Ram Straticharge engine operated at 2000 rpm are plotted in Figure 4-45 as a function of engine load and cam/oil ring arrangement (Ref. 4-48). Also shown in this figure are data from a standard DKW-AU 1000S, 58 CID, two-stroke engine and a modified DKW engine incorporating a Ram Straticharge head plus fuel injection. Best fuel consumption was achieved with the Heintz engine equipped with a 90-deg cam. While the modified DKW engine had higher fuel consumption than the Heintz engine, it was superior to the standard DKW engine, particularly at part-load where the bsfc improvements were as high as 15 percent.

The tests reported here were conducted with regular gasoline or naphtha, and engine knock was never observed on the prechamber engines, indicating a lower octane requirement of this engine-type relative to conventional spark-ignition engines.

4.3.4.2.4 Potential Problem Areas

The principal problem encountered on the Heintz two-stroke Ram Straticharge engine is related to the continuous fuel

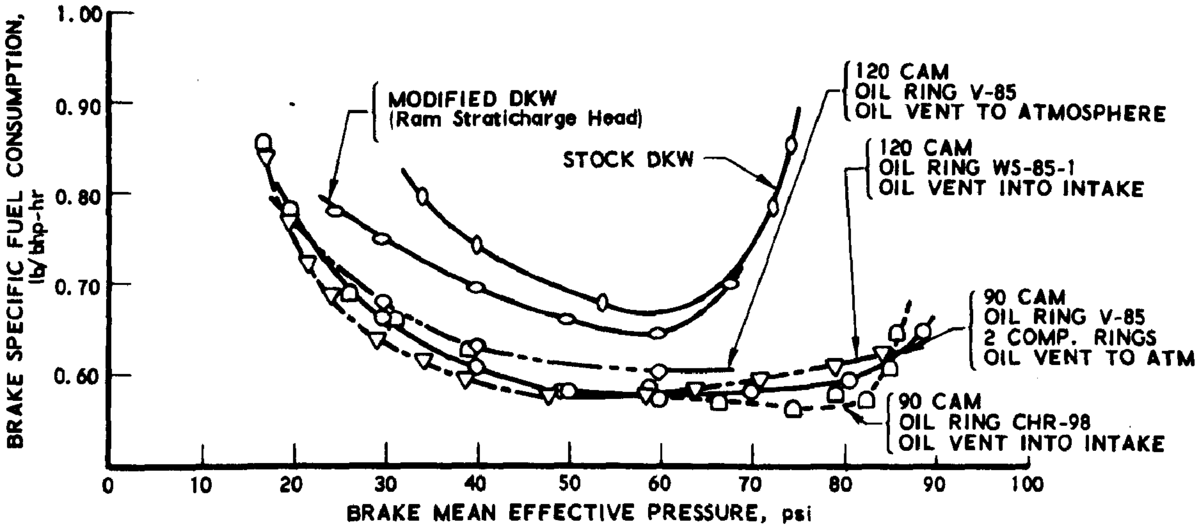


Figure 4-45. Brake specific fuel consumption versus brake mean effective pressure; Heintz two-stroke Ram Straticharge engine; 2000 rpm (Ref. 4-48)

injection system utilized on the engine. As the prechamber valve opens, a slug of fuel, which was accumulated during the preceding period of valve closure, passes through the prechamber into the main chamber and directly out into the exhaust. As a result, the specific fuel consumption of the engine and the HC emission are adversely affected. A number of techniques have been proposed to alleviate this problem, including timed fuel injection and relocation of the fuel injectors away from the poppet valve (Ref. 4-48). Other potential problem areas are related to failures of the poppet valve stem and face. Modification of the cam profile has been suggested as a potential solution to reduce the acceleration and deceleration rates of the valve.

4.3.4.2.5 Current and Projected Status

While the two-stroke Ram Straticharge combustion concept has demonstrated improved specific fuel consumption characteristics relative to equivalent conventional two-stroke engines, the efficiency of this engine type is considerably lower than that of four-stroke engines. In addition, the HC emission would have to be reduced substantially to meet current and projected light-duty vehicle emission regulations. Incorporation of timed fuel injection, improved engine scavenging, and a variable-speed Roots blower are projected to improve both fuel economy and HC emission.

Because of the declining interest on the part of the engine manufacturers in two-stroke light-duty automotive engines, no additional efforts are planned by Heintz/Stanford University on this particular concept.

4.3.5 University of California at Berkeley

Research on prechamber spark-ignition engines has been in progress at the University of California at Berkeley for some time. Current efforts involve the determination of the emission and fuel consumption characteristics of a CFR engine modified for prechamber operation.

4.3.5.1 Engine Description

The prechamber CFR engine is shown schematically in Figure 4-46 (Ref. 4-49). It consists of a cetane cylinder head which was modified to accept a fuel injector and a spark plug in the prechamber The prechamber occupies a volume of 1.94 cu in., or 38 percent of the total clearance volume. Injection of a fuel-rich mixture into the prechamber is accomplished with a Spica injector pump fitted with a Mercedes-Benz injector nozzle. A lean mixture is carbureted into the main chamber. To date, exploratory tests have been conducted on

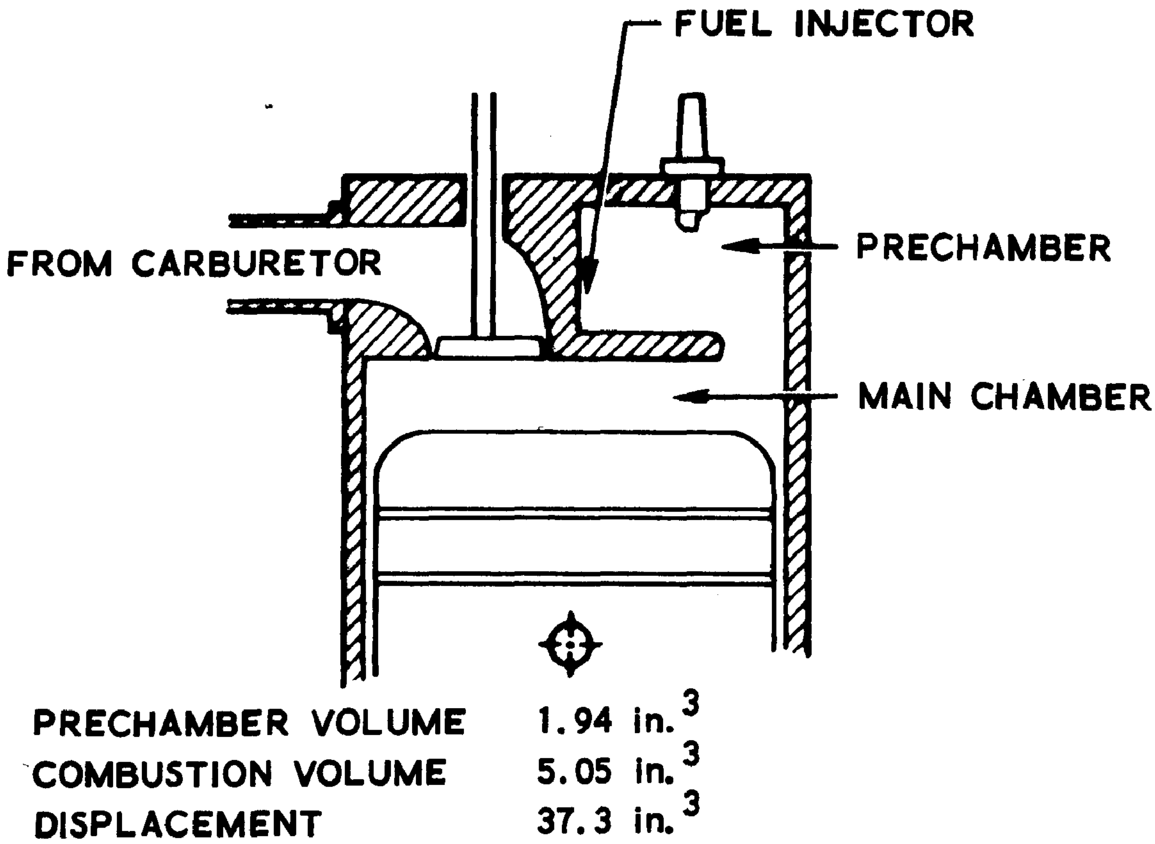


Figure 4-46. University of California prechamber engine configuration (Ref. 4-49)

gasoline (Chevron Supreme) and a gasoline/methanol blend containing 20 percent methanol (by volume).

Injection of the fuel was initiated at 135 deg before top dead-center (BTDC). Both emissions and power output were found to be insensitive to variations in injection timing ranging from 170 to 50 deg BTDC. However, the power output was quite sensitive to ignition timing and all data were taken at MBT timing which was 15 deg BTDC over a very wide range of air-fuel ratios for both fuels tested.

The engine displayed a wide operating range with respect to both prechamber and main chamber equivalence ratios. Three delivery rates were chosen for the injector and for each setting wide ranges of prechamber, main chamber, and overall air-fuel ratios were achieved by adjusting the carbureted mixture.

On gasoline, steady operation of the engine was maintained for main chamber air-fuel equivalence ratios between 0.64 and 0.34 and prechamber equivalence ratios between 1.1 and 1.4, corresponding to overall equivalence ratios between 0.94 and 0.62. Similar results were found for the gasoline/20 precent methanol mixture.

4.3.5.2 Emission Characteristics

Initial emission data obtained on the University of California prechamber engine are presented in Table 4-30 for both gasoline and gasoline/20 percent methanol.

Table 4-30. RANGE OF EMISSION DATA; UNIVERSITY OF CALIFORNIA PRECHAMBER ENGINE (Ref. 4-49)

Fuel	Emissions, gr/iph-hr		
	HC	CO	NO_x
Gasoline	2.8 - 31	4.0 - 80	1.4 - 5
Gasoline/20% Methanol	3.0 - 39	10 - 130	0.9 - 6

While the exhaust emissions show a complicated dependence on prechamber and main chamber equivalence ratio, the following trends have been derived from the test data.

1. For a given fuel injector delivery rate to the prechamber, CO decreases and HC increases with increasing air-fuel ratio.
2. For a given overall air-fuel ratio, both CO and HC decrease with increasing prechamber air-fuel ratio.
3. NO_x increases as the prechamber is leaned. With a relatively rich prechamber, NO_x assumes a minimum at an overall equivalence ratio of about 0.77.

Operation of the engine with air-fuel ratios adjusted for minimum fuel consumption resulted in the emissions listed in Table 4-31, along with typical data for a conventional CFR engine without prechamber.

Table 4-31. EMISSIONS FOR BEST FUEL CONSUMPTION SETTING; UNIVERSITY OF CALIFORNIA PRECHAMBER ENGINE (Ref. 4-49)

Engine	Fuel	Fuel-air equivalence ratio	Emissions, gr/ihp-hr		
			HC	CO	NO
Prechamber CFR	Gasoline	0.7	3.5	7.9	3.9
Prechamber CFR	Gasoline/ 20% Methanol	0.7	3.0	15.0	3.4
Conventional[a] CFR	Gasoline	1.1	2.0	60.0	6.0
Conventional[a] CFR	Gasoline	0.82[b]	0.7	0.5	26.0

[a]Data from Wimmer and Lee (Ref. 4-22)

[b]Lean limit

It appears that a reduction in NO can be obtained in the conventional engine only at the expense of extremely high CO emissions. Conversely, the dual-chamber engine shows simultaneous reduction in NO and CO, but shows relatively high HC emission.

The alcohol blended fuel showed slight reductions in HC and NO, and some increase in CO, when compared with the unblended gasoline at the same equivalence ratio.

4.3.5.3 Fuel Consumption Characteristics

With gasoline, the minimum fuel consumption of a conventional CFR engine occurs near the lean limit and is about 0.40 lb/ihp-hr (Ref. 4-22). The minimum fuel consumption measured for the University of California prechamber engine was 0.378 lb/ihp-hr at an overall equivalence ratio of 0.7, which represents an improvement over the conventional engine of about 6 percent.

With the alcohol mixture, the prechamber engine showed a minimum fuel consumption of 0.394 lb/ihp-hr, which is slightly higher than that of the gasoline. However, due to the difference in density and heat content of gasoline and the gasoline/methanol blend, comparison of fuel consumption on a lb/ihp-hr basis is not a good indication of the efficiency of the combustion process. For this reason, the fuel consumption should be reported in terms of the available Btu's burned per indicated horsepower-hour. On this basis, the alcohol mixture shows an 8 percent higher cycle efficiency than gasoline, when compared at the same equivalence ratio.

In general, best fuel consumption was obtained with a slightly rich prechamber mixture (equivalence ratio 1.1). For a given injector delivery rate, the optimum fuel consumption was obtained at an overall equivalence ratio of about 0.75.

4.3.5.4 Current and Projected Status

The prechamber engine research work conducted at the University of California is part of an NSF-sponsored effort which is aimed at achieving stabilization of lean combustion in automotive engines. Future efforts are scheduled to include (1) optimization of the prechamber geometry, (2) operation of the prechamber on methanol and other alternative fuels, and (3) evaluation of the knocking characteristics and octane requirements of prechamber engines.

4.3.6 University of Rochester (Broderson Concept)

Research work on the Broderson concept of charge stratification (Refs. 4-50 and 4-51) was conducted at the University of Rochester during the time period between the late 1940s and the late 1960s. In the course of these efforts, a number of engines were equipped with Broderson-type prechambers and were tested on the engine dynamometer using both gasoline and gaseous fuels. While some attempts were made to optimize the prechamber geometry with respect to fuel economy and driveability, additional improvements might be achieved by further modifying certain system design parameters.

4.3.6.1 Engine Description

In principle, the Broderson patents cover two different design approaches. The first configuration consists of (1) a small precombustion chamber, (2) a fuel injection system, (3) an auxiliary intake valve for scavenging which operates synchronously with the main chamber intake valve, (4) a conventional spark plug, and (5) a small communicating passage connecting the prechamber and main chamber (Ref. 4-50). In this design, all fuel required for engine operation is injected into the prechamber. Charge stratification is achieved by utilizing the change in direction of the air flow through the communicating passage as the piston passes bottom dead-center. Fuel injected into the prechamber during the suction stroke is carried into the main chamber. Conversely, the fuel injected during the compression stroke remains in the prechamber and forms a near-stoichiometric mixture in the vicinity of the spark plug. Control of the air-fuel ratio in the two chambers is achieved by adjusting the start of the injection process and the duration of injection. At idle and light loads, injection takes place only during the compression stroke. As the load demand increases, the injection period is extended, starting during the latter phase of the suction stroke.

The rate of combustion in the prechamber is essentially constant at all loads, because the air-fuel ratio is kept nearly constant. However, the flame propagation characteristics in the main chamber may be varied by varying the turbulence level in that chamber and the shape and location of the communicating flow passage. While the concept, as originally conceived by Broderson, was intended to operate without main chamber throttle, a certain amount of throttling might be desirable at light loads to assure smooth combustion in that operating regime.

Conceptually, the second configuration patented by Broderson (Ref. 4-51) is very similar to the approach discussed above, except that a conventional carburetor is utilized in the main intake circuit to provide a lean mixture for the main chamber instead of pure air. Again a near-stoichiometric mixture is maintained in the prechamber at all times.

The prechamber research effort at the University of Rochester was primarily concerned with the second configuration, which was incorporated into a single-cylinder CFR engine and into one cylinder of an L-141 military (Jeep) engine. As shown in Figure 4-47, the prechamber adapted to the CFR engine consisted of a 0.75-in. diameter cylindrical section, with the fuel injector threaded into the top of the chamber (Ref. 4-52). The auxiliary intake valve which has a port diameter of 0.375 in., was actuated by a flexible cable connected to the rocker arm of the main intake valve. This arrangement resulted in near-synchronous opening of the two valves. The prechamber volume, measured up to the throat of the converging-diverging passage was 0.92 cu in., or 22.2 percent of the total clearance volume based on an engine compression ratio of 10. Cooling of the prechamber was accomplished by means of air which was forced over the cooling fins of the unit.

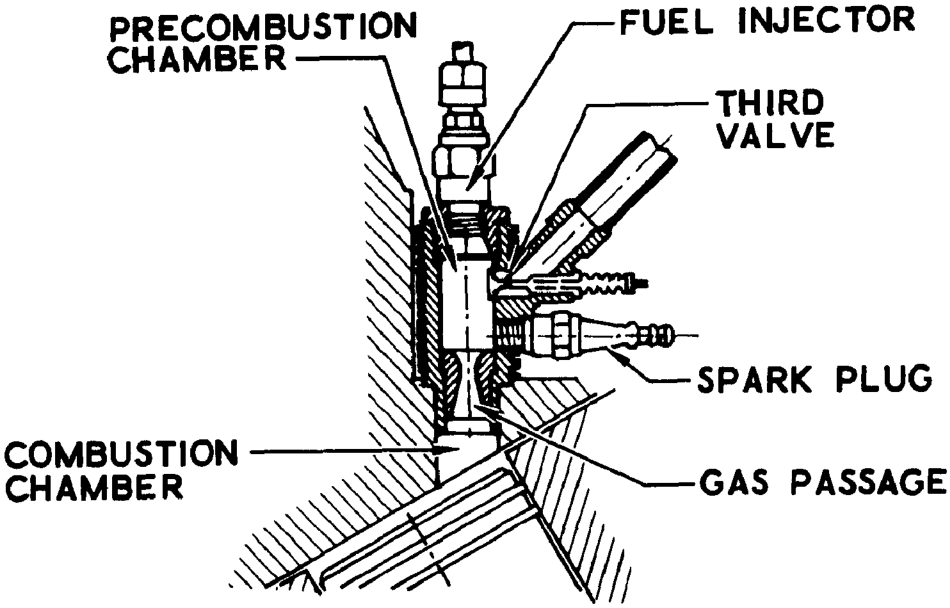


Figure 4-47. CFR engine installation of Broderson prechamber engine concept. (Ref. 4-52)

The prechamber system incorporated into the L-141 engine was similar to the CFR engine configuration, except that a larger connecting nozzle was utilized and the spark plug was moved closer to the injector. For the selected compression ratio of 8.8, the resulting prechamber volume of 1.32 cu in. represents 29 percent of the total clearance volume (Ref. 4-53). In this design, the prechamber valve was actuated by a small tappet which was driven by an extended camshaft arrangement.

4.3.6.2 Performance Characteristics

The first experimental data from the Broderson prechamber concept were taken on a modified Palmer T-head engine, which was operated at 800 rpm and compression ratios of 7.24 and 8.15. In the light-load regime, the thermal efficiency of this engine was up to 20 percent better than that of the standard engine (Ref. 4-54).

Encouraged by these results, the University of Rochester incorporated a Broderson prechamber into a single-cylinder Palmer diesel engine and tested this engine at compression ratios of 7.18 and 8.19 and speeds between 600 and 1000 rpm (Ref. 4-55). While the thermal efficiency of the engine was slightly improved relative to the baseline engine, the observed efficiency was lower than expected. The apparent loss in performance was attributed to chamber wall wetting caused by the particular fuel injection system utilized on the engine. Subsequent tests conducted with propane on this engine (Ref. 4-56), and on the CFR engine (Refs. 4-57 and 4-58) confirmed this hypothesis.

The modified CFR engine was submitted to extensive testing using gasoline as the fuel (Ref. 4-53). Initially, the prechamber unit was optimized in a preliminary test series by varying the external wall temperature between 200 and 340°F, the position of the spark plug and the spark plug gap, and the spark plug heat range. These changes had minor effects on engine performance. In addition, a number of different converging-diverging nozzle configurations were evaluated. A nozzle throat diameter of 0.125 in. resulted in good overall performance characteristics. At 1200 rpm, the indicated specific fuel consumption varied between about 0.37 lb/ihp-hr at 1 ihp to about 0.34 lb/ihp-hr at 5 ihp. Even lower isfc values were obtained on this engine in the low-to-medium-load regime by using a 0.25-in. nozzle and installing it in an inverted position, i.e. with the small angle section facing the prechamber. As shown in Figure 4-48

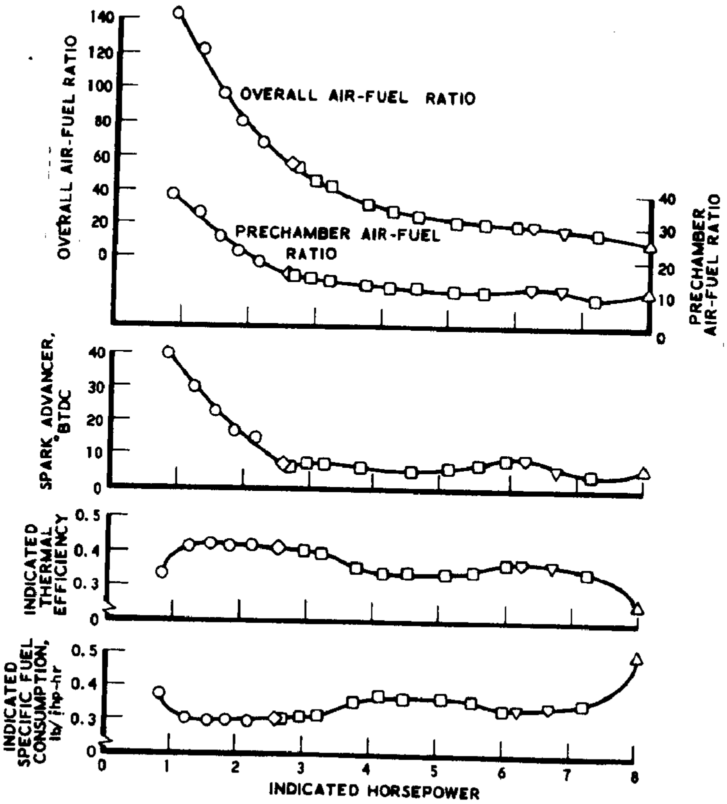


Figure 4-48. Broderson prechamber/CFR engine performance characteristics; 1200 rpm; 0.25-in. nozzle; premium gasoline; compression ratio 12:1 (Ref. 4-53)

(Ref. 4-53), a maximum indicated efficiency of 42 percent was achieved in the 1.5 and 2 ihp power-output regime, compared to about 38 percent for the 0.125-in. nozzle. The spark advance curves show rather low values except for very light loads, indicating good flame propagation characteristics throughout the operating range. While the spark timing shown in Figure 4-48 was optimum for this engine, the effect of timing on isfc was rather small. Similar results were obtained at 1800 rpm.

Based on visual inspection, no smoke was observed at light loads. However, traces of smoke were noted at half load, and smoke levels of medium intensity were obtained at full load.

Detonation-like combustion noise has been encountered at light loads and also at higher load levels when the 0.25-in. nozzle was utilized. Tests with different nozzle sizes indicated that noise was directly related to the mass flow rate of the hot combustion gases through the nozzle at light loads and the very rapid rate of combustion occuring at heavy loads. Incorporation of a larger nozzle resulted in higher noise intensity, whereas some reduction in the noise level was achieved with smaller nozzles at the expense of frequent occurance of combustion instability.

Exploratory tests conducted with the prechamber valve closed indicated that the engine was operable, but a loss in fuel economy of about 10 percent was encountered (Ref. 4-49).

Test data from the modified L-141 engine are presented in Figure 4-49 (Ref. 4-53). In these tests, the engine was operated at 1500 rpm and fuel injection was initiated at about 100 deg before top dead-center. Also shown in the figure are indicated specific fuel consumption data for the nonmodified L-141 engine. Comparison of the data shows that some improvement in isfc was achieved with the prechamber engine in the light-load regime, while little difference was obtained for power settings above about 3 ihp. Similar to the CFR engine, the modified L-141 engine developed some combustion noise which varied with nozzle size and engine operating conditions. Comparable performance and noise characteristics were obtained on this engine at 1000 rpm and 2000 rpm.

Because of the nonavailability of adequate instrumentation at the University of Rochester, no emission data were taken on these engines. Also, no information is available regarding the operational characteristics of the modified engines under transient conditions.

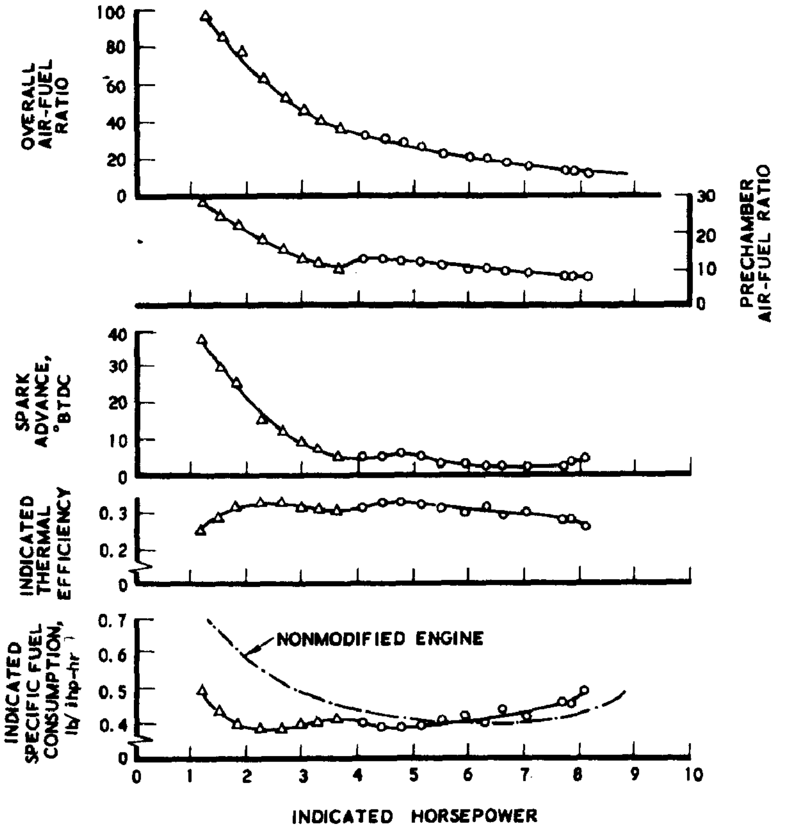


Figure 4-49. Broderson prechamber/L-141 engine performance characteristics; 1500 rpm; start of injection 100 deg BTDC; premium gasoline; compression ratio 8.8:1 (Ref. 4-53)

4.3.6.3 Potential Problem Areas

No serious problems were incountered in the various research programs conducted on the Broderson prechamber engine concept. While objectionable noise and exhaust smoke levels were observed under certain operating conditions, it appears that these problems might be alleviated by incorporation of additional design modifications and variations affecting the size and shape of the nozzle, prechamber, and fuel injector.

4.3.6.4 Current and Projected Efforts

Unthrottled operation of the Broderson prechamber engine concept has been successfully demonstrated at steady-state over wide ranges of air-fuel ratio and power output. As predicted from theory, the thermal efficiency of the Broderson concept at light loads was significantly higher than that of the standard engines, while comparable efficiency levels were achieved in the high-load regime. Further performance improvements might be achieved by optimizing the fuel injection system, valve timing, prechamber shape, nozzle geometry, and turbulence level in the mainchamber.

Currently there are no plans for further research on the Broderson prechamber engine concept.

4.3.7 University of Wisconsin (Newhall Concept)

The Newhall prechamber or divided chamber concept is in an early development stage, and testing has been confined to single-cylinder research conducted at the University of Wisconsin, and by the Ford Motor Company.

4.3.7.1 System Description

The Newhall prechamber concept is shown schematically in Figure 4-50 (Ref. 4-59). In this design, the prechamber which occupies about 65 percent of the total clearance volume, communicates with the remaining part of the combustion chamber through a rather large passage. All fuel is injected into the prechamber at an intermediate point in the compression stroke and the air is drawn into the engine through the intake valve. The spark plug is located near the dividing passage. Upon ignition, the flame propagates into the prechamber away from the passage region and the high pressure formed in the prechamber during combustion forces the high temperature combustion products into the main chamber.

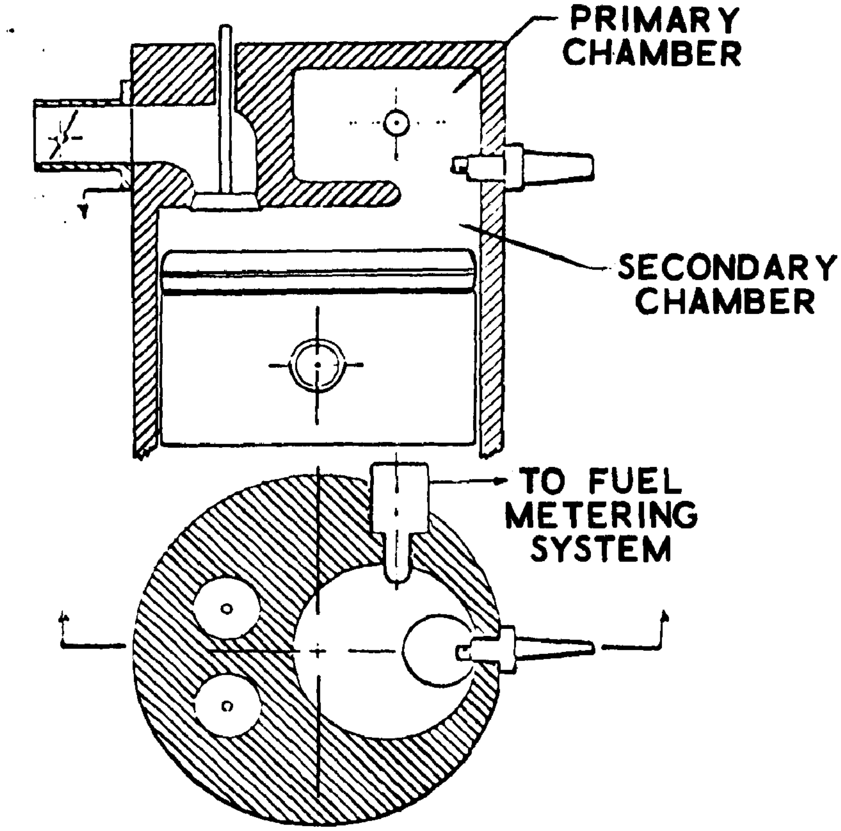


Figure 4-50. Newhall prechamber engine (Ref. 4-59)

Typically, the prechamber operates at rich air-fuel mixtures while the overall air-fuel ratio is leaner than stoichiometric. For instance, for an overall equivalence ratio of 0.75, the equivalence ratio of the prechamber mixture during combustion is approximately 1.2 (Ref. 4-60).

Compared to a conventional engine, this prechamber is characterized by high initial combustion rates. While the rates might be reduced somewhat by enlarging the passage area, the pressure rise rate will always be considerably higher than that of a conventional combustion chamber (Ref. 4-60). As a result, retarded ignition timing can be employed. In most of the tests conducted to date on this engine, the ignition timing was adjusted to 5 deg BTDC.

The power output of the engine is varied by means of a conventional air throttling valve combined with simultaneous control of the fuel injection rate.

4.3.7.2 Emission Characteristics

In the Newhall concept, achievement of low NO_x emissions is attributed to the mixing and quenching effects which occur as the combustion products enter the main chamber containing low temperature air. The fuel rich mixture in the prechamber yields low NO_x concentrations and while more oxygen is provided when the gases enter the main chamber, the simultaneous quenching of the combustion products with cool air minimizes the formation of additional NO. Another factor which inhibits the formation of NO is the relatively large quantity of residual exhaust remaining in the prechamber which acts much in the same way as exhaust gas recirculation (Ref. 4-60). While the gas temperature in the main chamber is low enough after mixing, it is sufficiently high to assure rapid oxidation of CO and HC in the presence of oxygen.

All the emission data currently available were obtained on a single-cylinder Waukesha CFR engine which was equipped with a Newhall prechamber. The operating conditions of the engine are listed in Table 4-32 (Ref. 4-60).

The emission characteristics of this engine were measured at constant speed, using a flame ionization detector for HC, an ultraviolet spectroscopic technique for NO_x, and thermal conductivity gas chromatography for CO, CO_2, O_2, and N_2. Early tests shown in Ref. 4-60 indicate little change in HC and CO with ignition timing variation, while NO_x increased as timing was advanced. At 5-deg BTDC ignition timing and an overall air-fuel equivalence ratio of 0.7, the NO_x emission was approximately 0.9 g/ihp-hr while CO and HC were 30 g/ihp-hr and 0.53 g/ihp-hr, respectively.

Table 4-32. OPERATING CONDITIONS OF THE CFR/NEWHALL PRECHAMBER ENGINE (Ref. 4-60)

Cylinder displacement, cubic inches	37.33
Compression ratio	8.0
Prechamber volume ratio	0.65
Start of fuel injection timing, deg BTDC	110
Fuel injector opening pressure, psia	500
Engine speed, rpm	1600 (nominal)
Coolant temperature, °F	160 (nominal)
Fuel	Isooctane
Throttle setting	Wide open

Tests conducted on the same engine using an improved injector to avoid spray impingement on the combustion chamber walls have shown substantial improvements in the HC and CO emissions. These results are shown in Figure 4-51, indicating that HC and CO were reduced by more than on order of magnitude relative to the earlier tests (Ref. 4-59).

Comparison of the NO_x emission of a conventional uncontrolled engine with the Newhall prechamber engine is shown in Figure 4-52 for two different prechamber volumes (Ref. 4-61). As indicated, very low NO_x levels were achieved for a prechamber volume of approximately 50 percent of the clearance volume. In these tests, the engines were operated at full-throttle, which represents the most severe condition for NO_x. At part-throttle, NO_x dropped to about half of the full-throttle value, while little change was observed in HC and CO.

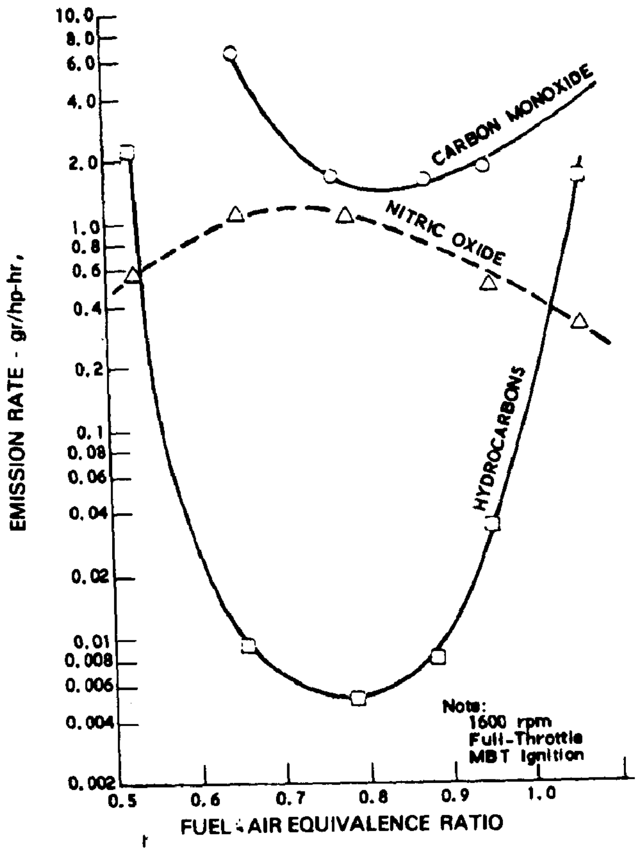


Figure 4-51. Preliminary emission data; Newhall prechamber engine (Ref. 4-59)

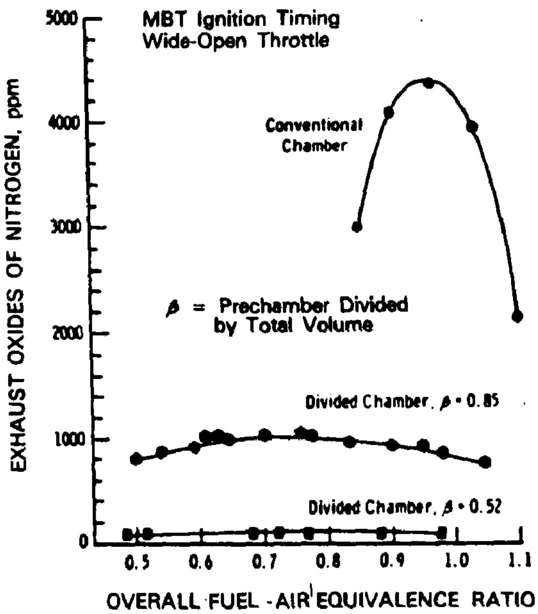


Figure 4-52. NO_x emission for Newhall prechamber engine and conventional engine (Ref. 4-61)

4.3.7.3 Fuel Consumption Characteristics

Indicated specific fuel consumption data from the Newhall single-cylinder prechamber engine, are shown in Figure 4-53 as a function of the overall equivalence ratio (Ref. 4-60). For equivalence ratios below 0.7, the measured fuel consumption is comparable to that of conventional spark-ignition engines. However, the fuel economy deteriorates at higher equivalence ratios due to late combustion in the main chamber.

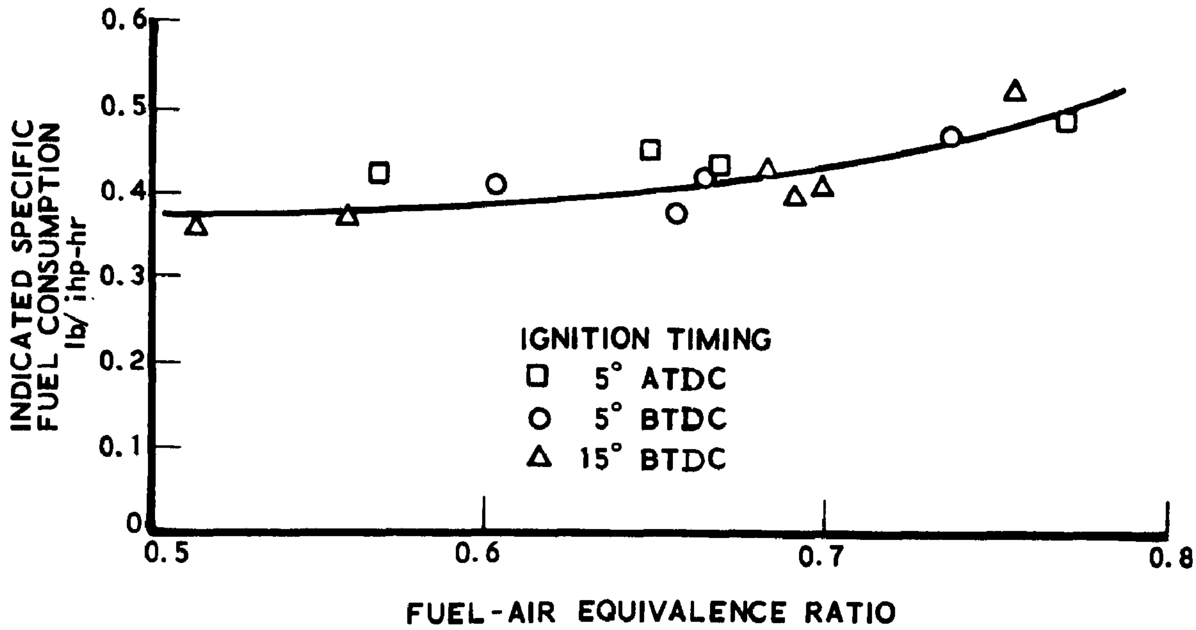


Figure 4-53. Indicated specific fuel consumption versus fuel-air equivalence ratio; Newhall prechamber engine concept (Ref. 4-60)

4.3.7.4 Engine Performance Characteristics

While the development work conducted to date on the Newhall prechamber has been confined to a single-cylinder engine, some observations made during these tests would apply to full-scale engines as well.

Comparison of the indicated mean effective pressure of the prechamber engine and the same engine equipped with a conventional chamber showed a 20 to 25 percent loss in maximum power which was attributed to lean mixture operation and incomplete scavenging of the prechamber. This is an undesirable feature of the concept that could be of particular concern for small vehicles which have generally little or no power reserve.

The operation of the prechamber was characterized by engine roughness resulting from high combustion pressure rise rates. While an increase in the prechamber to main chamber passage area combined with retarded ignition timing alleviated this problem to some degree, the observed pressure rise rate remained higher than that of conventional engines (Ref. 4-60).

The prechamber engine is relatively insensitive to fuel octane number because of late fuel injection and the rapid rate of burning. Therefore, the end gas is exposed only for a very short time to conditions which would promote autoignition and detonation. This was verified by using various isooctane/heptane mixtures as fuel. In all cases, no noticeable changes were observed in the engine pressure traces (Ref. 4-60).

4.3.7.5 Potential Problem Areas

The combustion roughness observed in the engine is a potential problem area. However, improvements in the combustion characteristics might be achieved by optimizing the size of the prechamber passage. Because of the observed loss of about 20 to 25 percent in the power output capability of the engine the displacement would have to be increased by that amount to achieve comparable performance in a vehicle.

Careful matching of the fuel spray and prechamber geometry would be essential to prevent fuel impingement on the prechamber walls and to maintain low HC and CO levels over the whole spectrum of engine operating conditions.

4.3.7.6 Current and Projected Status

Experimental work on the Newhall prechamber concept which had been under way at the University of Wisconsin for a number of years was terminated in 1971. While incorporation of the concept into new engine designs would be possible, application as a retrofit device for in-use vehicles is not considered to be feasible.

4.3.8 Russian Prechamber Concepts

Prechamber engine research is being conducted in the Soviet Union under the sponsorship of the USSR Academy of Science and the Central Scientific Research Institute for Automobiles and Automobile Engines, with some participation by the automotive industry. Pertinent results published by a number of Russian investigators are briefly discussed in the following subsections.

4.3.8.1 Sokolik and Karpov

In an effort to extend the lean limit of spark-ignition engines, Sokolik and coworkers have studied a number of prechamber engine configurations using prechamber volumes in the range of 2 to 3 percent of the total clearance volume (Ref. 4-62). As part of this program, single-cylinder engine tests were conducted using a dual-carburetion system to provide a rich mixture to the prechamber and a lean mixture to the main chamber. In addition, the ignition and combustion processes occurring in lean mixture engines were studied by means of a special bomb arrangement. Based on these tests, the following conclusions have been reached.

1. The ignition delay in the main chamber decreases with decreasing pressure drop across the communicating passage and with prechamber enrichment. Operation with long ignition delays is cause for rough combustion.

2. Incorporation of a small prechamber has resulted in an extension of the lean limit of the engine to fuel-air equivalence ratios of about 0.5.
3. The combustion duration in the prechamber engine was reduced by about 50 percent relative to a standard engine and was only slightly affected by variations in spark timing.
4. The octane requirement of the prechamber engines was lower than that of the standard engine because of the shorter time available for preheating and detonation of the end gas. Fuel octane numbers of 60 to 70 appeared to be adequate for engine compression ratios of about 7.
5. At part load, some throttling is required to achieve optimum fuel economy.
6. Relative to conventional engines, the fuel economy of the prechamber engine is somewhat better, accompanied by a reduction in HC and CO.

In another program, main chamber fuel injection was utilized in conjunction with injection of a prevaporized rich air-fuel mixture into the prechamber. In these tests, the fuel injection process was terminated before top dead-center and the spark was timed in such a manner that ignition of the main chamber occurred immediately after completion of main chamber injection.

4.3.8.2 Kobaidze

The work performed by this investigator was concerned with the study of prechamber engine combustion processes (Ref. 4-63). In this program, gas samples were taken from the prechamber and main chamber at 1.6 to 2-deg crank angle intervals to determine instantaneous CO, CO_2, and O_2 concentrations. The rate of oxygen depletion was used as a measure of the combustion rate. Flame velocities were measured by means of ionization gaps fitted into the prechamber and main chamber.

Based on test data, Kobaidze concluded that combustion in the prechamber proceeds vigorously even with rich air-fuel ratios of the order of 8. In this case, the flame velocity varied between 46 and

85 fps, indicating turbulent combustion in the prechamber. In the range of main chamber air-fuel ratios between 15 and 19.5, the flame velocity in the main chamber was 2 to 2.4 times the flame velocity obtained in conventional spark-ignition engines. This observation agrees with the results reported by Sokolik (Ref. 4-62).

4.3.8.3 Nilov

The Nilov prechamber design which followed the Russian practice of using a prechamber volume of only a few percent of the clearance volume, employs a third valve and dual carburetion, as shown in Figure 4-54 (Ref. 4-64).

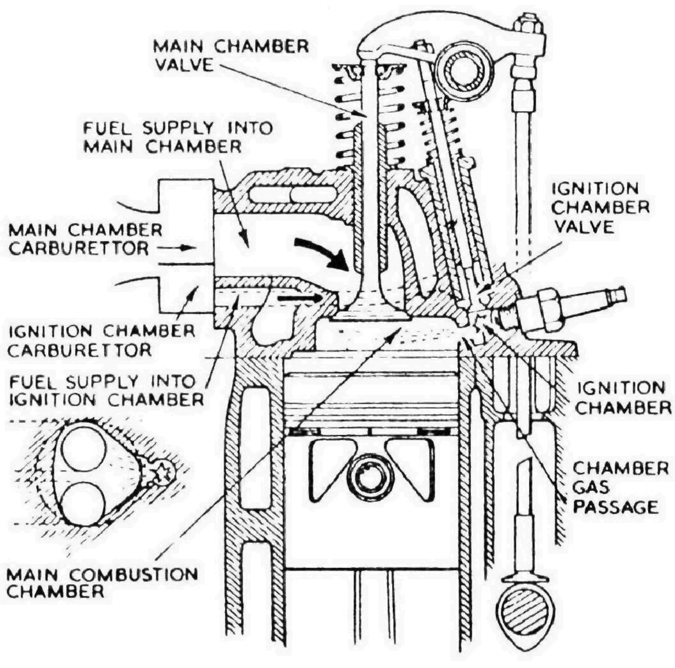


Figure 4-54. Nilov prechamber engine (Ref. 4-64)

Like other designs, the prechamber operates fuel-rich while the main chamber is fueled with a lean mixture. Rapid combustion reduces the fuel octane requirement of the engine relative to conventional engines, permitting the use of higher engine compression ratios. While no quantitative emission data were reported by Nilov, CO and the other products of incomplete combustion were substantially reduced (Ref. 4-64). The fuel economy of prechamber engine equipped vehicles was about 10 to 15 percent better than that of comparable conventional engine/vehicle configurations.

4.3.8.4 Gussak

Conceptually, the Gussak prechamber illustrated in Figure 4-55 (Ref. 4-65) is similar to the previously discussed Russian prechamber concepts. It has a prechamber volume of 2 to 3 percent of the clearance volume and the prechamber is designed for minimum surface-to-volume ratio. The prechamber and main chamber are fueled by two separate carburetors. The air-fuel ratio of the rich prechamber

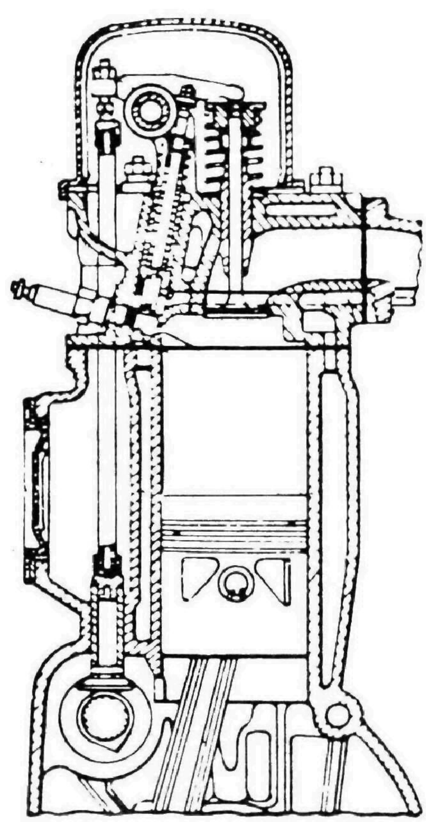


Figure 4-55. Gussak carburetor-type prechamber engine (Ref. 4-65)

mixture varies between about 6 and 10, while the main chamber is always lean, using air-fuel ratios up to 30 (Refs. 4-21 and 4-65).

Main chamber ignition is accomplished by the jet of chemically reactive products emanating from the prechamber. These products include peroxides, aldehydes, and highly reactive free radicals such as CH_2, CH, and hydrogen atoms, along with other unstable constituents. With jet ignition, the combustion rate in the main chamber was increased by a factor of 3 to 4 relative to standard spark-ignition engines. The engine was operated successfully with overall air-fuel ratios as high as 30. The octane requirement of the prechamber engine was somewhat lower than for conventional engines permitting an increase in engine compression ratio of about 0.8 units.

The specific fuel consumption of the prechamber engine varied between 190 gr/bhp-hr and 200 gr/bhp-hr which corresponds to a 20 to 30 percent improvement over the standard engine.

According to Ref. 4-65, the toxic exhaust species were almost completely eliminated in the prechamber engine. In particular, CO and benzopyrene were reduced by an order of magnitude.

4.4 OTHER CONCEPTS

Brief descriptions of a number of additional prechamber engine concepts are presented in Table 4-33. These include the configurations patented by Lange, Sakai, Jozlin, Mozokhin, Stumpfig, Freeman, Meyer, Barnes, Summers, Ricardo, Bishop, May, Schlamann, and Kerimov. Additional patent disclosures are listed in Appendix A.

Table 4-33. OTHER PRECHAMBER SPARK-IGNITION ENGINE CONCEPTS

Inventor	U.S. Patent No. and date	System description
K. Lange, and D. Gruden (Porsche) (Ref. 4-66)		This concept consists of a small unscavenged prechamber containing the spark plug which is surrounded by a separate, very small ignition cell. A small amount of the total fuel is injected into the prechamber, while the remainder is injected into the main chamber intake manifold. The engine has been successfully operated with overall air-fuel ratios up to 32. Relative to the baseline engine, NO_x of the prechamber engine was reduced by 80 to 90 percent, while fuel consumption, HC, and CO remained unchanged. Similar trends were obtained with methanol.
Yasuo Sakai, et al (Nissan Motor Co.) (Refs. 4-67 and 4-68)		This concept uses a small prechamber (about 10 percent of the clearance volume) equipped with a separate intake valve and spark plug. A rich prechamber mixture is supplied by means of a carburetor, while a separate carburetor supplies a lean mixture to the main chamber. Single-cylinder tests indicated low emissions without a loss in fuel economy.
J. A. Jozlin	3,710,764 16 Jan 74	A small prechamber equipped with a spark plug and a check valve is installed in the spark plug hole. The check valve allows free entry of air-fuel mixture from the main chamber during the compression stroke, but prevents any outflow except through a set of small orfices in the check valve. Upon spark ignition, a "jet flame" emanating from the prechamber ignites the mixture in the main chamber assuring complete combustion and operation with low grade fuels.

Table 4-33. OTHER PRECHAMBER SPARK-IGNITION ENGINE CONCEPTS (Continued)

Inventor	U.S. Patent No. and date	System description
N. G. Mozokhin, et al	3,682,146 8 Aug 72	This system consists of a small precombustion chamber incorporating a spark plug and fuel injector. Fuel is supplied to the prechamber (rich mixture) and to the main combustion chamber (lean mixture) by jet pumps which use some of the compression air via a special control valve. This gas bleed effectively varies the engine compression ratio, reducing it at high loads and permitting operation with low octane fuels.
F. Stumpfig	3,661,125 9 May 72	The small prechamber incorporating a spark plug and fuel injector is installed in the spark plug opening of the engine. Fuel is injected into the prechamber at the end of the exhaust stroke. During the compression stroke, air or lean air-fuel mixture is pushed from the main combustion chamber into the prechamber. The elongated cylindrical prechamber with hot walls promotes evaporation of the injected fuel, thus providing an ignitable air-fuel mixture. Hot gases expand from the prechamber into the main chamber and ignite the air-fuel mixture in the main chamber. This device can be operated with two or four-stroke engines using gasoline or diesel fuel.
J. H. Freeman W. D. Dysart	3,207,141 21 Sep 65	In this design, the prechamber volume is 15 to 35 percent of the total clearance volume. The concept uses an elongated main chamber placed in line with the prechamber passage to ensure rapid combustion of the charge and to improve the knock characteristics of the engine.

Table 4-33. OTHER PRECHAMBER SPARK-IGNITION ENGINE CONCEPTS (Continued)

Inventor	U.S. Patent No. and date	System description
J. N. Bishop	3,195,519 20 Jul 65	A small prechamber is placed inside the main combustion chamber formed by a cavity in the piston head and separated from the main chamber by a wall with one or more transfer passages. The spark plug is fitted in the center of the prechamber. Fuel is injected into the main chamber close to the prechamber passage during the compression stroke and the motion of the piston sweeps fuel-rich mixture into the prechamber while the main chamber remains lean. The main chamber air supply is unthrottled.
J. T. M. Schlamann	2,758,576 14 Aug 56	The prechamber volume varies between 15 to 35 percent of the total clearance volume. Fuel is injected into the prechamber during the compression stroke to form a rich mixture. The main chamber receives a carbureted lean mixture and pure air at idle. The prechamber air is supplied from the main chamber during the compression stroke. The engine load is controlled by varying the main combustion chamber mixture. Air throttling is not required for this engine.
N. A. Kerimov R. I. Mekhtiyev (Ref. 4-69)		This design utilizes a small prechamber equipped with a spark plug and a separate air intake valve. The prechamber fuel is supplied by an injector forming a fuel-rich mixture. A separate injector supplies fuel to the main combustion chamber. The amount of fuel injected is varied as a function of load.

Table 4-33. OTHER PRECHAMBER SPARK-IGNITION ENGINE CONCEPTS (Continued)

Inventor	U.S. Patent No. and date	System description
W. E. Meyer	2,735,413 21 Feb 56	This prechamber is intended for use in two-stroke engines. The design has a circular geometry and tangential passage to the main chamber to impart a strong swirl motion to the air during the compression stroke. The spark plug is located in the prechamber periphery. All fuel is injected into the prechamber during the compression stroke, and operation at lean overall mixtures is postulated.
W. B. Barnes	683,162[a] 26 Oct 52	This engine incorporates a very large prechamber relative to the total clearance volume. All fuel is injected into the prechamber intake manifold while pure air is drawn into the main chamber. The prechamber is equipped with a cam-actuated intake valve.
C. E. Summers	1,568,638 5 Jan 26	This engine, which was tested by General Motors in the 1920s, has a small spherical prechamber which contains the spark plug and a separate cam-actuated intake valve. Two carburetors are employed to supply a rich mixture (A/F = 3.5) to the prechamber and a lean mixture (A/F = 40) to the main chamber. The engine is designed for high swirl in the prechamber.

[a]U.K. Patent

Table 4-33. OTHER PRECHAMBER SPARK-IGNITION ENGINE CONCEPTS (Continued)

Inventor	U.S. Patent No. and date	System description
H. R. Ricardo	1,271,942 9 Jul 1918	The prechamber which utilizes a pressure actuated intake valve is charged with a very rich mixture, while a lean mixture or pure air is supplied to the main chamber. A converging-diverging nozzle is utilized as the communicating passage between the prechamber and main chamber. Ignition is accomplished by means of a conventional spark plug located in the prechamber.

4.5 STATIONARY ENGINE MANUFACTURERS

4.5.1 Colt Industries

The Fairbanks Morse Engine Division of Colt Industries, Beloit, Wisconsin, is a major manufacturer of stationary diesel and spark-ignition engines. In the early 1950s, the company adopted the prechamber concept to its line of multicylinder spark-ignition stationary gas engines primarily because of the superior ignition and combustion characteristics of prechamber engines relative to open chamber configurations (Ref. 4-70).

4.5.1.1 Engine Description

The prechamber spark-ignition gas engine design currently marketed by Fairbanks Morse is illustrated in Figure 4-56 showing two prechambers or ignition cells per cylinder (Ref. 4-71). This particular engine is of the blower-scavenged, two-stroke type and utilizes opposed pistons. Its design brake mean effective pressure is 96 psi (compression ratio 9.75:1) and the operating speed varies between 500 and 900 rpm. Most of the fuel required for engine operation is admitted to the cylinders through a gas valve, while the remainder of the fuel is injected into the prechambers. The fuel quantity entering the cyliners is controlled by the header pressure which is regulated by means of a throttle valve as a function of engine speed.

The prechamber design utilized by Fairbanks Morse is depicted in Figure 4-57 (Ref. 4-71). Each cell is water-cooled and consists of a small combustion volume occupying about 1.5 percent of the total clearance volume, a conventional spark plug, a gas injection system, and a small communicating flow passage at the prechamber exit. The principal design objectives of the prechamber are (1) assuring positive ignition of the main chamber charge over wide ranges of engine

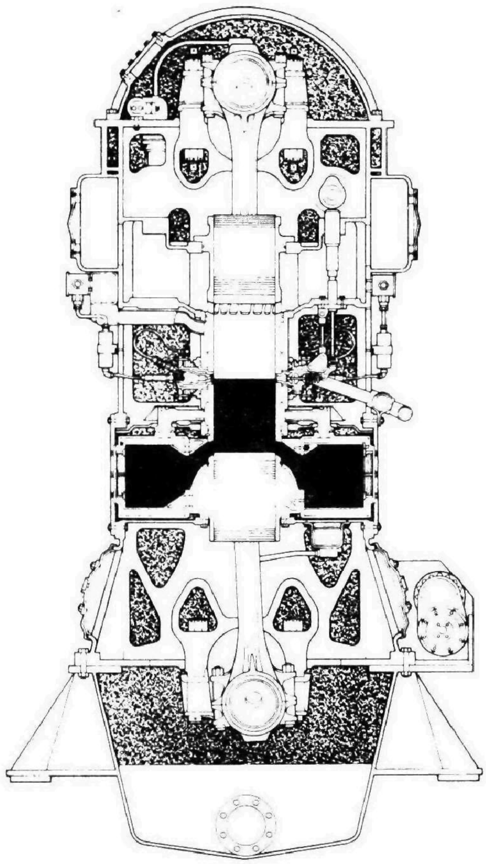


Figure 4-56. Fairbanks Morse opposed piston, two-stroke prechamber stationary spark-ignition gas engine (Ref. 4-71)

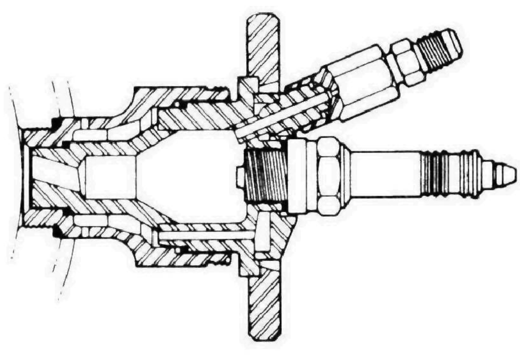


Figure 4-57. Fairbanks Morse prechamber design (Ref. 4-71)

load and air-fuel ratios and (2) alleviating the detonation problem which is frequently encountered in open chamber gas engines at the minimum bsfc point. The fuel flow rate injected into the prechambers is closely regulated by means of a check valve and orifice arrangement, providing a near-stoichiometric air-fuel mixture in the region of the spark plug. Unlike the prechamber systems currently under consideration by the automotive industry, the Fairbanks Morse concept is designed for operation without a separate prechamber intake valve. In this particular system, the air required for combustion in the prechambers is provided through backflow from the cylinder during the compression stroke of the pistons.

4.5.1.2 Emission and Specific Fuel Consumption Characteristics

HC, CO, and NO_x mass emission data obtained by Fairbanks Morse on a single test of its 4-cylinder 38DS8-1/8 prechamber engine are presented in Figure 4-58 as a function of brake mean effective pressure and engine speed (Ref. 4-70). As indicated, the emissions are nearly independent of engine speed, except for NO_x which increases moderately with increasing speed. Both total HC and net HC (excluding methane) are shown. As indicated, methane is the dominant HC exhaust emission species, which is not surprising considering that the engine was operated on natural gas. Of particular interest is the fact that NO_x increases very slowly up to brake mean effective pressures of about 75 psi. However, beyond that point, NO_x increases quite rapidly.

For comparison, emission data obtained by Cooper-Bessemer on its blower-scavenged GMVA-8 two-stroke open chamber gas engine, operated at 300 rpm are shown in Figure 4-59 (Refs. 4-72 and 4-73). The NO_x emissions of this engine are substantially higher than those of the prechamber engine shown in Figure 4-58, except at

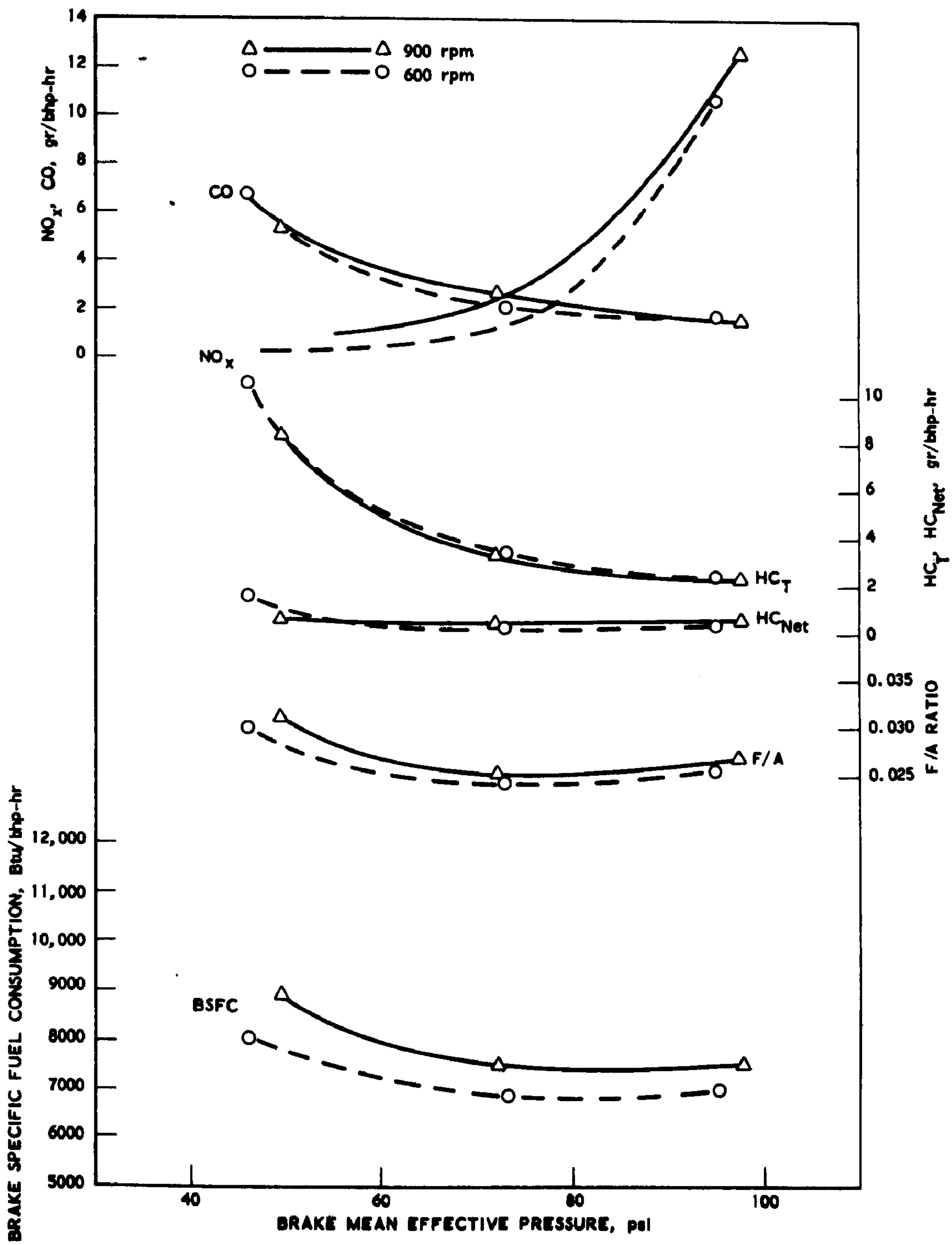


Figure 4-58. Emissions, fuel-air ratio and brake specific fuel consumption versus brake mean effective pressure; Fairbanks Morse prechamber engine 38DS8-1/8 (Ref. 4-70)

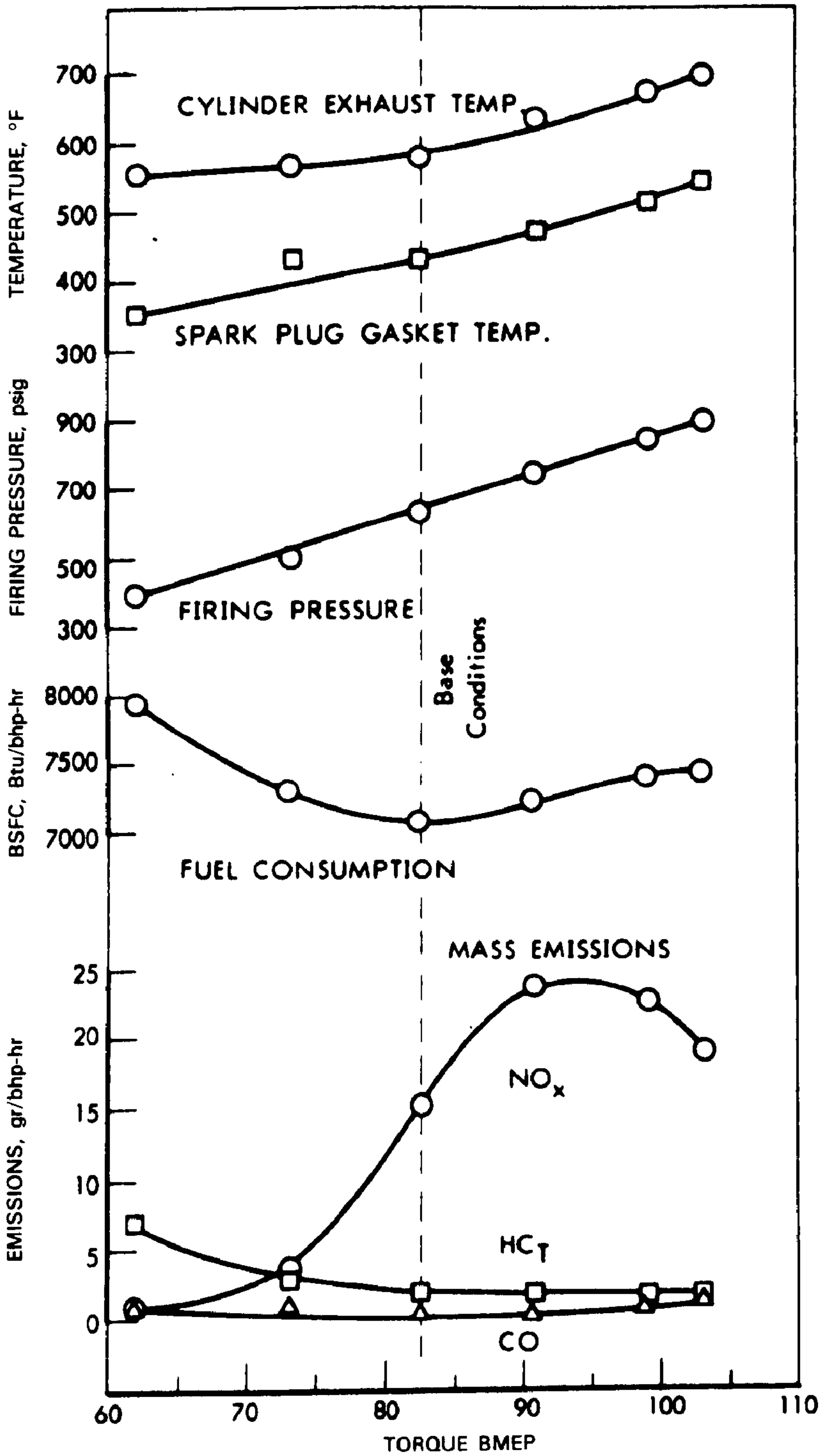


Figure 4-59. Effect of load at constant speed on emissions and performance; Cooper-Bessemer two-stroke spark gas engine; base conditions, 300 rpm (Refs. 4-72 and 4-73)

low loads where the difference in NO_x between the two engines tends to be smaller. For example, for a brake mean effective pressure of 90 psi, the NO_x emission of the prechamber engine is about 8 gr/bhp-hr versus 23 gr/bhp-hr for the open chamber engine.

The brake specific fuel consumption of the two engines is also presented in Figures 4-58 and 4-59, indicating lower values for the prechamber engine, particularly at high loads. This trend is attributed to the fact that prechamber engines are less prone to detonation, permitting engine operation at the optimum air-fuel ratio, although other design differences between the two engine types may contribute to the observed differences in specific fuel consumption. Conversely, open chamber engines are generally operated at air-fuel ratios higher than optimum to prevent the occurrence of detonation in the combustion chamber under all ambient air and engine operating conditions. It should be noted, however, that the above emission and fuel consumption comparisons are based on a very limited data sample, and may not adequately reflect the performance characteristics of other stationary engines.

4.5.1.3 Materials and Manufacturing

The prechambers utilized by Fairbanks Morse are fabricated from cold rolled steel, using hydrogen brazing techniques. Since the prechambers are water-cooled, the use of high-temperature nickel alloys has not been required to achieve the desired system reliability and durability.

4.5.1.4 Potential Problem Areas

To date, no problems have been encountered on the Fairbanks Morse prechamber engine (Ref. 4-70). The temperature of the prechamber walls is sufficiently low to assure reliable operation at all operating conditions. Also, surface corrosion of the communicating passage, which has been observed in automotive prechamber engines, has not occurred in the Fairbanks Morse engine.

While retrofitting of stationary engines might be possible in principle, Fairbanks Morse feels that this approach could not be justified for other engine types, such as those utilizing cylinder heads and intake and exhaust valves, because of the high cost of modification and relatively small number of engines in the field. Moreover, it is conceivable that different prechamber designs and control systems would have to be developed for the various engines, considering the variations in important engine design parameters, including the cylinder head and piston design, the degree of scavenging, the magnitude of combustion chamber swirl, and the fuel type used.

4.5.1.5 Current and Projected Status

The Fairbanks Morse prechamber spark-ignition engine which has been in production for many years has lower NO_x emissions than open chamber engines and exhibits good efficiency and durability characteristics.

While a limited amount of development work has been conducted by Fairbanks Morse in the areas of prechamber geometry optimization and mixture control refinement, they feel that substantial additional efforts would be required to achieve further emission reduction consistent with good efficiency and combustion stability (Ref. 4-70).

Future efforts on the engine will be concerned with the verification of the initial emission and fuel consumption data shown in Figure 4-58 and extension of the data bank in the high-load regime.

4.5.2 Other Developments

While Fairbanks Morse is the only manufacturer of stationary prechamber spark-ignition engines in the United States, a number of other manufacturers have conducted a limited amount of theoretical and/or experimental work related to prechamber concepts. These include Cooper-Bessemer, Ingersoll-Rand, and Worthington-CEI.

Laboratory tests conducted by Cooper-Bessemer indicate that more reliable ignition of lean mixtures was achieved with prechambers relative to open chamber configurations. Also, the emissions of the prechamber engine were reduced from the open chamber levels. Apparently, development efforts are continuing in that area (Ref. 4-74).

Approximately six years ago, Ingersoll-Rand was involved in a limited prechamber ignition test program, using a single-cylinder engine which was fitted with a prechamber occupying about 10 percent of the total clearance volume. With prechambers, the lean limit of the engine could be extended and the occurrence of detonation was minimized. While insufficient testing was done to establish engine efficiency, Ingersoll-Rand feels that the inherent increase of the heat losses due to the addition of the prechamber might negate any potential performance gain that might result from operation with leaner mixtures (Ref. 4-75).

Worthington feels that the development of a stationary prechamber engine would be too involved and uncertain to justify the required capital expenditure at this time (Ref. 4-76).

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SECTION 5

PRECHAMBER ENGINE EVALUATION

This section of the report presents a summarization and evaluation of the automotive and stationary spark-ignition prechamber engine concepts/systems identified in Section 4, relative to their applicability to new engine designs and retrofit installations. Subsection 5.1 is concerned with automotive applications, addressing such issues as prechamber engine classification, operating characteristics, emission and fuel consumption characteristics, odor, aldehyde, smoke, noise, engine durability and maintenance requirements, engine and vehicle performance, vehicle driveability, fuel requirements, concept assessment, and economics. Subsection 5.2 examines the applicability of prechambers to new and in-use heavy-duty stationary engines.

5.1 AUTOMOTIVE ENGINES

5.1.1 Prechamber Engine Classification

The principal design goal of all prechamber spark-ignition engines devised to date is the achievement of good thermal efficiency combined with low exhaust emissions. In most systems, the prechamber containing the spark plug is supplied with a rich air-fuel mixture, while a lean mixture or even pure air is inducted into the main chamber. Upon ignition of the rich mixture surrounding the spark plug, the pressure in the prechamber rises very rapidly, forcing a highly reactive jet of hot combustion products into the main chamber, where the combustion process is then completed.

To achieve the common objective, the various inventors and investigators of prechamber engine concepts have pursued different design approaches with respect to prechamber size and the type of prechamber and main chamber air and fuel supply systems employed. While any of these distinguishing features could be utilized as the principal classification parameter, the ratio of the prechamber volume to the total clearance volume has been selected to characterize the various prechamber engine configurations considered in this study. Three engine classes were then established - small prechambers, medium-size prechambers, and large prechambers. The small prechambers have prechamber volume ratios below about 8 percent, while the medium-size and large-size prechambers employ volume ratios of about 8 percent to 30 percent, and above 30 percent, respectively.

Pertinent design features of a number of small prechamber engine developments are listed in Table 5-1. As indicated, the majority of these engines utilize scavenged prechambers incorporating cam-actuated intake valves or check valves plus carburetion for both the prechamber and main chamber. Conversely, the Ford and Cornell concepts incorporate unscavenged prechambers plus carburetion, while Volkswagen combines unscavenged prechambers and fuel injection. Research and development work is continuing on most of the concepts listed in Table 5-1.

The medium-size prechamber engine class is summarized in Table 5-2. With the exception of the unscavenged prechamber configurations by Volkswagen and Thermo Electron Corporation, these engines incorporate a separate prechamber intake manifold and a cam-actuated third valve. While carburetion is utilized in most of these engines, a number of manufacturers have experimented with prechamber fuel injection, including Ford Motor Company, Volkswagen,

Table 5-1. PRECHAMBER ENGINE DESIGN CHARACTERISTICS - SMALL PRECHAMBERS

Organization	Prechamber volume ratio,[a] %	Prechamber air supply	Method of fuel supply		Test engine	Test vehicle	Development status
			Prechamber	Main chamber			
Ford Motor Company (torch ignition engine)	Small	From main chamber	From main chamber	Carburetion	Single-cylinder 351 CID, V-8	1972 Gran Torino	Early development
Volkswagen A.G.	5	From main chamber	Fuel injection	Fuel injection	Single-cylinder	None	Early development
Combustion Control (Morghen concept)	3	Intake manifold and check valve	Carburetion	Carburetion	Single-cylinder multicylinder	1962 Falcon M-151 vehicle	In development
Phillips Petroleum Co.	2	Intake manifold and check valve	Premixed	Premixed	Modified CFR engine	-	Dormant
Teledyne Continental Motors	3	Intake manifold and check valve	Carburetion	Carburetion	L-141	-	In development
Walker Manufacturing Co. (Stratofire engine)	3	Intake manifold and check valve	Carburetion	Carburetion	1960 G.M. Corvair	1960 G.M. Corvair	Dormant
California State University at Sacramento	3	Intake manifold and check valve	Carburetion	Carburetion	1962 Ford Falcon 1964 Ford Falcon	1962 Ford Falcon 1964 Ford Falcon	Research continuing
Cornell University (Cornell spark plug)	0.5 - 1.0	From main chamber	From main chamber	Carburetion	232 CID, 6-cyl. American Mtrs.	1971 American Mtrs. Matador	Research continuing
USSR developments	2-3	Intake manifold and cam-actuated third valve	Carburetion	Carburetion or fuel injection	Single-cylinder		In development

[a]Prechamber volume divided by total clearance volume.

Table 5-2. PRECHAMBER ENGINE DESIGN CHARACTERISTICS - MEDIUM-SIZE PRECHAMBERS

Organization	Prechamber volume ratio,[a] %	Prechamber air supply	Method of fuel supply		Test engine	Test vehicle	Development status
			Prechamber	Main chamber			
Ford Motor Company (three-valve, carbureted)	8.9	Intake manifold and cam-actuated third valve	Carburetion	Carburetion	400 CID, V-8	1973 Ford LTD	In development
Ford Motor Company (three-valve, fuel-injected)	12.0	Intake manifold and cam-actuated third valve	Fuel injection	Carburetion	140 CID, 4-cylinder	None	Unknown
General Motors Corp. (jet ignition stratified charge)	<10	Intake manifold and cam-actuated third valve	Carburetion	Carburetion	Single-cylinder, 140 CID, 4-cylinder 350 CID, V-8	350 CID engine/ vehicle configuration	In development
Honda Motor Company (CVCC)	~10	Intake manifold and cam-actuated third valve	Carburetion	Carburetion	119 CID, 4-cyl. 91 CID, 4-cyl. 140 CID, 4-cyl. 350 CID, V-8	1975 Honda Civic 1975 Honda Civic 1972 G.M. Vega 1973 G.M. Impala	Honda Civic in production
Nissan - Datsun (NVCC)	~10	Intake manifold and cam-actuated third valve	Carburetion	Carburetion	Unknown	Unknown	In development
Volkswagenwerk A. G.	~10	Intake manifold and cam-actuated third valve	Carburetion	Carburetion	Single-cylinder	None	Unknown
	25 - 30	From main chamber	Med. pressure fuel injection	Low pressure fuel injection	Single-cylinder 4-cylinder	VW Beetle	In development
Eaton Corporation	8 and 19	Intake manifold and cam-actuated valve	Carburetion	Carburetion	120 CID 4-cyl. Ford Pinto	None	In development
Thermo Electron Corp.	15	From main chamber	Fuel injection or carburetion	Fuel injection or carburetion	Modified CFR Saab 2-stroke Modified L-141	Saab Ford Capri	In development

[a] Prechamber volume divided by clearance volume

Table 5-2. PRECHAMBER ENGINE DESIGN CHARACTERISTICS - MEDIUM-SIZE PRECHAMBERS (Continued)

Organization	Prechamber volume ratio,[a] %	Prechamber air supply	Method of fuel supply		Test engine	Test vehicle	Development status
			Prechamber	Main chamber			
The Aerospace Corporation	8	Intake manifold and flapper valve	Propane	Carburetion	Single-cylinder Wisconsin model AGND	None	Dormant
Stanford University (Heintz Ram Straticharge)	18.6	Intake manifold and cam-actuated third valve	Fuel injection	Fuel injection	1957, 392 CID, V-8 Chrysler	None	Dormant
	15	Intake manifold and cam-actuated third valve	Carburetion	Carburetion	1974 American Motors 232 CID 6-cylinder	None	Research continuing
	-	Intake manifold and cam-actuated intake valve	Fuel injection into prechamber intake manifold	From prechamber	2-stroke, 198.6 CID, 4-cylinder; Roots blower	None	Dormant
University of Rochester (Broderson concept)	20 - 30	Intake manifold and cam-actuated third valve	Fuel injection	From prechamber or carburetion	Various single-cylinder engines	None	Dormant

[a]Prechamber volume divided by total clearance volume

Thermo Electron Corporation, and the University of Rochester. As shown in the table, a considerable amount of vehicle testing has been conducted by a number of manufacturers, particularly the Honda Motor Company, whose CVCC engine has been in production since December 1973. Development of the other engines is in progress except for the Aerospace Corporation and University of Rochester concepts, which are currently not being pursued.

Design information from three large prechamber engine configurations is presented in Table 5-3. In these engines, air is forced into the prechamber from the main chamber during the compression stroke. In the Ford and University of Wisconsin designs, all fuel is injected into the prechamber, while the University of California concept utilizes fuel injection into the prechamber combined with main chamber carburetion. Except for a limited amount of multicylinder engine testing conducted by Ford Motor Company, research and development work on large prechamber configurations has been restricted to single-cylinder engines. Work in this area is continuing at Ford and at the University of California at Berkeley.

5.1.2 Operating Characteristics

The two-stage combustion experiments conducted by a number of investigators indicate the formation of multiple ignition zones in the lean main chamber mixture. As a result, the flame speed is increased by about 100 to 200 percent relative to conventional spark-ignition engines. In addition, a 50 percent reduction in the ignition lag has been reported by Ford Motor Company. Therefore, prechamber engines can be operated efficiently with retarded spark timing, providing a potential of further NO_x reduction.

The air-fuel ratio of the mixture inducted into the prechambers varies between about 3 and 10, depending upon the type of fuel supply system utilized. The mixture is then further diluted by the lean

Table 5-3. PRECHAMBER ENGINE DESIGN CHARACTERISTICS - LARGE PRECHAMBERS

Organization	Prechamber volume ratio,[a] %	Prechamber air supply	Method of fuel supply		Test engine	Test vehicle	Development status
			Prechamber	Main chamber			
Ford Motor Company	Large	From main chamber	Fuel injection	From pre-chamber	Single-cylinder 400 CID, V-8	None	In development
University of California at Berkeley	38	From main chamber	Fuel injection	Carburetion	Modified CFR	None	Research continues
University of Wisconsin (Newhall concept)	65	From main chamber	Fuel injection	From pre-chamber	Modified CFR	None	Dormant

[a]Prechamber volume divided by total clearance volume

mixture or pure air flow entering the prechamber during the compression stroke.

In principle, many prechambers are capable of stable operation in the ultra-lean mixture regime. While unthrottled operation would be desirable from an efficiency point of view, most of the prechamber engines currently under consideration have been limited to overall air-fuel ratios of about 20 to 24 to minimize the quench effects and the associated increase in HC and CO normally encountered with ultra-lean mixture operation.

To date, several investigators have made concerted efforts to improve the combustion process in prechamber engines, and further emission and engine efficiency improvements are projected by optimizing the critical prechamber design and operating parameters. These include the size and shape of the prechamber and communicating passage, the prechamber and main chamber air-fuel ratios, and the flow turbulence levels in the two chambers.

5.1.3 Emission and Fuel Consumption Characteristics

To provide a basis for the following discussion, selected performance data are presented in Tables 5-4 through 5-6 for automotive single-cylinder and multicylinder engine and vehicle configurations incorporating small prechambers, medium-size prechambers, and large prechambers, respectively.

5.1.3.1 Small Prechambers

5.1.3.1.1 Emissions

Based on the single-cylinder engine tests conducted by Phillips Petroleum Company and the multicylinder engine work performed by Teledyne Continental Motors, it is concluded that automotive spark-ignition engines equipped with small prechambers have the potential of achieving NO_x emission levels of about 1 gr/ihp-hr and CO

Table 5-4. SELECTED PRECHAMBER ENGINE PERFORMANCE TEST DATA - SMALL PRECHAMBERS

Organization	Engine test data							Vehicle test data							Potential problem areas
	Engine identification	Overall air-fuel ratio	Speed/imep[a]	Emissions, gr/ihp-hr HC	Emissions, gr/ihp-hr CO	Emissions, gr/ihp-hr NO_x	Specific fuel consumption, lb/ihp-hr	Vehicle identification	Test procedure	Emission control devices	Emissions, gr/mile HC	Emissions, gr/mile CO	Emissions, gr/mile NO_x	Fuel economy, mpg	
Ford Motor Company (torch ignition engine)	-	-	-	-	-	-	-	1972 Gran Torino	1975 FTP 4000-lb I. W.[b]	Modulated EGR air injection	0.87	7.23	1.36	11	Insufficient HC/CO control
Combustion Control (Morghen concept)	-	-	-	-	-	-	-	1962 Ford Falcon	7-mode, hot cycle	None	37% reduction	94% reduction	69% reduction	18% better at 50 mph	Prechamber manifold fuel condensation
Phillips Petroleum Corp.	Modified CFR engine	25	1000/70	2.0	2.5	1.0	0.43	-	-	None	-	-	-	-	High HC
Teledyne Continental Motors	L-141	-	1500/60[c]	High	Low	Low at low load	13% lower than standard engine[c]	-	-	None	-	-	-	-	Prechamber overheating; valve failure; high HC
Walker Manufacturing Company	-	-	-	-	-	-	-	1960 G. M. Corvair	Road test	None	Little improvement relative to baseline vehicle			Slightly better than baseline	High HC. Prechamber overheating
California State University at Sacramento	-	-	-	-	-	-	-	1962 Ford Falcon	1972 FTP	None	14.2	6.0	1.4	Slightly better than baseline	High HC
Cornell University (Cornell spark plug)	-	-	-	-	-	-	-	1971 American Motors Matador	1972 FTP	None	7.6	7.0	1.85	Similar to baseline	High HC. Plugging of holes in spark plug cover
USSR developments	Single-cylinder	<30	-	-	Very low	-	10-30% better than baseline	-	-	-	-	-	-	-	-

[a] Speed in rpm; imep in psi

[b] Chassis dynamometer inertia weight

[c] Brake specific values

Table 5-5. SELECTED PRECHAMBER ENGINE PERFORMANCE TEST DATA - MEDIUM-SIZE PRECHAMBERS

Organization	Engine test data							Vehicle test data							Potential problem areas
	Engine identification	Overall air-fuel ratio	Speed/imep[a]	Emissions, gr/ihp-hr HC	Emissions, gr/ihp-hr CO	Emissions, gr/ihp-hr NO_x	Specific fuel consumption, lb/ihp-hr	Vehicle identification	Test procedure	Emission control devices	Emissions, gr/mile HC	Emissions, gr/mile CO	Emissions, gr/mile NO_x	Fuel economy, mpg	
Ford Motor Company (three-valve, carbureted)	-	-	-	-	-	-	-	1973 Ford LTD	CVS-hot cycle 4500-lb I.W.[c]	Thermal reactor	0.43	5.7	1.6	9.1	Insufficient HC/CO control
Ford Motor Company (three-valve, fuel-injected)	-	-	-	4.0[b]	15.1[b]	1.4[b]	Same as standard engine	-	-	-	-	-	-	-	Prechamber fuel supply difficult to control
General Motors Corp. (jet ignition stratified charge)	50 CID single-cylinder	22	1600/70	1.8	2.7	2.2	-	350 CID engine/vehicle configuration	1975 FTP 5000-lb I.W.[c]	Thermal reactor early fuel vaporization	0.2-0.33	2.9-3.3	1.4-1.6	Same or slightly less than conventional engine	Mass production of dual carburetor; high raw HC/CO
Honda Motor Company (CVCC)	-	-	-	-	-	-	-	1975 Honda Civic (2-liter)	1975 FTP 2000-lb I.W.[c]	Thermal reactor intake air heating	0.18-0.24	1.75-2.12	0.65-1.03	19.4-22.1	Close carburetor manufacturing tolerances
	-	-	-	-	-	-	-	1975 Honda Civic (1.5 liter)	1975 FTP 2000-lb I.W.[c]	Thermal reactor	0.38-0.56	4.05-4.35	1.07-1.26	25.1-27.5	-
	-	-	-	-	-	-	-	1972 G.M. Vega	1975 FTP 2500-lb I.W.[c]	Thermal reactor	0.26	2.6	1.16	18.9	-
Nissan-Datsun (NVCC)	-	-	-	-	-	-	-	Unknown	1975 FTP 2750 I.W.[c]	Unknown	0.28-0.37	2.9-3.5	1.15-1.12	17.2-17.6	-
	-	-	-	-	-	-	-	1973 G.M. Impala	1975 FTP	Thermal reactor	0.32	2.8	1.68	11.5[d]	-
Volkswagenwerk A.G.	Single-cyl. 10% prechamber	-	-	High	High	Low	-	VW Beetle	1975 FTP 2250-lb I.W.[c]	None	2.1-2.4	4.4-8.0	0.75-0.96	22-26	High HC/CO requires after-treatment device
Thermo Electron Corp.	Modified CFR	-	-	-	-	-	10-20 percent lower than std. CFR	1967 Saab (2-stroke)	1975 FTP 2250-lb I.W.[c]	Catalyst	6.3	3.3	0.3	-	High HC
								1971 Ford Capri (modified L-141 engine)	1975 FTP	Modulated EGR, catalyst	1	1	0.7	18% better than baseline	High HC
Eaton Corporation	120 CID Ford Pinto; 8% prechambers	23	2500/78	0.29[e]	0.95[e]	0.61[e]	0.375	-	-	-	-	-	-	-	High raw HC
The Aerospace Corporation	Single-cyl. Wisconsin Model AGND	20-22	1800/-	35% reduction	Very low	60% reduction	Similar to baseline engine	-	-	-	-	-	-	-	Flapper valve durability
University of Rochester (Broderson concept)	Modified CFR, 22% prechamber	50	1200/50	-	-	-	0.31	-	-	-	-	-	-	-	-
	Modified L-141, 29% prechamber	40	1500/50	-	-	-	15 percent better than baseline	-	-	-	-	-	-	-	-
Stanford University (Heintz Ram Straticharge)	1957, 392 CID Chrysler	23	2000/56.9[d]	0.01[f]	0.018[f]	0.014[f]	At light loads, better than baseline	-	-	-	-	-	-	-	High HC, Prechamber overheating
	1974, 232 CID American Mtrs.	23	2000/39.1[d]	6[b]	6[b]	3[b]	0.62[d]	-	-	-	-	-	-	-	-
	2-stroke, 108.6 CID, 4-cyl, with Roots blower	-	2000/50[d]	0.024[b]	-	-	0.58[d]	-	-	-	-	-	-	-	Valve failure

[a] Speed in rpm; imep in psi
[b] Brake specific values
[c] Chassis dynamometer inertia weight
[d] Hot start
[e] With thermal reactor and swirl
[f] Emission index: pollutant mass flow/fuel mass flow

Table 5-6. SELECTED PRECHAMBER ENGINE PERFORMANCE TEST DATA - LARGE PRECHAMBERS

Organization	Engine test data							Vehicle test data							Potential problem areas
	Engine identification	Overall air-fuel ratio	Speed/imep[a]	Emissions, gr/ihp-hr			Specific fuel consumption, lb/ihp-hr	Vehicle identification	Test procedure	Emission control devices	Emissions, gr/mile			Fuel economy, mpg	
				HC	CO	NO_x					HC	CO	NO_x		
Ford Motor Company	Single-cylinder	-	1500/40-70	1.1-1.2	5-6	0.5	0.377-0.395	-	-	-	-	-	-	-	Fuel injection system; wall wetting
University of California at Berkeley	Modified CFR	21	-	-	-	-	0.378	-	-	-	-	-	-	-	High emissions
University of Wisconsin (Newhall concept)	Modified CFR	21	1600/full throttle	0.007	3.4	1.25	0.42	-	-	-	-	-	-	-	Combustion roughness

[a]Speed in rpm; imep in psi

emissions of about 3 gr/ihp-hr under steady-state, low-to-medium-load operating conditions. However, the HC emissions from these two engines are rather high, particularly in the case of the Continental Motors configuration, and no information is available regarding their performance under transient conditions. As expected, NO_x increases rapidly with increasing load while HC shows a tendency to decline.

Relative to standard 1972 model year vehicles, the emissions reported by Ford Motor Company for its torch ignition engine equipped 1972 experimental Gran Torino vehicle are quite low. However, since modulated EGR was employed in the engine in conjunction with air injection into the exhaust manifold, only part of the observed emission reduction is directly related to the use of the prechambers. Unless the raw HC and CO emissions of the engine could be further reduced, external emission control devices such as catalytic or thermal reactors would be required to meet the 1977 federal emission standards for light-duty vehicles. While similar NO_x and CO emission levels were achieved on the other vehicles considered in Table 5-4, the HC emissions from these vehicles were considerably higher than for the Gran Torino. Although the available data sample is rather limited, there is strong evidence of an inherent HC problem in spark-ignition engines employing small prechambers.

5.1.3.1.2 Fuel Consumption

As shown in Table 5-4, the specific fuel consumption of the Teledyne Continental Motors prechamber engine at medium load was 13 percent lower than that of the nonmodified baseline engine. Conversely, in the high-load regime, the fuel consumption of the two engines was practically identical. Even larger improvements in specific fuel consumption were reported by a number of Russian investigators for unspecified loads.

To date, the apparent fuel consumption advantage of prechamber engines under steady-state, light-load operating conditions could not be duplicated in vehicles tested over the Federal Driving Cycle. As indicated in Table 5-4, the fuel economy of these vehicles was equal to or only slightly better than that of the corresponding baseline vehicles. It should be emphasized, however, that the NO_x and CO emissions from the prechamber automobiles were substantially below the baseline levels.

5.1.3.2 Medium-Size Prechambers

5.1.3.2.1 Emissions

A considerable amount of engine and vehicle test data is available from this particular prechamber engine class. As shown in Table 5-5, the Eaton Corporation was able to achieve very low HC, CO, and NO_x emissions on a Ford Pinto engine equipped with an emission control system consisting of Eaton prechambers, a custom-made main chamber intake swirl port, and a thermal exhaust reactor. However, the raw HC and CO emissions from the prechamber engine were considerably higher than those obtained from the baseline engine which was operated at an air-fuel ratio of 17.4. While the NO_x emissions obtained by other investigators were considerably higher than the Eaton values, the raw HC emissions from the various engines are in reasonable agreement. Similar to the previously discussed small prechamber engines, the engines incorporating medium-size prechambers appear to have inherently high HC emissions which would have to be reduced by internal or external techniques to meet future vehicle emission standards.

Except for the high HC emission observed on the 1967 Saab automobile incorporating Thermo Electron Corporation prechambers, the HC, CO, and NO_x emissions from automobiles equipped with medium-size prechambers are quite low. In fact, as shown in Table

5-5, some of the vehicles have achieved emission levels below the 1977 federal standards. These include a 5000-lb General Motors vehicle equipped with a 350 CID prechamber engine, the 2000-lb Honda Civic with 2-liter CVCC engine, and General Motors Vega and Impala automobiles converted by Honda for CVCC. However, each of these vehicles was equipped with a thermal reactor for added CO and HC control. In addition, an early fuel evaporation (EFE) system was utilized by General Motors while Honda employed intake air heating, combined with some spark retard on at least one of its test vehicles, as a means of reducing the HC and CO emissions during the cold-start phase of the test cycle.

The prechamber engine powered Volkswagen Beetle which was tested without aftertreatment devices had NO_x emissions below 1 gr/mile. However, the observed average HC and CO emissions of about 2.3 gr/mile and 6 gr/mile, respectively, were considerably above the 1977 federal standards, indicating the need of some kind of aftertreatment device.

Comparison of the prechamber vehicle data listed in Table 5-5 indicates that NO_x is rather low for lightweight vehicles, but increases rapidly with increasing vehicle inertia weight. Conversely HC and CO tend to be independent of vehicle weight.

5.1.3.2.2 Fuel Consumption

Like the previously discussed small prechamber configuration, the data from single-cylinder engines employing medium-size prechambers indicate specific fuel consumption improvements up to about 20 percent at light loads relative to equivalent nonmodified engines. While improvements of similar magnitude were reported by Thermo Electron for its test vehicle, the fuel economy of the Honda CVCC Civic and Volkswagen prechamber Beetle vehicles was equal to or slightly lower than that of the corresponding baseline vehicles.

Conversely, fuel economy improvements up to 10 percent were reported by Honda for the Vega and Impala vehicles incorporating CVCC. Since the NO_x emission level of the prechamber engine equipped vehicles was reduced significantly from the baseline levels without a loss in fuel economy, this concept merits further investigation and development for potential use in future light-duty vehicles.

5.1.3.3 Large Prechamber

Typical emission and specific fuel consumption data from single-cylinder engines incorporating large prechambers are presented in Table 5-6. As indicated, the NO_x emission of the Ford engine in the low-to-medium-load regime is quite low while HC and CO are rather high, indicating a need for aftertreatment devices to meet future emission standards. As expected, the NO_x levels obtained by the University of Wisconsin at full load are higher than the Ford part-load values, while CO, and particularly HC, are lower.

5.1.4 Odor, Aldehyde, and Smoke

While quantitative odor data from prechamber engines are lacking, it appears that the odor characteristics of these engines are similar to those of conventional engines.

With respect to aldehyde exhaust emissions, the test data provided by Honda for a General Motors Impala vehicle converted to CVCC indicate aldehyde concentrations similar to conventional gasoline engines. Conversely, somewhat higher aldehyde levels were reported by Phillips Petroleum Company for its single-cylinder prechamber engine, particularly at high air-fuel ratios. In view of these differences, additional tests would be required before a meaningful assessment of the aldehyde emission characteristics of prechamber engines would be possible.

The available information regarding smoke emissions from prechamber engines is limited to visual observations made by the

University of Rochester, indicating no smoke at light engine loads and medium smoke intensity at full load.

5.1.5 Engine Noise

Relative to conventional engines, a number of investigators have reported higher noise levels for their prechamber engines, combined with occasional combustion roughness. The detonation-like noise encountered by the University of Rochester and Thermo Electron Corporation is attributed by these investigators to the higher pressure rise rates encountered in prechamber combustion, particularly under high-load conditions. Some reduction in the noise level was achieved by the University of Rochester by reducing the size of the communicating passage between the prechamber and main chamber.

The noise characteristics of the Honda CVCC engine have been shown to be comparable to equivalent conventional engines.

5.1.6 Engine Durability and Maintenance Requirements

Except for the Honda Motor Company, none of the prechamber engine developers has conducted formal durability test programs on its prechamber concepts. While the Honda CVCC engine has demonstrated durability characteristics similar to conventional engines, a number of mechanical problems were encountered in other designs related to overheating of the prechamber and failure of the prechamber intake valve and valve stem. However, these problems might be alleviated by means of prechamber design modifications and the use of better materials.

To date, no special maintenance problems or needs have been identified for any of the prechamber engine concepts presently under development. Based on the results from extensive durability test programs, Honda has concluded that the maintenance procedures required for its CVCC engines are similar to those of conventional spark-ignition engines. Also, the low mileage emission levels of these engines can be preserved by means of conventional maintenance procedures.

5.1.7 Engine and Vehicle Performance

As expected from theoretical considerations, the specific power output capability (maximum power per cubic inch displacement) of prechamber engines operating with lean overall air-fuel mixtures is lower than that of current equivalent automotive spark-ignition production engines. For example, a number of prechamber engine investigators, including Volkswagen, Phillips Petroleum Company, Thermo Electron Corporation, and the University of Wisconsin, have reported peak power losses varying between about 4 and 25 percent, depending upon the selected overall air-fuel ratio. Of course, the power loss could be counteracted by increasing the displacement of the engine at the expense of higher cost and some reduction in part-load fuel economy.

General Motors and other organizations are considering main chamber mixture enrichment to an air-fuel ratio of about 13:1 as a means of restoring engine power. While this approach appears to be attractive for economic reasons, it would result in a substantial increase in the emissions, particularly NO_x, under full-load conditions.

Based on vehicle tests conducted by General Motors, Honda Motor Company, and Volkswagen, the performance of their prechamber engine-equipped test vehicles was similar to that of conventional automobiles. Apparently, the prechamber vehicles exhibited good acceleration and response characteristics and the small power decrement was hardly noticed.

5.1.8 Vehicle Driveability

To date, very little information has been released by the automotive industry and other organizations regarding the driveability characteristics of vehicles equipped with prechamber engines.

Based on the limited test programs conducted by Ford Motor Company, General Motors,-Honda Motor Company, Combustion Control, and Walker Manufacturing Company, it is concluded that the driveability of prechamber vehicles should be comparable to that of conventional automobiles. In all cases, the prechamber engines started readily and in general, the vehicles demonstrated good acceleration performance. While some loss in maximum engine power was encountered in the lean air-fuel mixture regime, this loss would be eliminated by mixture enrichment at the expense of an increase in the emissions, particularly NO_x.

The driveability index of a 1972 torch ignition engine powered Ford Gran Torino vehicle varied between 5 and 7, compared with about 5.5 for the average standard automobile, and about 6 for luxury cars. In this case, the driveability index used by Ford covered a range between 0, indicating very poor driveability, and 10, indicating excellent driveability.

5.1.9 Fuel Requirements

Conflicting information has been reported by a number of investigators relative to the knocking characteristics and fuel octane requirements of prechamber engines. According to General Motors and Honda Motor Company, the octane requirement of their prechamber engines is comparable to that of equivalent conventional engines. In the case of General Motors, this was expected because, at full load, the same air-fuel ratio of about 13:1 was utilized for the prechamber and main chamber mixtures and for the conventional engines. Conversely, Volkswagen, Teledyne Continental Motors, Thermo Electron Corporation, University of Wisconsin, and a number of Russian sources have reported lower octane requirements for their prechamber engines relative to conventional engines. This advantage is attributed to the more rapid completion of the combustion process in the prechamber resulting in shorter exposure times of the end gas to the conditions that would promote fuel autoignition and detonation.

It is conceivable that other engine design and operating parameters might affect the octane sensitivity of prechamber engines, including the prechamber geometry, size of the communicating passage, turbulence level, and spark timing. Further investigation of these parameters is needed to provide a better understanding of the knocking characteristics of prechamber engines.

Distillate fuels such as JP-4, CITE, and diesel fuel have been used successfully in the Teledyne Continental Motors and Stanford University/Heintz prechamber engines. Although gasoline priming was required to start the Heintz engine, utilization of heavier distillate fuels in place of gasoline is potentially advantageous from a crude oil usage point of view, and should be further investigated.

5.1.10 Concept Assessment

5.1.10.1 New Engine Designs

5.1.10.1.1 General

In principle, all prechamber concepts and configurations discussed in Section 4 of this report are applicable to new light-duty automotive engines. However, the degree of difficulty and cost involved in the manufacture and incorporation of the different systems varies greatly, depending upon the particular system design considered For example, the configurations incorporating a cam-actuated prechamber intake valve and/or fuel injection into the prechamber are considerably more complex and costly than the designs utilizing unscavenged prechambers or prechamber check valves. These factors are further evaluated in Subsections 5.1.10.1.2 through 5.1.10.1.4.

All prechamber configurations tested to date have achieved some reduction in NO_x relative to equivalent conventional engines. However, the observed improvement in NO_x was invariably accompanied by inadequate HC and CO control. In particular, the HC emissions were quite high and comparable to the untreated levels obtained in current production engines. While efforts are continuing to reduce the raw HC and CO emissions from prechamber engines, most investigators feel that aftertreatment devices such as catalytic or thermal reactors would be required, possibly in conjunction with spark retard, to meet the 1977/78 federal standards for light-duty vehicles. Since spark retard is detrimental to fuel economy, minimization of the raw HC and CO emissions is considered to be the principal near-term development objective for prechamber engines.

5.1.10.1.2 Small Prechambers

The Cornell spark plug represents the simplest form of a prechamber system devised to date for potential use in automotive engines. Mass production of this device could be implemented fairly easily using existing tooling and conventional materials. Incorporation of the system which is designed to fit into the spark plug well, would require no engine modification. However, the concept is limited to engine operation in the fuel-rich mixture regime and its durability has not yet been adequately demonstrated.

No major fabrication and materials problems are projected by Ford Motor Company for its unscavenged torch ignition engine and by the developers of prechamber concepts incorporating pressure actuated check valves. Stainless steel appears to be adequate for the prechamber, whereas nickel alloys are preferred for the check valve.

While several of the small prechamber configurations, notably Ford's torch ignition engine, have demonstrated promising emission and fuel consumption characteristics, a number of potential problem areas would have to be resolved before these concepts could be seriously considered for use in new automobile engines. These include overheating of the uncooled prechamber body, fuel condensation in the prechamber fuel vaporizer and supply system, and reliable operation of the check valve throughout the operating range of the engine. In addition, formal durability test programs would have to be conducted to determine the emissions and fuel economy characteristics of these engines as affected by mileage accumulation. Until such time as this has been accomplished, no meaningful projections can be made regarding the prospects of this particular prechamber engine class.

5.1.10.1.3 Medium-Size Prechambers

Stimulated by the successful development and production of the Honda CVCC prechamber engine, the automotive industry has been concentrating its recent stratified charge engine development efforts in the area of medium-size prechambers. Like Honda, all investigators utilize a camshaft-actuated prechamber intake valve, combined with dual carburetion or fuel injection systems and a catalytic or thermal reactor for additional HC and CO control.

The manufacture of this type of prechamber engine would require significant modifications and additions to existing engine production lines. In particular, machining of the cylinder head (which includes the prechambers) would be considerably more involved than in conventional engines, and most likely would require the acquisition of new machine tooling. Other component changes would include the dual carburetor and the actuating mechanism for the third valve. General Motors has expressed some concern regarding the prospects of mass-producing the type of dual carburetion system required for its engine. This system would demand much closer production tolerances than current production carburetors to achieve the required accuracy and consistency in the prechamber and main chamber air-fuel ratios under all vehicle operating conditions. As a result, the cost of the carburetion system would increase considerably over current levels, and the use of low-pressure electronic or mechanical manifold fuel injection systems of the type employed in several of Volkswagen's current automobile models might prove to be advantageous with respect to system reliability, performance, and production cost.

With respect to materials, General Motors feels that conventional materials would be adequate for the fabrication of its prechamber engine except, perhaps, for the prechamber proper and the communicating passage which might require the use of better materials such as austenitic steel.

While the HC and CO emissions from the various test vehicles employing medium-size prechambers and thermal reactors are below or only slightly higher than the 1977 federal standards, the NO_x levels of these vehicles are considerably above the 1978 standard, particularly in the case of high-weight vehicles. However, as illustrated in Figure 5-1, NO_x could be further reduced by means of spark retard and/or EGR at the expense of substantial increases in the untreated HC emission and fuel consumption. These losses are comparable to those projected in Ref. 5-2 for conventional engines using EGR and mixture enrichment, and substantially higher than for conventional engines employing reduction catalysts. However, the development of reduction catalysts has not yet progressed to the point where these systems would be feasible for use in vehicle installations.

While no prechamber engine durability data are available, except for Honda's 1.5 and 2-liter certification vehicles, it appears that the durability of this type of prechamber engine should be comparable to conventional engines. Also, based on Honda's data, very little or no emission and fuel economy degradation is expected for these engines over 50,000 miles.

Except for Honda, whose prechamber engine is in production, the automotive industry is awaiting the successful completion

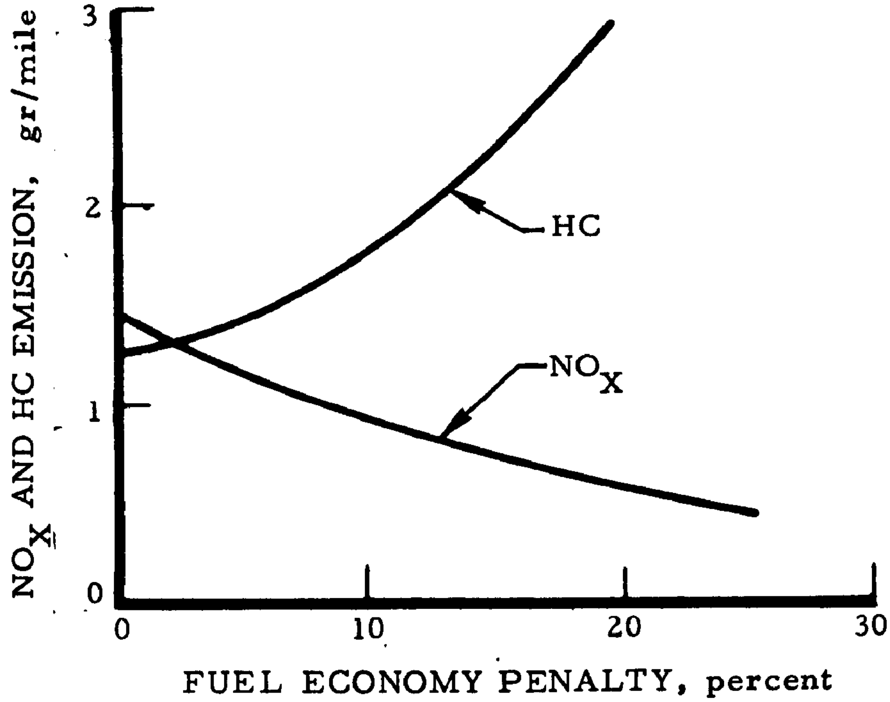

Figure 5-1. NO_x and HC emissions versus fuel economy penalty; General Motors prechamber vehicle; 4000-lb inertia weight; 1975 FTP (Ref. 5-1)

of their prechamber engine development programs before making decisions regarding the future of these engines. Other factors affecting this decision include the level of future NO_x emission standards and the degree of success achieved in the development of reliable low-cost NO_x catalysts for use in conventional engines operating near-stoichiometric. Since prechamber engines operate in the lean regime, NO_x catalysts are not applicable, and the achievement of very low NO_x levels in these engines would have to be accomplished by other means at the expense of lower fuel economy. In any case, because of the high capital investment requirements associated with mass production of these engines and the long leadtime of the tooling industry, prechamber engine production would probably start with one or two engine lines per manufacturer. According to General Motors, this could be accomplished within 24 to 30 months after program approval.

Based on the current state of the art, it appears that minimum NO_x emission levels of about 1.5 gr/mile (1975 FTP) could be achieved in prechamber-engine-equipped standard-size automobiles with reasonable fuel economy. Conversely, subcompact cars in the 2000-lb weight class are expected to achieve practical NO_x emission levels of about 0.6 - 0.8 gr/mile.

5.1.10.1.4 Large Prechambers

To date, the development of large prechamber concepts has not progressed much beyond the feasibility state. While single-cylinder tests indicate promising emission and fuel consumption characteristics, insufficient data are currently available to make meaningful projections regarding the emissions and fuel economy of such engines in vehicle installations.

In principle, the manufacturing procedure for these engines would be comparable to the previously discussed medium-size prechamber configurations. While no third valve would be used in these engines, the requirement of direct fuel injection into the prechamber would increase the complexity and cost of the fuel injection system relative to the manifold-type fuel injection systems considered by a number of investigators of small and medium-size prechamber concepts.

5.1.10.2 Retrofit Application

Based on the evaluation of all available prechamber engine design and performance information, it is concluded that only very few of the prechamber engine concepts devised to date would be applicable as retrofit devices for in-use light-duty vehicles. These concepts, which are listed in Table 5-7, fall into the category of small prechambers. While a number of additional prechamber concepts have

Table 5-7. CANDIDATE RETROFIT CONCEPTS

Organization	Concept designation	Prechamber volume ratio,[a] %	Prechamber air supply	Method of fuel supply		Test engine/vehicle	Other components	Development status	Test data
				Prechamber	Main chamber				
Ford Motor Company	Torch ignition engine	Small	From main chamber	From main chamber	Carburetion	Single-cylinder; 1972 Gran Torino	EGR	Early development	Table 5-4
Combustion Control	Morghen concept	3	Intake manifold, check valve	Auxiliary carburetor	Standard carburetor with smaller jets	1962 Ford Falcon 1974 M-151	Fuel vaporizer; timing adjustments; carburetor modific.	Early development	Table 5-4
Phillips Petroleum	-	2	Intake manifold, check valve	Auxiliary carburetor	Standard carburetor	Modified CFR	Aftertreatment device to reduce HC	Dormant	Table 5-4
Teledyne Continental Motors	Walker/ Continental concept	3	Intake manifold, cone valve	Auxiliary carburetor	Standard carburetor	L-141	Aftertreatment device to reduce HC	In development	Table 5-4
California State University at Sacramento	Morghen concept	3	Intake manifold, check valve	Auxiliary carburetor	Standard carburetor	1964 Ford Falcon	Aftertreatment device for HC control	Research continuing	Table 5-4
Cornell University	Cornell spark plug	0.5-1.0	From main chamber	From main chamber	Standard carburetor	1971 American Motors Matador	HC/CO device	Research continuing	Table 5-4
The Aerospace Corporation	-	8	Intake manifold, check valve	Auxiliary carburetor	Standard carburetor	Waukesha Model AGND	-	Dormant	Table 5-5

[a]Prechamber volume divided by total clearance volume

been patented during the past 50 years for potential use in retrofit applications, these configurations are not included in the table because of a lack of reliable design and performance data.

For economic reasons, all prechamber concepts employing a camshaft-actuated prechamber intake valve and/or fuel injection system were eliminated from the list of candidate retrofit devices. As previously discussed in Section 5.1.10, incorporation of cam-operated third valves would require substantial modification of the cylinder head and these changes are considered to be too complex and costly for application to in-use vehicles.

The Cornell spark plug represents a very simple and inexpensive piece of hardware that could be installed very easily in the spark plug well of existing engines. While some reduction in NO_x has been demonstrated in engines operating with rich mixtures, the Cornell device has no effect on HC, CO, and fuel economy and, therefore, its application would be of limited value. However, since the concept might have merits in conjunction with other emission control techniques/devices, further research and development work on this device might be desirable. Because of a lack of sufficient emission and durability data, a meaningful assessment of the device is not possible at this time.

While Ford's torch ignition engine is being developed for potential use in new automobile engines, the concept might be considered also for retrofit applications. In principle, the system, which has demonstrated low emissions and good fuel economy in one vehicle, is very simple and the auxiliary carburetor or third valve would not be required. However, the only available test data are from a vehicle equipped with air injection into the exhaust (manifold reactor) and operated at an air-fuel ratio of 16, and there is concern that many in-use automobiles might not be able to achieve satisfactory driveability at that mixture ratio. Also, reboring of the spark plug hole might be required which would be costly and, perhaps, even impossible for some engines. For these reasons, a meaningful feasibility assessment of this concept is currently not possible.

The other prechamber concepts listed in Table 5-7 incorporate an auxiliary carburetor, a suction pressure actuated prechamber intake valve, and a fuel vaporizer system. As shown in Table 5-4, some improvement in the emissions and fuel economy relative to nonmodified vehicles was reported by a number of investigators, notably Combustion Control and Teledyne Continental Motors. However, a number of problems have been encountered on these concepts, including failure and erratic operation of the prechamber check valve, overheating of the prechamber body, fuel condensation in the prechamber supply lines, and poor vehicle driveability. All these factors would have to be resolved before meaningful conclusions could be reached regarding the potential benefits of these systems in automotive retrofit installations.

5.1.11 Economic Considerations

5.1.11.1 General

Very little quantitative information is currently available regarding the manufacturing and installation costs of the various prechamber engine concepts described in Section 4 of this report. To date, the automobile manufacturers have been reluctant to disclose the projected cost or, in the case of the Honda Motor Company, the real cost of their prechamber engines relative to equivalent conventional engines. Conversely, most of the other organizations involved in prechamber engine research and development have given very little attention to the cost of their system on a mass-production basis.

Following are first-order cost estimates made by the report team on the basis of very preliminary information provided by several sources.

5.1.11.2 Initial Cost

The Honda CVCC engine represents the only prechamber engine that has reached the mass-production stage. Relative to conventional automotive engines, it incorporates a more complex cylinder head, a cam-actuated prechamber intake valve, a dual carburetor, and a thermal reactor. According to Honda, the cost of its four-cylinder CVCC engine to the consumer is about $160 higher than that of an equivalent conventional engine. Most likely, similar cost figures would apply to the carbureted cam-actuated three-valve configurations under development by a number of other manufacturers, including Ford Motor Company, General Motors, Volkswagen, and Eaton Corporation. Of course, the cost would be somewhat higher for inline 6 and V-8 engines. As discussed in Refs. 5-2 and 5-3, these figures are comparable to the cost of the emission control systems employed by most manufacturers to meet the 1975 California emission standards.

The manufacturing cost of cam-actuated three-valve systems utilizing fuel injection would be somewhat higher than the figures quoted above to account for the development and manufacturing costs of the fuel injection system. Included in this category are configurations considered by Ford Motor Company, Volkswagen, Stanford University, and the University of Rochester. Conversely, the Thermo Electron concept would be less expensive because of the absence of a third valve.

As expected, lower costs are projected for the prechamber configurations employing pressure actuated prechamber valves. Based on very preliminary data provided by Combustion Control, the manufacturing cost of these prechamber systems, including burden and profit, is estimated to be in the area of $90 for a six-cylinder engine, and somewhat higher for a V-8. Considering an allowance for dealer markup, the retail cost of the system would probably be of the order of $125 to $150. As discussed in Section 4, these systems are potentially applicable as retrofit devices for in-use automobiles. In this case, an additional installation cost of about \$25 to \$40 would have to be included for a total retrofit system cost of \$150 to \$200. Considering the uncertainties in the emission and fuel consumption characteristics of these systems, it appears that incorporation into in-use vehicles could not be justified at this time.

The manufacturing cost of large prechamber concepts of the type investigated by Ford and the Universities of Wisconsin and California would be comparable to that of the Honda system. While no third valve is used in these designs, the related cost savings would be counteracted by the higher cost of the fuel injection equipment.

The Cornell spark plug represents the least expensive prechamber configuration devised to date. The cost of each plug is estimated to be about two or three times that of a conventional spark plug. However, as previously noted, the Cornell concept is restricted to engines operating in the rich mixture regime, providing some NO_x reduction without affecting HC, CO, and fuel economy.

5.1.11.3 Operating Costs

Based on the data provided by Honda Motor Company and General Motors, the fuel economy of vehicles equipped with their prechambers is comparable to that of conventional 1974 model year vehicles. However, the raw HC and NO_x emissions from the prechamber engine are slightly lower and CO is substantially lower than the emissions from the conventional engine. Conversely, some fuel savings were reported by several investigators, particularly when older vehicles were utilized as the baseline. Unless these discrepancies can be resolved, it would have to be assumed that the fuel economy of vehicles powered by prechamber engines is similar to conventional engines.

A number of investigators have indicated that the fuel octane requirement of their prechamber engines would be lower than for equivalent conventional engines, resulting in potential crude oil savings at the refinery. In the research octane number range between 85 and 95, reduction of the octane requirement by one unit results in a 1 percent saving in crude oil. Additional savings might be possible due to the fact that unleaded gasoline could be used in prechamber engines. Since insufficient information is currently available on the octane requirement of the various prechamber engine concepts over their whole range of operating conditions, a meaningful evaluation of these factors is not possible at this time.

5.1.11.4 Maintenance Cost

Based on a number of 50,000-mile durability tests conducted by Honda Motor Company on its CVCC Civic automobile, it appears that the maintenance cost of prechamber engines of the Honda type might be lower than for conventional engines. Since catalysts are not required in prechamber engines to meet the emission standards for HC and CO, further savings might be realized relative to conventional vehicles employing catalysts. Because of their limited durability, these catalysts may require periodic replacement. Also, the emission of toxic sulfates which has been observed in vehicles equipped with catalytic emission control systems (Ref. 5-4) would not be a problem in prechamber engines.

5.2 STATIONARY GAS ENGINES

Only one stationary spark-ignition prechamber engine is currently in production in the United States. This engine utilizes two small unscavenged prechambers with gaseous fuel injection, each of which occupies about 1.5 percent of the total clearance volume. In this engine, the two-stage combustion process is aimed at assuring positive ignition without detonation under all operating conditions, particularly at the minimum specific fuel consumption point, rather than extending the lean limit or minimizing the emissions.

While these objectives have been met at lower NO_x levels, none of the other manufacturers of stationary spark-ignition engines has development plans for such engines. According to these manufacturers, the cost and risk factors associated with the development of stationary prechamber engines would be very high and could not be justified at this time.

Although retrofitting of stationary engines in the field might be possible in principle, it appears that this approach would not be economically feasible considering the high cost of conversion and the uncertainties regarding the benefits in terms of emission and fuel consumption improvements that might be realized by this procedure.

REFERENCES

5-1. P. R. Johnson, S. L. Genslak, and R. C. Nicholson, "Vehicle Emission System Utilizing a Stratified Charge Engine," SAE Paper No. 741157, presented at the International Stratified Charge Engine Conference, Troy, Michigan, 30 October - 1 November 1974.

5-2. "Report by the Committee on Motor Vehicle Emissions," The Environmental Protection Agency and the National Academy of Sciences, NAS, Washington, D. C., February 1973.

5-3. W. U. Roessler, A. Muraszew, and R. D. Kopa, "Assessment of the Applicability of Automotive Emission Control Technology to Stationary Engines," The Aerospace Corporation report prepared for the National Environmental Research Center of the EPA, Report No. EPA-650/2-74-051, July 1974.

5-4. "Automobile Emission Control - The Technical Status and Outlook as of December 1974," a report to the Administrator, Environmental Protection Agency, prepared by the Emission Control Technology Division, Mobile Source Pollution Control Program, EPA, January 1975.

APPENDIX A

PRECHAMBER ENGINE PATENTS

This Appendix presents a compilation of patents related to spark-ignition prechamber devices and concepts which were granted by the United States Patent Office between 1914 and 1974. A number of additional patents which are referenced in the listed patent disclosures are not included in Table A-1.

Table A-1. UNITED STATES PRECHAMBER SPARK-IGNITION ENGINE PATENTS

Inventor	Assignee	Title	U.S. Patent No.	Date filed	Date granted
Urataro Asaka	Honda Giken Kogyo Kabushiki Kaisha	Auxiliary Chamber Construction for Internal Combustion	3,844,259	11-29-1972	10-29-1974
S. Yagi and M. Atsumi	Honda Giken Kogyo Kabushiki Kaisha	Carburetor	3,842,810	11-27-1972	10-22-1974
Fiji Taguchi	Honda Giken Kogyo Kabushiki Kaisha	Intake and Exhaust Manifold Device of Internal Combustion Engine	3,832,984	6-22-1973	9-3-1974
T. Date and S. Yagi	Honda Giken Kogyo Kabushiki Kaisha	Auxiliary Chamber and Torch Nozzle for Internal Combustion Engine	3,830,205	12-29-1972	8-20-1974
G. Vogelsang and I. Geiger	Volkswagenwerk Aktiengesellschaft	Cylinder Arrangement for Combustion Engines Having a Pre-Chamber or Ante-Chamber and a Combustion Chamber	3,799,140	7-12-1971	3-26-1974
Bela Karlowitz	----	Method for Emission Control for Spark Ignition Engines	3,776,212	10-22-1970	12-4-1973
I. Geiger and G. Decker	Volkswagenwerk Aktiengesellschaft	Cylinder Arrangement Having a Combustion and a Precombustion Chamber Therein and a Separate Fuel Supply or Dosing Means Therefor	3,763,834	7-12-1971	10-9-1973

Table A-1. UNITED STATES PRECHAMBER SPARK-IGNITION ENGINE PATENTS (Continued)

Inventor	Assignee	Title	U.S. Patent No.	Date filed	Date granted
E. Braun and W. Brodbeck	Daimler-Benz Aktiengesellschaft	Rotary Piston Internal Combustion Engine With Externally Controlled Ignition by Means of a Spark Plug	3,738,331	4-26-1971	6-12-1973
Gustav Vogelsang	Volkswagenwerk Aktiengesellschaft	Cylinder Arrangement Having a Precombustion Chamber for Combustion Engines	3,738,333	7-9-1971	6-12-1973
R. C. Warner	Eldapat General Inc.	Anti-Fouling Spark Ignition Devices	3,710,772	8-7-1970	1-16-1973
Joseph A. Jozlin	William T. Sevald	Ignition Apparatus	3,710,764	2-26-1971	1-16-1973
N. G. Mozokhin, et al.	Gorkovsky Automobilny Zavod	System of Fuel Inject, and Precombustion-Chamber Spray Ignition in Piston and Rotary-Piston Internal Combustion Engines	3,682,146	3-4-1971	8-8-1972
F. Stumpfig	----	Method and Apparatus for Adapting Engine to Stratified Charge Operation	3,661,125	1-29-1968	5-9-1972
T. Suzuki, et al.	Kabushiki Kaisha Toyota Chuo Kenkyusho	Internal Combustion Engine With Sub-Combustion Chamber	3,659,564	3-20-1970	5-2-1972
T. Suzuki, et al.	Kabushiki Kaisha Toyota Chuo Kenkyusho	Internal Combustion Engine With Sub-Combustion Chamber	3,543,736	11-1-1968	12-1-1970

Table A-1. UNITED STATES PRECHAMBER SPARK-IGNITION ENGINE PATENTS (Continued)

Inventor	Assignee	Title	U.S. Patent No.	Date filed	Date granted
L. G. Clawson	----	Internal Combustion Engine	3, 508, 530	5-23-1968	4-28-1970
R. L. Fryer, et al.	----	Inlet Valve for Internal Combustion Engine	3, 479, 997	5-13-1968	11-25-1969
E. A. Von Seggern, et al.	----	Internal Combustion Engine, Fuel Supply System and Process	3, 443, 552	12-13-1966	5-13-1969
A. W. Evans, et al.	----	Ignition Amplifying Apparatus	3, 406, 667	9-29-1966	10-22-1968
J. S. Bernard	----	Method of Conditioning Liquid Fuels	3, 270, 722	4-22-1964	9-6-1966
N. N. Gitlin, et al.	----	Device for Modifying Spark Ignition in Carburetor Engines into Torch Ignition	3, 213, 839	4-5-1963	10-26-1965
J. H. Freeman Jr., et al.	----	Internal Combustion Engine with Ignition Cell	3, 207, 141	5-14-1963	9-21-1965
I. N. Bishop, et al.	Ford Motor Company	Combustion Chamber for an Internal Combustion Engine	3, 195, 519	3-28-1963	7-20-1965
E. A. Von Seggern, et al.	----	Excess Air Cycle Engine and Fuel Supply Means	3, 174-470	6-14-1963	3-23-1965
C. H. May, et al.	----	Combustion System for Internal Combustion Engines	3, 124, 113	6-20-1962	3-10-1964

Table A-1. UNITED STATES PRECHAMBER SPARK-IGNITION ENGINE PATENTS (Continued)

Inventor	Assignee	Title	U.S. Patent No.	Date filed	Date granted
R. M. Heintz	----	Stratified Charge Two-Cycle Engine	3,113,561	1-10-1961	12-10-1963
C. H. May, et al.	Walker Manufacturing Company	Ignition Device for Combustion Engines	3,066,662	8-26-1960	12-4-1962
C. H. May	Walker Manufacturing Company	Ignition Device for Internal Combustion Engines	3,066,661	8-26-1960	12-4-1962
R. M. Heintz	----	Internal Combustion Engine	2,983,268	5-7-1959	5-9-1961
K. Froehlich	Nordberg Manufacturing Company	High Compression Spark Ignited Gas Engine and Method	2,914,041	6-20-1956	11-24-1959
R. M. Heintz	----	Internal Combustion Engine	2,884,913	3-14-1958	5-5-1959
C. Stillebroer, et al.	Shell Development Company	Stratified Charge Internal Combustion Engine	2,849,992	12-15-1955	9-2-1958
Wolf-Dieter Bensinger	Daimler Benz A.G.	Cylinder for a Four-Cycle Internal Combustion Engine	2,803,230	12-12-1955	8-20-1957
J.T.M. Schlamann	Shell Development Company	Internal Combustion Engine with Ante-Chamber and Method of Operating Same	2,758,576	4-14-1952	8-14-1956
W.E. Meyer, et al.	The Texas Company	Internal-Combustion Engines	2,735,413	12-5-1952	2-21-1956

Table A-1. UNITED STATES PRECHAMBER SPARK-IGNITION ENGINE PATENTS (Continued)

Inventor	Assignee	Title	U.S. Patent No.	Date filed	Date granted
N.O. Broderson	----	Method of Operating Internal-Combustion Engines	2,690,741	7-31-1952	10-5-1954
G. F. Wright	Boeing Airplane Company	Antechamber Type Spark Plug Mechanism	2,642,054	5-20-1950	6-16-1953
A. Bagnulo	----	Engine with Stratified Mixture	2,422,610	10-26-1938	6-17-1947
H. D. Regar	----	Spark Plug	2,238,852	7-8-1939	4-15-1941
M. Mallory	----	Internal Combustion Engine	2,199,706	11-18-1937	5-7-1940
C. C. Groth	----	Internal Combustion Engine	2,196,860	2-8-1938	4-9-1940
J.A.H. Barkeij	----	Internal Combustion Engine	2,173,081	9-5-1933	9-12-1939
M. Mallory	----	Two-Cycle Internal Combustion Engine	2,156,665	2-25-1937	5-2-1939
G. K. Steward	----	Internal Combustion Engine	2,153,598	4-2-1936	4-11-1939
F. C. Mock	Eclipse Aviation Corporation	Internal Combustion Engine	2,142,280	6-29-1933	1-3-1939
M. Mallory	----	Internal Combustion Engine	2,121,920	2-8-1937	6-28-1938

Table A-1. UNITED STATES PRECHAMBER SPARK-IGNITION ENGINE PATENTS (Continued)

Inventor	Assignee	Title	U.S. Patent No.	Date filed	Date granted
A. E. Greene	----	Internal Combustion Engine	2,093,433	6-9-1933	9-21-1936
M. Mallory	----	Internal Combustion Engine	2,091,412	7-7-1936	8-31-1937
M. Mallory	----	Internal Combustion Engine	2,091,411	6-15-1936	8-31-1937
M. Mallory	----	Internal Combustion Engine	2,091,410	12-28-1935	8-31-1937
F. C. Mock	Eclipse Aviation Corporation	Internal Combustion Engine	1,998,785	1-11-1932	4-23-1935
H. M. Little	American Gyro Company	Spark Plug	1,929,748	8-15-1932	10-10-1933
J. O. Snyder	----	Compression Control Device for Internal Combustion Engines	1,925,086	7-23-1931	9-5-1933
J. J. McElhinney	----	Internal Combustion Engine	1,882,513	6-9-1930	10-11-1932
J. E. Shepherd, et al.	----	Starting Device for Internal Combustion Engines	1,841,643	9-24-1927	1-19-1932
G. S. Edlin, et al.	----	Internal Combustion Engine	1,772,988	8-14-1928	8-12-1930
R. Vreeland, et al.	----	Ignition Device	1,700,603	11-17-1927	1-29-1929

Table A-1. UNITED STATES PRECHAMBER SPARK-IGNITION ENGINE PATENTS (Continued)

Inventor	Assignee	Title	U.S. Patent No.	Date filed	Date granted
G. W. Smith, Jr.	White Motor Company	Internal Combustion Engine	1,696,060	12-21-1925	12-18-1928
F. M. Jobes	----	Internal Combustion Engine	1,649,700	4-23-1924	11-15-1927
F. M. Jobes	----	Internal Combustion Engine	1,629,795	3-4-1924	5-24-1927
M. J. Dikeman	----	Ignition Flash Plug	1,596,240	9-8-1924	8-17-1926
W. P. Rudkin	----	Internal Combustion Engine	1,584,657	1-17-1923	5-11-1926
E. Bugatti	----	Internal Combustion Engine	1,555,454	6-1-1923	9-29-1925
H. C. Kirby	----	Ignition Device for Internal Combustion Engines	1,483,730	5-19-1923	2-12-1924
G. W. Smith, Jr.	----	Internal Combustion Engine	1,483,619	12-2-1921	2-12-1924
L. C. Hall	----	Impulse Starter and Ignition Booster for Internal Combustion Engines	1,473,725	1-18-1922	11-13-1923
F. A. Smith	----	Ignition Device for Internal Combustion Engines	1,422,794	1-19-1920	7-11-1922
F. A. Smith	----	Ignition Device for Internal Combustion Engines	1,392,364	4-2-1921	10-4-1921

Table A-1. UNITED STATES PRECHAMBER SPARK-IGNITION ENGINE PATENTS (Continued)

Inventor	Assignee	Title	U.S. Patent No.	Date filed	Date granted
J. R. Simpson	----	Internal Combustion Engine	1,386,965	6-7-1915	8-9-1921
F. A. Smith	----	Ignition Device for Internal Combustion Engines	1,375,424	1-19-1920	4-19-1921
G. L. Meyer	----	Gas Engine Ignition	1,349,846	3-12-1918	8-17-1920
F. and E. Carter	----	Means for Igniting the Charge in Internal Combustion Engines	1,345,999	3-26-1919	7-6-1920
C. M. Stroud	----	Spark Plug	1,310,970	12-1-1916	7-22-1919
H. R. Ricardo	----	Internal Combustion Engine	1,271,942	2-1-1916	7-9-1918
O.K. Nicolaysen	----	Explosive Engine	1,264,548	10-22-1914	4-30-1918
E. D. Irwin	----	Internal Combustion Engine	1,204,986	3-27-1915	11-14-1916
F. V. Eastman	----	Ignition Device for Internal Combustion Engines	1,181,122	6-15-1914	5-2-1916
A. L. Penquite	----	Spark Plug Attachment	1,162,804	2-8-1915	12-7-1915
H. C. Waite	----	Internal Combustion Engine	1,135,083	6-8-1914	4-13-1915
L. S. Gardner	----	Internal Combustion Engine	1,095,102	6-10-1912	4-28-1914

APPENDIX B

VISITS AND CONTACTS

During the data-gathering phase of the study, the following organizations were visited or contacted by telephone.

Organization	Primary Contacts(s)
California State University Sacramento, California	Prof. F. H. Reardon
Cooper Bessemer Corporation Mount Vernon, Ohio	Mr. J. W. Holmes
Combustion Control Subsidiary of Systron Donner Corporation Berkeley, California	Mr. J. S. Winter
Cornell University Ithaca, New York	Prof. E. L. Resler, Jr.
Environmental Protection Agency Ann Arbor, Michigan	Dr. J. Bascunana
Fairbanks Morse Engine Division Colt Industries Beloit, Wisconsin	Mr. C. L. Newton
Ford Motor Company Dearborn, Michigan	Mr. T. J. Galbreath
General Motors Corporation Warren, Michigan	Mr. T. Fisher
Honda R. and D. Company, Ltd. Saitama, Japan	Mr. S. Yagi

Organization	Primary Contact(s)
Ingersoll Rand Corporation Painted Post, New York	Dr. C. K. Powell
Phillips Petroleum Company Bartlesville, Oklahoma	M. D. B. Wimmer
Stanford University Stanford, California	Prof. A. L. London
Teledyne Continental Motors Muskegon, Michigan	Mr. S. Berenyi
Thermo Electron Corporation Waltham, Massachusetts	Mr. L. G. Clawson
Waukesha Motor Corporation Waukesha, Wisconsin	Mr. N. Cox
Worthington-CEI, Inc. Buffalo, New York	Mr. L. Atwood
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--	Dr. K. Morghen Minden, Nevada

APPENDIX C

UNITS OF MEASURE–CONVERSIONS

Environmental Protection Agency policy is to express all measurements in Agency documents in metric units. With a few exceptions, this report uses British units. For conversion to the metric system, use the following conversions:

To convert from	to	Multiply by
°F	°C	5/9 (°F-32)
ft	meters	0.304
ft^2	$meters^2$	0.0929
ft^3	$meters^3$	0.0283
in.	cm	2.54
$in.^2$	cm^2	6.45
Btu	kcal	0.252
Btu/lb	cal/g	0.556
hp	kW	0.746
lb/10^6 Btu	g/10^6 cal	1.80
lb/$in.^2$	mm Hg	51.71
lb/hr	g/hr	453.6

TECHNICAL REPORT DATA
(Please read Instructions on the reverse before completing)

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16. ABSTRACT The report reviews the performance, emission, and operational characteristics of prechamber (or divided chamber) spark iginition engine concepts, including an analysis and evaluation of the applicability of these concepts to new automotive and stationary engines and retrofit installations. Relative to conventional automotive engines, prechamber engines exhibit very low carbon monoxide emissions accompanied by some reduction in the emission of nitrogen oxides. However, the hydrocarbon emission from prechamber engines is similar to that of conventional engines employing non-catalytic emission control systems, indicating a need for aftertreatment devices such as lean thermal reactors or catalytic converters. The fuel consumption of vehicles equipped with prechambers is similar to or slightly better than that of equivalent conventional vehicles at comparable levels of emission control.

17. KEY WORDS AND DOCUMENT ANALYSIS

a. DESCRIPTORS		b. IDENTIFIERS/OPEN ENDED TERMS	c. COSATI Field/Group
Air Pollution	Fuel Consumption	Air Pollution Control	13B, 21D
Spark Ignition Engines	Thermal Reactors	Stationary Sources	21B, 18I
Motor Vehicle Engines	Catalytic Converters	Prechamber	13F
Carbon Monoxide			07B, 07A
Nitrogen Oxides			
Hydrocarbons			07C

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