CONS/1101-1

MASTER

ENERGY

CONSERVATION

LEAN COMBUSTION IN AUTOMOTIVE ENGINES: AN ASSESSMENT OF THE ADDITION OF HYDROGEN TO GASOLINE AS COMPARED TO OTHER TECHNIQUES

February 1976

Work performed under Contract No. E(04-3)-1101-PHA-3

Aerospace Corporation
El Segundo, California

ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION

Division of Transportation Energy Conservation

DISCLAIMER

NOTICE

Available from the National Technical Information Service, U. S. Department of Commerce, Springfield, Virginia 22161

Price Paper Copy $8.00 (domestic)
$10.50 (foreign)
Microfiche $3.00 (domestic)
$4.50 (foreign)

ATR-76(7526)-1

CONS/1101-1
Distribution Category UC-96

LEAN COMBUSTION IN AUTOMOTIVE ENGINES: AN ASSESSMENT OF THE ADDITION OF HYDROGEN TO GASOLINE AS COMPARED TO OTHER TECHNIQUES

Prepared by
The Mobile Systems Group
Environment and Energy Conservation Division
THE AEROSPACE CORPORATION
El Segundo, California 90245

February 1976

Prepared for
ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
Transportation Energy Conservation Division
Washington, D.C.

Contract No. E(04-3)-1101, PA-3

The work described in this report was sponsored by the Transportation Energy Conservation Division of the Energy Research and Development Administration (ERDA). It was initiated when this Division, formerly known as the Alternative Automotive Power Systems (AAPS) Division, was a part of the Environmental Protection Agency (EPA). The work was initiated under EPA Contract No. 68-01-0417. Work was completed under ERDA Contract No. E(04-3)-1101, PA-3.

FOREWORD

This report, prepared by The Aerospace Corporation for the U.S. Energy Research and Development Administration (ERDA), Division of Transportation Energy Conservation, presents the results of an examination of the feasibility, practicability, performance, fuel economy, and emissions of the concept of the addition of hydrogen to gasoline for use as an automobile fuel, including a comparison of the hydrogen additive concept with other methods of achieving lean operation.

A concise review of important findings is presented in the "Highlights" section. The remaining sections provide detailed discussions of each study topic and are of interest primarily to the technical specialist. Section 2 presents a brief review of the technical background and aspects related to the theoretical attributes of lean combustion. It describes the basic emission reduction features of lean operation and identifies the thermal efficiency improvement potential of lean operation, together with the mechanisms which tend to limit such potential. Section 3 describes the basic features and characteristics of the methods of hydrogen storage and addition considered in the study. In the case of fuel reformer systems, this description includes system configuration, method of operation, operating conditions, product composition, and thermal efficiency effects. The effects of hydrogen addition on engine and vehicle characteristics are summarized in Section 4. These effects encompass emissions, exhaust temperature, engine thermal efficiency, vehicle fuel economy, engine power, fuel requirements, engine knock, and driveability. Section 5 examines the potential for incorporating engine design changes when lean operation with hydrogen addition is employed. These changes include higher compression ratio, reduced lean limit hydrogen requirements, and engine size [cubic inch displacement (CID)] effects. The onboard fuel reformer systems characterized in Section 2 are summarized and compared in Section 6 in terms of process operation, operating conditions, product gas composition, thermal efficiency, complexity, and relative advantages and disadvantages. Section 7 discusses the overall engine-fuel reformer system in terms of system design requirements and performance characteristics. The Jet Propulsion Laboratory (JPL) system is used as the basis for this characterization. Section 8 reviews the emissions and fuel economy characteristics of other contending lean combustion approaches. These include modifications to the conventional homogeneous charge spark ignition engine and various stratified charge engines. Specific attention is focused on the oxides of nitrogen (NO_x) versus fuel economy characteristics of these approaches. A comparison of all lean combustion approaches examined in the study is made in Section 9. This comparison includes the relative state of development as well as emissions and fuel economy effects. An output of this comparison is an estimate of the relative applicability of the various approaches versus postulated NO_x emission standard levels.

ACKNOWLEDGMENTS

Appreciation is acknowledged for the guidance and assistance provided by Messrs. Graham Hagey and E. Eugene Ecklund of the Energy Research and Development Administration, Transportation Energy Conservation Division, who served as technical monitors for this study.

Dr. Andrew Burke and Mr. Richard Kopa of The Aerospace Corporation made valuable contributions to the study.

Merrill G. Hinton

Merrill G. Hinton, Group Director
Mobile Systems

Approved by:

Joseph Meltzer

Joseph Meltzer, General Manager
Environment and Energy
Conservation Division

CONTENTS

CONTENTS (Continued)

CONTENTS (Continued)

CONTENTS (Continued)

FIGURES

FIGURES (Continued)

FIGURES (Continued)

FIGURES (Continued)

FIGURES (Continued)

FIGURES (Continued)

TABLES

TABLES (Continued)

TABLES (Continued)

HIGHLIGHTS

An examination was made of the feasibility, practicability, performance, fuel economy, and emissions of the concept of the addition of hydrogen to gasoline for use as an automobile fuel. The specific hydrogen addition concepts evaluated included onboard storage of hydrogen as a bottled gas, as a cryogenic liquid, and as a regenerable gas in a metal hydride storage system, and the onboard generation of hydrogen by the reformation of gasoline in a fuel reformer (or gas generator). Both partial oxidation and steam reforming fuel reformers were considered.

For perspective, comparisons were made of the hydrogen addition concept with the conventional spark ignition engine baseline and other lean engine concepts, e.g., advanced lean carbureted engines and stratified charge engines.

The following are brief highlights summarizing the major study results.

FEASIBILITY OF HYDROGEN ADDITION

1. Hydrogen addition via fuel reformation is a feasible method of achieving ultralean engine operation. This has been demonstrated for the partial oxidation fuel reformer approach in engine plus reformer dynamometer tests conducted by JPL and by the Siemens Corporation in the air-fuel ratio range of 25 to 30 (equivalence ratio Φ range of 0.5 to 0.6).
2. The amount of hydrogen required to sustain such ultralean combustion is substantial, ~ 10% to 15% by weight of hydrogen in the fuel mixture. This requires that a significant portion of the gasoline used be reformed. In the case of the JPL reformer concept, over one half of the gasoline flow has to be reformed to sustain a low equivalence ratio ($\Phi \sim 0.55$) operation. By design intent, the Siemens approach is to reform (gasify) all of the gasoline used by the engine.

3. The other methods of hydrogen storage and addition (compressed gas, cryogenic liquid, and metal hydride storage) are too voluminous, heavy, or costly for onboard vehicle use, at least in the foreseeable future.

EMISSION EFFECTS

1. The low NO_x and carbon monoxide (CO) emissions obtained during lean operation with fuel reformation are characteristic of lean operation, per se, and are not related to any intrinsic properties or characteristics of fuel reformation.
2. NO_x levels are directly related to the degree of leanness of combustion. In vehicle tests, using bottled hydrogen gas to simulate fuel reformer operation, JPL and General Motors (GM) achieved NO_x levels of 0.3 to 0.4 gm/mi, as measured by the 1975 federal test procedures (FTP). In these cases, the operating equivalence ratio Φ was variable with load, increasing from $\Phi = 0.55$ at low power levels.
3. The Siemens, engine plus gas generator, dynamometer test data indicate that NO_x levels with the gas generator at $\Phi = 0.7$ are only one-tenth those occurring at $\Phi = 0.91$ with the stock carburetor operating with gasoline only.
4. Ultralean combustion systems exhibit high hydrocarbon (HC) levels. Therefore, some form of exhaust aftertreatment (e.g., catalysts or thermal reactors) is required to reduce the high HC levels.
5. The engine exhaust gas temperature decreases significantly with operation at low equivalence ratios. This is due primarily to lower peak cycle temperatures resulting from less energy release per unit weight of charge air (less fuel added). On the basis of the JPL engine plus reformer tests, the exhaust temperature is typically 250 to 350°F lower at 1500 rpm and 200 to 300°F lower at 3000 rpm.

6. Low exhaust gas temperatures can degrade the effectivity of catalytic converters and thermal reactors. JPL experimentally examined this effect by adding a 1975 model year GM catalytic converter to its bottled hydrogen test vehicle. The 1975 FTP tests indicate the catalytic converter is ~ 77% efficient in reducing HC emissions, from 3.1 gm/mi without catalyst to 0.72 gm/mi with catalyst.

FUEL ECONOMY EFFECTS

1. No directly measured vehicle fuel economy data for vehicles incorporating fuel reformers are available. Data from the JPL and GM bottled hydrogen test cars are not adequate for this purpose since the fuel reformer efficiency effects can have a significant impact on overall vehicle fuel economy.
2. On the basis of lean combustion characteristics alone, lean operation with fuel reformation should give both lower NO_x levels and a fuel economy improvement over conventional-engine-powered vehicles controlled to higher NO_x levels with exhaust gas recirculation (EGR) and spark retard. The amount of fuel economy improvement would depend on the efficiency of the fuel reformer.
3. The Siemens engine plus gas generator data indicate that operation with the fuel reformer at an equivalence ratio Φ of 0.65 gives ~ 13% improvement in fuel consumption when compared to the stock carburetor operating with gasoline only at $\Phi = 0.91$. The JPL engine plus reformer data indicate a 10% to 20% improvement in brake thermal efficiency at $\Phi = 0.5$ to 0.6 over the stock carbureted engine with gasoline at $\Phi = 0.92$.

PERFORMANCE EFFECTS

1. A significant maximum power loss for the same engine displacement (CID) results from very lean operation. The loss in power is approximately in proportion to the operating equivalence ratio;

e.g., maximum power at $\Phi = 0.5$ is ~ 50% of that achievable in the same displacement engine at $\Phi = 1$ (stoichiometric) conditions.

2. Smaller size (CID) conventional engines can be used to match the power and fuel economy characteristics attributable to ultralean concepts, but an undetermined increase in NO_x emissions would occur because of higher engine loadings for the same vehicle speed and load requirements.
3. In principle, the JPL system concept provides a dual-fuel system approach which permits operation with gasoline only at the higher speed and load conditions. While this could overcome the maximum power loss of ultralean operation, it would require a sophisticated control system to vary reformer output in consonance with vehicle acceleration and speed demands.

DESIRABILITY OF UTILIZING FUEL REFORMERS

1. The desirability of utilizing fuel reformation for automotive application appears to depend principally on both future NO_x emission standards and on the status of other developments in lean engine technology.
2. Advanced lean carburetion (A/F = 17-19, $\Phi = 0.79 - 0.88$) can achieve FTP NO_x levels less than 1.5 gm/mi while providing fuel economy improvements of 10% to 15% compared with vehicles controlled to the same NO_x level with EGR and conventional carburetors. This capability has been demonstrated in tests of the Dresser and Ethyl carburetors.
3. Stratified charge engines have achieved NO_x levels in the 0.4 gm/mi range with EGR.
   a. Open chamber engines, utilizing the Texaco Controlled Combustion System (TCCS) or the Ford Programmed Combustion Process (PROCO), exhibit improved fuel economy (~ 10%) at these low NO_x levels in comparison

    to the original conventional spark ignition engine in the same vehicle. The original conventional engine also produces appreciably higher NO_x levels (3-4 gm/mi). Increasing the NO_x level in the TCCS and PROCO engines by reducing EGR flow rates results in still further improvements in fuel economy over the conventional engine at the same NO_x level.

    b. The Honda prechamber engine data exhibit a mixed picture. The Honda CIVIC vehicle with the prechamber engine had a fuel economy loss (~ 25%) while decreasing the NO_x emissions to ~ 0.4 gm/mi from the value of ~ 1.4 gm/mi obtained with the standard or conventional engine. A 1972 Chevrolet Vega, modified by Honda with the compound vortex controlled combustion (CVCC) prechamber engine, had a 10% fuel economy improvement over the standard Vega engine and at the same time decreased NO_x from 3.8 to 1.2 gm/mi.

4. Both reduction and tricomponent catalysts also offer the potential of achieving a 0.4 gm/mi NO_x level. They would be limited to slightly rich operation ($\Phi > 1.0$) to reduce NO_x and thus could not achieve the fuel economy gains possible with lean engine operation. Their use could, however, negate the current fuel economy losses (10% - 15%) attendant to the achievement of NO_x levels of 2 to 3 gm/mi with EGR and spark retard.

5. If future NO_x emission standards are set at 1.5 gm/mi or above, advanced lean carburetion concepts and stratified charge engines could be utilized to meet these NO_x goals and improve fuel economy over that obtainable from conventional engines using EGR and spark retard. There is no evidence to indicate that ultralean fuel reformer concepts would produce fuel economy superior to advanced lean carburetion or stratified charge engines; therefore, there does not appear to be an incentive to develop fuel reformers at this statutory NO_x level.

6. If future NO_x emission standards are set at 1.0 gm/mi or below, advanced lean carburetion may be ineffective as it has not demonstrated a capability for controlling NO_x at this level. Both ultralean homogeneous combustion with fuel reformation and stratified charge engines offer the potential to meet this statutory NO_x level and improve fuel economy over conventional engine values. Catalysts have the potential for controlling NO_x at this level, but not for improving fuel economy over conventional engine operation.
7. Stratified charge engine usage would probably require extensive development in each engine size (CID) needed, in order to optimize emissions and performance. Fuel reformer development could possibly be conducted in large measure independently of engine hardware with a modular approach and scaling techniques. Thus, successful development of a fuel reformer system could result in lower development costs than a series of stratified charge engines modified to meet a range of vehicle sizes.
8. Reformer and associated control system development remains to be achieved before vehicle system operational compatibility can be demonstrated or projected with reasonable certainty, including compatible HC control techniques. Further reformer development activity appears to be reasonable unless it is clear that statutory NO_x emission standards are to be above 1.0 gm/mi.

FUEL REFORMER CHOICES AND DEVELOPMENT REQUIREMENTS

1. Only the JPL and Siemens fuel reformer concepts are being actively pursued at this time. Both concepts utilize the partial oxidation approach with a catalyst. The Siemens reformer operates richer (A/F = 1.5) than the JPL reformer (A/F = 5) and also utilizes EGR. The JPL approach is to obtain chemical equilibium in the fuel reformation process, whereas the Siemens nonequilibrium approach is to obtain a gaseous fuel product, not necessarily the maximum possible hydrogen yield.

2. Steam reforming concepts have been in varying stages of research and exploratory development in the past. While steam reforming offers the potential for increased hydrogen yields from the processed gasoline, the attendant problems of water storage, water freezing, and soot formation have served to focus current interest on the partial oxidation approach as most suitable for automotive application.

3. Both the JPL and Siemens reformer concepts produce about the same amount of hydrogen per pound of gasoline processed (~ 0.1-lb hydrogen/lb gasoline). The thermal efficiency of the Siemens reformer is ~ 95% compared with ~ 80% for the JPL reformer. This efficiency difference is largely due to the achievement of chemical equilibrium in the JPL reformer as opposed to mere gasification of the gasoline in the Siemens reformer.

4. The Siemens concept requires that all gasoline be processed through the reformer and thus entails the power loss inherent in lean operation at all vehicle operating conditions. The JPL concept provides for varying the amount of gasoline processed as a function of vehicle speed and load demand (e.g., all-gasoline operation under maximum power demands). However, this approach negates the possibility of increasing the engine compression ratio to improve engine efficiency since the engine is thereby constrained to operate on the octane requirement of the gasoline alone.

5. Regardless of the specific reformer concept that one might prefer, any reformer development activities undertaken should initially be concentrated in the following areas:

   a. Reformer characterization, including reformer startup, durability, and efficiency.

b. Control system development, including transient characteristics and the ability to schedule variable flow rates.

c. Associated HC control techniques, e.g., thermal reactors or catalytic converters having low temperature lightoff properties and close proximity to the exhaust manifold.

SECTION I

INTRODUCTION

1.1 STUDY BACKGROUND, OBJECTIVES, AND SCOPE

Many of the more promising near-term low emission alternative engine approaches under development by the auto companies embody the principle of lean overall air-fuel ratio operation, e.g., the Honda CVCC prechamber engine and the Ford PROCO and Texaco TCCS open-chamber stratified charge engines. One method suggested to enable lean operation with the conventional homogeneous charge spark ignition engine is to utilize hydrogen enrichment of gasoline to extend the lean operating limits. Several industrial firms as well as the National Aeronautics and Space Administration (NASA) and JPL, have conducted limited experimental investigations of a hydrogen reformer system which converts a portion of the vehicle gasoline fuel supply to hydrogen (and other constitutents), blends that reformer product with gasoline in the carburetor, and then uses this mixture as an engine fuel. The principal technical problem introduced by this approach is the addition of a fuel reformer system with its accompanying uncertainties as to performance and overall compatibility with the vehicle system.

The JPL initial fuel reformer work was conducted in 1973 under NASA sponsorship as part of the NASA Technology Applications and Aeronautics Programs. In December 1973, a brief examination of the data from the JPL program, as well as a review of other literature on the subject, indicated that the disclosed and available information at that time was insufficient to support a determination and evaluation of reformed fuel system performance (in terms of both emissions and fuel economy). The areas of major uncertainty included the following:

a. Possibility of high HC emission levels during lean operation.

b. Actual emission levels for all constituents (HC, CO, and NO_x) when reformed products are burned in the engine (the majority of the data was based on hydrogen plus gasoline mixtures).

c. Fuel reformer output and performance in a finite and reasonable size.

d. Fuel economy over the Federal Driving Cycle (FDC) or other normal driving conditions.

e. Definition of operating characteristics of the reformer-engine system over the full range of normal vehicle operation.

f. Reformer-vehicle performance capability over a full range of normal vehicle operation in customer use, including speed and load conditions not encountered in the FDC.

In order to resolve these uncertainties, EPA/AAPS contracted with JPL to perform a series of experimental tests and analyses based on the JPL fuel reformer concept. This program encompassed the following major elements:

a. Catalytic fuel reformer characterization

b. Definition of system and operational characteristics

c. Thermodynamic cycle analyses

d. V-8 engine tests

e. Cooperative Fuel Research Council (CFR) engine tests

f. Estimation of performance potential and system compatibility

In addition to the JPL program, EPA/AAPS also contracted with The Aerospace Corporation for an analytical study. The objectives of this study were (1) to systematically examine the feasibility, practicability, performance, fuel economy, and emissions of the concept of the addition of hydrogen to gasoline for use as an automobile fuel; (2) to examine onboard reformer technology as it applies to this concept; (3) to review and assess data relevant to hydrogen and hydrogen reformer gasoline concepts; and (4) to assess and compare the hydrogen additive concept with other methods of achieving lean operation. Thus, the Aerospace study was to utilize data generated at JPL as well as other relevant data sources to determine the value, the advantages and disadvantages, of the hydrogen enrichment approach versus other contending lean-burn technologies. Definitive comparisons of fuel reformer approaches with the conventional spark ignition engine baseline and other lean engine concepts was regarded as prerequisite to a determination by AAPS if additional support for fuel reformer system development was warranted.

1.2 ACQUISITION OF RELEVANT DATA

The data and information reported herein were acquired and developed between 1 September 1974 and 31 January 1975. Much of the information related to fuel reformer characterization and the reformer plus engine operation is based on data taken by JPL during the EPA/AAPS program and is contained in an unpublished rough draft report submitted to EPA in November 1974. These data were compared and correlated with similar data for fuel reformer or hydrogen addition programs from the International Materials Corporation, Siemens Corporation, Phillips Petroleum Company, GM, and the University of Arizona.

The comparative data for other types of modifications to the homogeneous lean combustion spark ignition engine are based on recent work done by Dresser Industries and the Ethyl Corporation in the areas of improved carburetion and charge preparation. The comparative data for stratified charge engines are limited to the Texaco TCCS, the Ford PROCO and Fast Burn, and the Honda CVCC concepts.

1.3 ORGANIZATION OF THIS REPORT

Section 2 presents a brief review of the technical background aspects related to the theoretical attributes of lean combustion. It describes the basic emission reduction features of lean operation and identifies the thermal efficiency improvement potential of lean operation, together with the mechanisms which tend to limit such potentials.

Section 3 describes the basic features and characteristics of the methods of hydrogen storage and addition considered in the study. In the case of fuel reformer systems, this description includes system configuration, method of operation, operating conditions, product composition, and thermal efficiency effects.

The effects of hydrogen addition on engine and vehicle characteristics are summarized in Section 4. These effects encompass emissions, exhaust temperature, engine thermal efficiency, vehicle fuel economy, engine power, fuel requirements, engine knock, and driveability.

Section 5 examines the potential for incorporating engine design changes when lean operation with hydrogen addition is employed. These changes include higher compression ratio, reduced lean limit hydrogen requirements, and engine size (CID) effects.

The onboard fuel reformer systems characterized in Section 2 are summarized and compared in Section 6 in terms of process operation, operating conditions, product gas composition, thermal efficiency, complexity, and relative advantages and disadvantages.

Section 7 discusses the overall engine and fuel reformer system in terms of system design requirements and performance characteristics. The JPL system is used as the basis for this characterization.

Section 8 reviews the emissions and fuel economy characteristics of other contending lean combustion approaches. These include modifications to the conventional homogeneous charge spark ignition engine and various stratified charge engines. Specific attention is focused on the NO_x versus fuel economy characteristics of these approaches.

A comparison of all lean combustion approaches examined in the study is made in Section 9. This comparison includes the relative state of development as well as emissions and fuel economy effects. An output of this comparison is an estimate of the relative applicability of the various approaches versus postulated NO_x emission standard levels.

SECTION II

TECHNICAL BACKGROUND

The general interest in lean combustion or lean engine operation has been largely related to the potential for both exhaust emissions reduction and increased engine thermal efficiency.

2.1 EXHAUST EMISSIONS REDUCTION

As illustrated in Figure 2-1, HC, CO, and NO_x concentrations in the exhaust of uncontrolled spark ignition engines are strongly a function of the operating air-fuel ratio. At the stoichiometric air-fuel ratio, NO_x production is very high, while HC and CO production is relatively low. For air-fuel ratios between approximately 17 and 19, levels for all three constituents

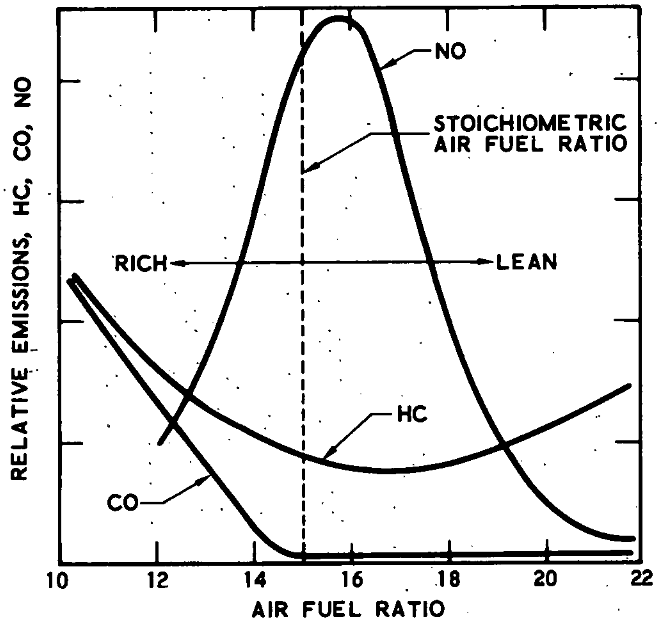


Figure 2-1. Effect of air-fuel ratio on emission levels: gasoline spark ignition engine (Ref. 2-1).

are reduced considerably from peak values. Currently, conventional engine operation is restricted to air-fuel ratios below ~ 17 in an effort to avoid excessive power loss and rough engine operation. Operation in the 17 to 19 range minimizes HC and CO levels and lowers that of NO_x, but the concurrent reduction of exhaust levels of all three species is far from sufficient to meet 1975-76 emission requirements. NO_x formation can be suppressed by operating in the "rich" regime (A/F ~ 11-13). In this region, however, HC and CO concentrations are very high, and vehicle fuel economy (mpg) is very poor. NO_x formation can also be suppressed by operating in the "very lean" regime (A/F > 20); in this region, CO concentrations are also quite low, although HC emissions are on a rising trend.

Current methods of NO_x control, i.e., EGR and spark retard, have resulted in degraded vehicle driveability and a loss in fuel economy. Projected future catalytic methods for NO_x control (reduction catalysts and tricomponent catalysts) require catalysts with durability characteristics that have not as yet been demonstrated. Thus, there has been considerable interest in methods for extending the lean operating limits of the homogeneous charge spark ignition engine in order to reduce NO_x. One such method is the addition of hydrogen to gasoline, which enables operation in the very lean regime because of the high flame speeds of the hydrogen in the hydrogen-gasoline mixture. The current state of the art and efficiency of such hydrogen addition systems are discussed in detail in Sections 3 through 7.

2.2 THERMAL EFFICIENCY EFFECTS

The efficiency of cyclic processes such as occur in spark ignition internal combustion engines is defined as the ratio of the useful work W accomplished by the process to the heat energy Q liberated by the combustion of fuel:

$$\eta = \frac{W}{Q} \tag{2-1}$$

The real engine cycle is usually represented by an equivalent constant-volume air cycle. The equivalent air cycle has a similar sequence of processes, the same ratio of maximum-to-minimum volume (compression ratio), the same pressure and temperature at a chosen reference point, and the same heat input per unit mass of air. With the assumption of a reversible adiabatic process (no heat losses) and of a perfect gas, the efficiency of the equivalent air cycle (Otto Cycle) is

$$\eta = 1 - \frac{1}{r}^{K-1} \qquad (2\text{-}2)$$

where r is the compression ratio and K is the ratio of specific heats (C_p at constant pressure and C_v at constant volume) of the working gases:

$$K = \frac{C_p}{C_v}$$

For an ideal gas, the gas constant R is equal to $C_p - C_v$, and therefore we find

$$K = \frac{C_p}{C_p - R} \qquad (2\text{-}3)$$

From equations (2-2) and (2-3), it is evident that the efficiency of the equivalent air cycle depends only on the compression ratio and on the specific heat of the working gases. The specific heats of all gases increase with increasing temperature. Therefore, more of the heat released during combustion is absorbed by the gases (combustion products) if the process occurs at a high temperature. This means that at high combustion temperatures relatively less of the total heat energy is available for immediate conversion to useful work.

2.2.1 Effect of Air-Fuel Ratio

Changing the air-fuel ratio of the combustible mixture in an internal combustion engine produces changes in the proportional concentration of the individual species (gases) resulting from the combustion process and in the total specific heat of the mixture. The resulting effect on the theoretical efficiency of a constant-volume fuel-air cycle is shown in Figure 2-2 (Ref. 2-2). The equivalent constant-volume fuel-air cycle has the same charge composition and density as an actual spark ingition engine cycle; therefore, it is more closely representative of the engine cycle than the equivalent air cycle.

In Figure 2-2 the thermodynamic efficiency η is plotted as a function of the equivalence ratio ϕ (i.e., ratio of the stoichiometric air-fuel

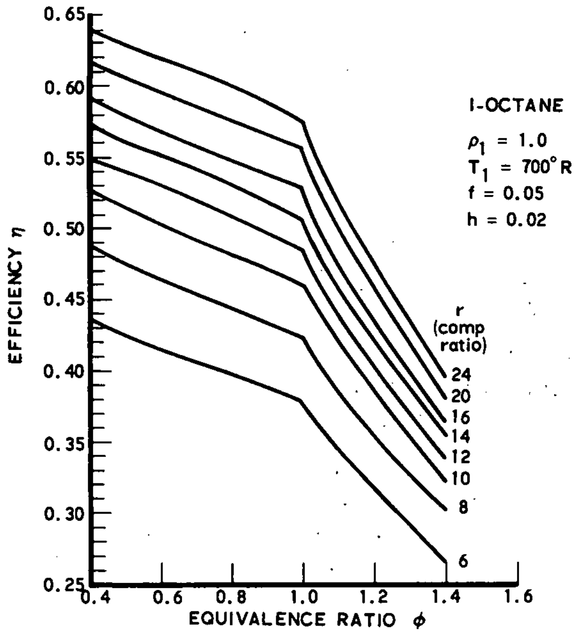


Figure 2-2. Theoretical efficiency of a constant-volume fuel-air cycle as a function of equivalence ratio with compression ratio as a parameter (Ref. 2-2)

ratio to the actual air-fuel ratio) with the compression ratio r as a parameter. The effect of the equivalence ratio Φ is twofold: (1) influencing the species concentration and (2) influencing the combustion temperature. For example, with a leaner air-fuel mixture the cycle efficiency increases because the total specific heat capacity of the combustion products decreases as a result of the larger proportion of nitrogen and oxygen, which have a lower specific heat than the water vapor and carbon dioxide, and of the lower specific heats of all components because of the lower combustion temperature resulting from the lower heat value of the lean mixture. Enrichment of the air-fuel mixture reverses the effect, and beyond the stoichiometric air-fuel ratio ($\Phi > 1$) the efficiency decreases more rapidly because of the additional strong contribution of the excess fuel (unburned HC) to the total specific heat of the combustion products (gases).

From the above discussion, it is apparent that the theoretical gains in the cycle efficiency of the reciprocating internal combustion engines when operated on ultralean air-fuel mixtures are substantial. In actual engine operation, additional effects can contribute to the improvement of the overall cycle efficiency:

1. Because of the resulting lower peak combustion temperature, the extent of dissociation of the combustion products is reduced and so is the heat absorbed in dissociation.
2. Due to the lower combustion temperature, the convective and radiation heat losses to the engine cylinder walls are substantially reduced.
3. The pumping losses of the engine actual work cycle (at part load) are reduced in proportion to the reduced throttling of the inlet air, because a substantially larger quantity of air is admitted at ultralean engine operation.

2.2.2 Effect of Flame Speed

In actual engine operation, the flame speed decreases with increasing leanness of the mixture, therefore, increased heat losses occur as the result of the prolonged duration of the combustion process. Furthermore, the cycle efficiency is reduced because a considerable part of the heat is released during the piston expansion stroke (and also during the late part of the compression stroke) rather than — as would be ideal — at the piston top dead center (TDC) (Ref. 2-3). As a result of these effects, the gains realizable in actual engine operation are quite modest, as exemplified by experimental test data. Figure 2-3 (Ref. 2-4) shows the experimentally determined indicated thermal efficiency as a function of air-fuel ratio, with compression ratio as a parameter. For example, at compression ratio CR = 10 the indicated thermal efficiency moderately increases with leaning of the mixture and reaches the highest value at about A/F = 22.2 (0.045 fuel-air ratio). Further leaning causes a rapid decrease in efficiency, and at an air-fuel ratio of about 25 to 1 (0.040 fuel-air ratio) the flame fails to propagate; thus the lean limit of engine operation is exceeded. The numerical

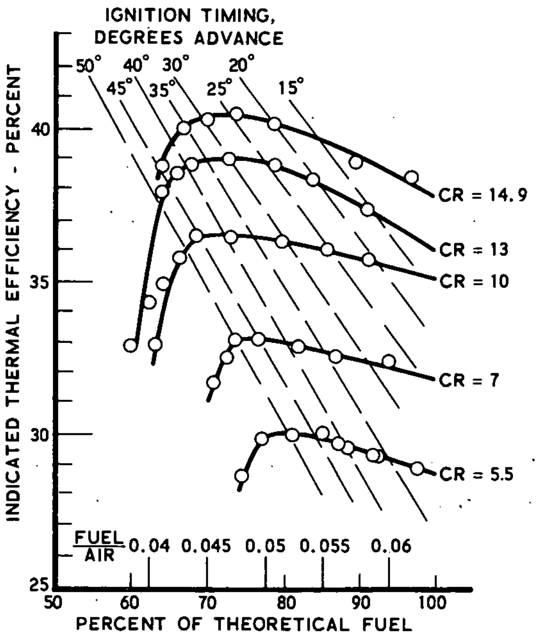


Figure 2-3. Indicated thermal efficiency as a function of air-fuel ratio with compression ratio as a parameter (Ref. 2-4).

value of this lean limit varies with fuel composition, engine design, and operating conditions, but it is generally in the neighborhood of 60% to 80% of the chemically correct mixture. The transition to zero flame speed near the lean limit of engine operation is abrupt. Conversely, the flame speed is highest where the air-fuel ratio is adjusted to that corresponding to the maximum temperature of the fuel-air cycle (maximum power).

The air-fuel ratio also has a strong effect on the ignition delay period, which is the time interval from the passage of the ignition spark to the beginning of flame development in the vicinity of the spark plug. With extremely lean mixtures, this delay period may increase by a factor of 4 to 6 relative to ignition delay in stoichiometric mixtures and therefore has to be compensated for by an advance in the ignition timing (Refs. 2-3 and 2-5).

Figure 2-4 presents a comparison of the theoretical and experimentally determined effect of air-fuel ratio on the indicated thermal efficiency of a spark ignition engine. The theoretical data are based on an analysis by Schweitzer and Grunder (Ref. 2-6), and the experimental data are taken from Figure 2-3 (Ref. 2-4). Also, data showing the effect of air-fuel ratio on flame speed, computed from Ref. 2-2 data, and chemical reaction speed, as determined by an independent experiment (Ref. 2-2), are plotted. All data are dimensionless and relative to the values at stoichiometric air-fuel ratio (A/F = 15). The trend of the experimentally determined data indicates that thermal efficiency increasingly deviates from the theoretical trend as the mixture is made leaner. The correlation between this deviation and the trend manifested by the flame speed is of interest. At an air-fuel ratio of approximately 24:1, the flame speed drops to almost half of its stoichiometric value, and any theoretical gain in indicated thermal efficiency is nearly cancelled because of the combined negative effects resulting from the slower burning in the engine cylinder. From Figure 2-4, it can be inferred that the difference between the theoretical and experimental indicated thermal efficiency is not linearly proportional to the reduction in

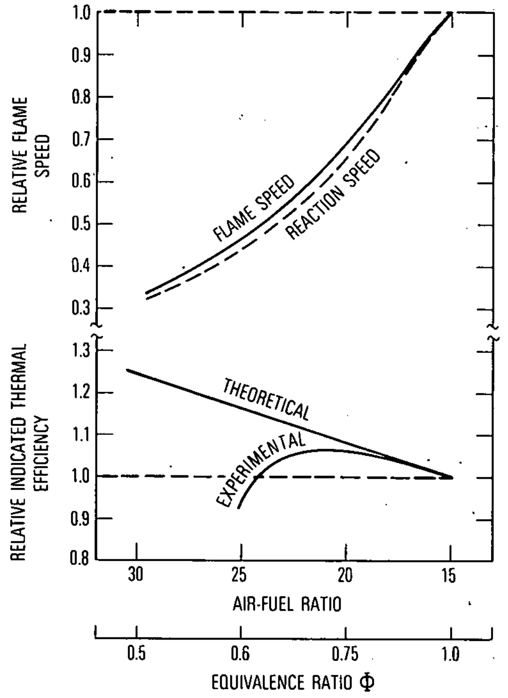


Figure 2-4. Comparision of theoretical and experimentally determined effect of air-fuel ratio on indicated thermal efficiency of the spark-ignition engine

flame speed. A small reduction in flame speed, up to ~20%, might still yield substantial improvement in the indicated thermal efficiency. However, a reduction in flame speed of ~ 50% appears to be critical; in excess of this no gains in indicated thermal efficiency may be expected. Nevertheless, some gains in brake thermal efficiency may still result because of the reduced engine pumping losses.

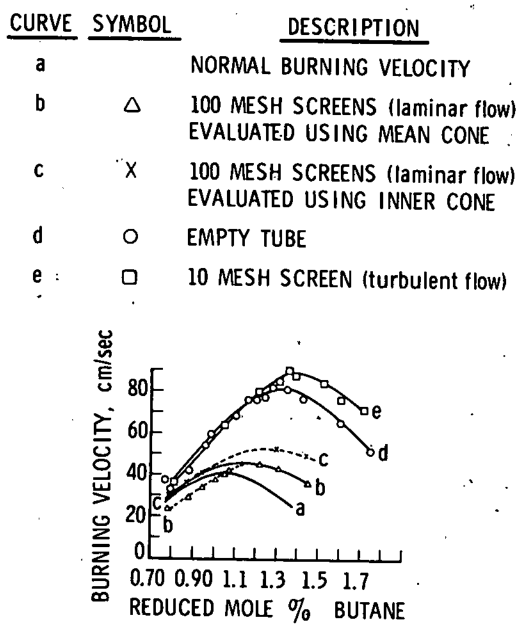
Another parameter which strongly affects the flame speed and the resulting engine efficiency is the quantity of the residual gases (exhaust gases) trapped in the engine cylinder and mixed with the fresh air-fuel charge. The dilution of fresh mixture with exhaust gases lowers the lean limit of permissible air-fuel ratio. Typically, in automotive multicylinder engines with large valve overlap, the air-fuel ratio may be severely limited as a result of considerable exhaust dilution.

2.2.3 Effect of Turbulence

Higher speed engines normally require only slightly greater spark advance settings than those running at normal or even quite low speeds (rpm). This means that the flame speed must increase nearly in proportion to engine speed. The increase of flame speed with increasing engine speed is due to the marked effect of mixture turbulence, as has been noted in many bomb experiments (Refs. 2-7 and 2-8) and in engine tests where the turbulence was varied independently of the engine speed (Ref. 2-9).

Depending on the engine design and on the engine operating regime, the turbulent gas motion in the engine cylinder may resemble a uniform vortex flow around the axis of the cylinder, a number of large scale turbulent vortices randomly moving in the cylinder space, or a small scale turbulence more or less uniformly distributed throughout the cylinder space. Generally, all three types of the turbulent gas motion, to a varying extent, are present during the combustion process. However, each type has a different effect on the ignition characteristic of the combustible mixture (Refs. 2-5 and 2-9), on the speed of flame propagation (and the resulting rate of the combustion process), and finally on the amount of heat loss due to the heat transfer from the burning gases to the relatively cold walls of the engine cylinder. As a result, engine performance and fuel economy are strongly influenced by the intensity and type of mixture turbulence.

The effect of various types of turbulent gas motions on the speed of flame propagation has been the subject of numerous studies (Refs. 2-10 through 2-16). An example of the kinetic limitation of an acceleration of flame speed by turbulence is illustrated in Figure 2-5 (Ref. 2-16). The flame velocity in the Bunsen flame of the butane-air mixture is plotted as a function of equivalence ratio, with the intensity of turbulence as a parameter. The test data indicate that the effect of turbulence on acceleration of the flame speed is strongly pronounced at stoichiometric and rich equivalence ratios but rapidly diminishes with leaning of the mixture. Under the test conditions reported, the effect of turbulence on flame speed vanishes when the stoichiometric mixture is diluted by $\sim$30% of excess air.



CURVE	SYMBOL	DESCRIPTION
a		NORMAL BURNING VELOCITY
b	△	100 MESH SCREENS (laminar flow) EVALUATED USING MEAN CONE
c	×	100 MESH SCREENS (laminar flow) EVALUATED USING INNER CONE
d	○	EMPTY TUBE
e	□	10 MESH SCREEN (turbulent flow)

Figure 2-5. Burning mixture flame velocity as a function of equivalence ratio for laminar and turbulent gas flow (Ref. 2-16)

In an actual engine combustion process, the resultant effect of the complex turbulence pattern on flame propagation is further influenced by the shape of the combustion chamber and the location of the spark plug (Refs. 2-9 and 2-17). Figure 2-6 (Ref. 2-9) presents specific fuel consumption as a function of air-fuel ratio with the shape of the combustion chamber as a parameter. The best fuel economy results when compact combustion chambers with the smallest surface-to-volume ratio (Heron type) are employed. The combustion time in this case is shortest; the heat losses are lowest; and the path of flame travel is short because of the compact shape of the combustion chamber. The turbulence is further increased by the shape of the piston, which during the compression stroke displaces the mixture from the circumference to the center of the combustion space. The heat losses are reduced because, in this case, a relatively smaller surface area of the combustion chamber is exposed to the burning mixture. At high engine speed, however, the overall effect may become negative because

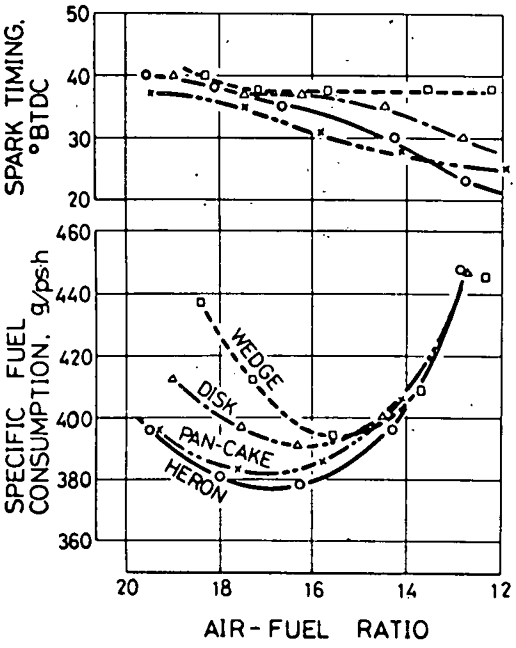


Figure 2-6. Specific fuel consumption as a function of air-fuel ratio with shape of the combustion chamber as a parameter (Ref. 2-9)

of excessively high turbulence and the resulting high heat transfer to the pistons and the engine cylinders (Ref. 2-18). In an extreme case, this may cause a mechanical failure due to burnout of the pistons or valves.

2.2.4 Effect of Mixture Stratification

One way to circumvent the problem of reduced flame speed, as was recognized early in the history of engine development, is by mixture stratification. In the stratified charge engine, the portion of the charge which surrounds the spark plug is nearly stoichiometric (fuel-rich), and the other portion is ultralean or air only. When the spark plug fires, the flame propagates swiftly through the stoichiometric portion of the charge. After combustion is complete and before much expansion has occured, both portions are mixed together to form an overall lean mixture.

However, in this procedure the losses due to high combustion temperatures, dissociation, and high specific heats are manifested in the stoichiometric portion of the charge. From a theoretical point of view, the stratified charge combustion process is inherently less efficient than the combustion process of the homogeneous lean mixture for the same overall air-fuel ratio of the charge. Moreover, because of high local temperatures and the nearly stoichiometric air-fuel ratio, a relatively large amount of nitric oxide is formed in the burning portion of the charge. Also, the unburned hydrocarbon concentration tends to be high, mainly from flame quenching in the transition zone between the stoichiometric (rich) portion and the air-only portion of the charge.

The numerous presently known design concepts of stratified charge engines may be divided into two general groups: open-chamber engines and divided-chamber engines. In the open-chamber configurations, exemplified by the Texaco TCCS and Ford PROCO engines, the charge stratification is effected in a single combustion chamber which is similar to that of a conventional spark ignition engine. In the divided-chamber stratified charge engines or prechamber engines, exemplified by Honda's CVCC engine, the charge stratification is effected by localization of the stoichiometric or fuel-rich mixture in the prechamber, which is connected by means of a narrow passage with the main chamber containing an ultra-lean or air-only charge.

The latter method of charge stratification is more simple and can be accomplished with the conventional carburetion system, whereas undivided- or open-chamber stratified charge engines require a relatively expensive high pressure fuel injection system. However, the inherent disadvantage of the divided-chamber method is that, because of large surface-to-volume ratio of the prechamber, connecting passage, main chamber configuration and the high charge turbulence, the heat transfer losses are high and tend to negate the fuel economy gains theoretically realizable in this type of engine. The overall effect on fuel economy and exhaust emissions depends on the design optimization of a given concept and is discussed in more detail in Section 8.

REFERENCES FOR SECTION 2

2-1. An Assessment of the Effects of Lead Addition in Gasoline on Emission Control Systems Which Might be Used to Meet the 1975-76 Motor Vehicle Emission Standards, TOR-0172(2787)-2, The Aerospace Corporation, El Segundo, Calif. (15 November 1971).

2-2. C. F. Taylor, The Internal Combustion Engine in Theory and Practice, Vol. II, The Massachusetts Institute of Technology Press, Cambridge, Mass. (1968).

2-3. L. C. Lichty, Internal Combustion Engines, Fifth Ed., McGraw-Hill Book Company, Inc., New York and London (1939), p. 106.

2-4. J. A. Bolt and D. H. Holkeboer, "Lean Fuel/Air Mixture for High-Compression Spark-Ignited Engines," SAE Transactions, Vol. 70, p. 195 (1962).

2-5. C. F. Taylor, The Internal Combustion Engine in Theory and Practice, Second Ed., Vol. I, The Massachusetts Institute of Technology Press, Cambridge, Mass. (1966).

2-6. P. H. Schweitzer and L. J. Grunder, "Hybrid Engines," SAE Transactions, p. 541 (1963).

2-7. J. A. Bolt and D. L. Harrington, "The Effects of Mixture Motion Upon the Lean Limit and Combustion of Spark-Ignited Mixtures," SAE Paper No. 670467, SAE Meeting, Chicago (May 1967).

2-8. S. Kumagai and I. Kimura, "The Effect of Turbulence on Flame Propagation in Gases," Fourth Symposium (International) on Combustion, Williams and Wilkins Co., Baltimore (1953), p. 667.

2-9. T. Tanuma, K. Sasaki, T. Kaneko, and H. Kawasaki, "Ignition, Combustion, and Exhaust Emissions of Lean Mixtures in Automotive Spark Ignition Engines," SAE Paper No. 710159, SAE Congress, Detroit (January 1971).

2-10. D. J. Patterson, "Cylinder Pressure Variations; A Fundamental Combustion Problem," SAE Paper No. 660129, SAE Congress, Detroit (January 1966).

2-11. W. R. Mickelsen and N. E. Ernstein, "Growth Rates of Turbulent Free Flames," Sixth Symposium (International) on Combustion, The Combustion Institute, Reinhold Publishing Corporation, New York (1956), p. 325.

2-12. *Basic Considerations in the Combustion of Hydrocarbon Fuels with Air*, National Advisory Committee for Aeronautics, Report No. 1300, Lewis Flight Propulsion Laboratory, Cleveland (1959).

2-13. B. Lewis and G. Von Elbe, *Combustion, Flames and Explosions of Gases*, Academic Press, Inc., New York and London (1961).

2-14. B. Lewis, R. N. Pease, and H. S. Taylor, *Combustion Process*, Princeton University Press, Princeton, New Jersey (1956).

2-15. D. A. Frank - Kamenetskii, *Diffusion and Heat Transfer in Chemical Kinetics*, Plenum Press, New York and London (1969).

2-16. K. Wohl, L. Shore, H. Von Rosenberg, and C. W. Weil, "The Burning Velocity of Turbulent Gases," *Fourth Symposium (International) on Combustion*, Williams and Wilkins Co., Baltimore (1953), p. 620.

2-17. Y. Sakai, H. Miyazaki, and K. Mukai, "The Effect of Combustion Chamber Shape on Nitrogen Oxides," SAE Paper No. 730154, SAE Congress, Detroit (January 1973).

2-18. H. R. Ricardo, *The High-Speed Internal-Combustion Engine*, Fifth Ed., Blackie and Sons, Ltd., London and Glasgow (1972).

SECTION 3

METHODS OF HYDROGEN STORAGE AND ADDITION

The supplementary hydrogen fuel needed to sustain ultralean combustion in an automotive engine must either be stored onboard the vehicle or be generated from the primary fuel (gasoline) by means of an onboard reformer. The simplest approach for providing the required hydrogen is to store it at high pressure, 2000 psi, in bottles (cylinders) located in a place remote from the engine. Because of its simplicity and flexibility, this approach has been used in most experimental studies (Refs. 3-1 and 3-2) of ultralean combustion using hydrogen-gasoline mixtures. Hydrogen can also be stored onboard a vehicle as a cryogenic liquid at -423°F in a vacuum-insulated container or can be chemically bonded to metal atoms as a metal hydride. All of the above methods of storing hydrogen onboard a vehicle incur large weight, volume, and cost penalties and, in addition, require distribution of gaseous or liquid hydrogen to central refueling stations. A second approach to providing the supplementary hydrogen involves the passage of a portion of the gasoline fed to the engine through an onboard fuel reformer. Hydrogen can be produced from gasoline either by steam reforming or partial oxidation. Both types of reformers for automotive use have been designed and tested in the laboratory. The various methods of storing and generating hydrogen onboard vehicles and controlling its flow to the engine are discussed and compared in the following sections.

3.1 ONBOARD HYDROGEN FUEL SYSTEMS

3.1.1 Compressed Gas and Cryogenic Liquid Systems

The supplementary hydrogen can be stored in a container onboard a vehicle as a gas or a liquid. The volume and weight characteristics of various hydrogen storage systems are compared in Table 3-1 with those of the primary fuel (gasoline) for the case of adding 10% by weight of H_2 to the gasoline flow to the engine. When hydrogen is stored as a gas it must be stored at high pressure to increase its density. Even at 2000 psi, the density of hydrogen is only 0.7 lb/ft^3, resulting in an energy density of only 30,000 Btu/ft^3 compared with 865,000 Btu/ft^3 for gasoline. As indicated in Table 3-1, the large system volume and weight for the storage of gaseous hydrogen seems to preclude this approach for any except experimental vehicles for which the simplicity and flexibility of the hydrogen flow control system becomes the dominant factor. A simple flow control system providing a constant hydrogen flow rate is shown in Figure 3-1a, while a somewhat more complex system which provides a constant hydrogen-gasoline fraction is shown in Figure 3-1b.

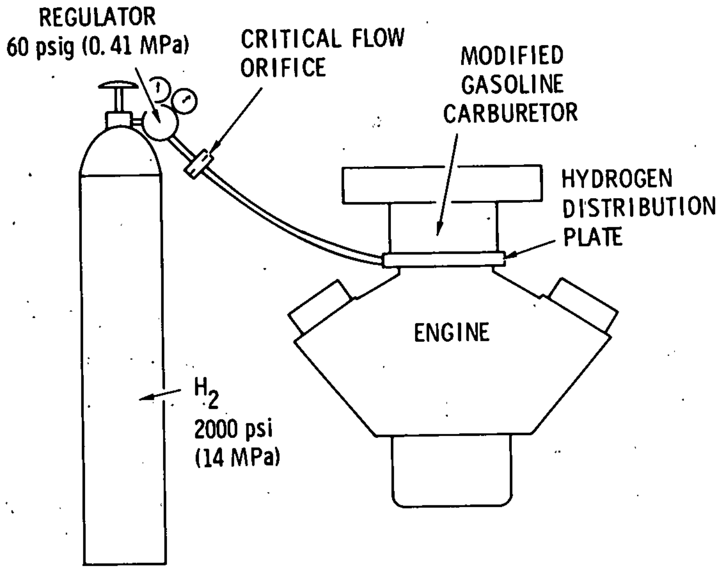
Table 3-1. Comparison of Weight and Volume for the Storage of Supplementary Hydrogen Onboard a Vehicle Using Various Methods[a]

Parameter	Primary Fuel (Gasoline)	Supplementary Hydrogen		
		Compressed Gas (2000 psi)	Liquid (-423°F)	Metal Hydride (MgNi)
Weight of fuel, lb[b]	108	10.8	10.8	10.8
Volume, ft^3[c]	2.4	20	3.5	2.5
Weight, lb[c]	130	500	80	150
Energy density, Btu/ft^3[c]	868,000	28,000	159,000	222,000

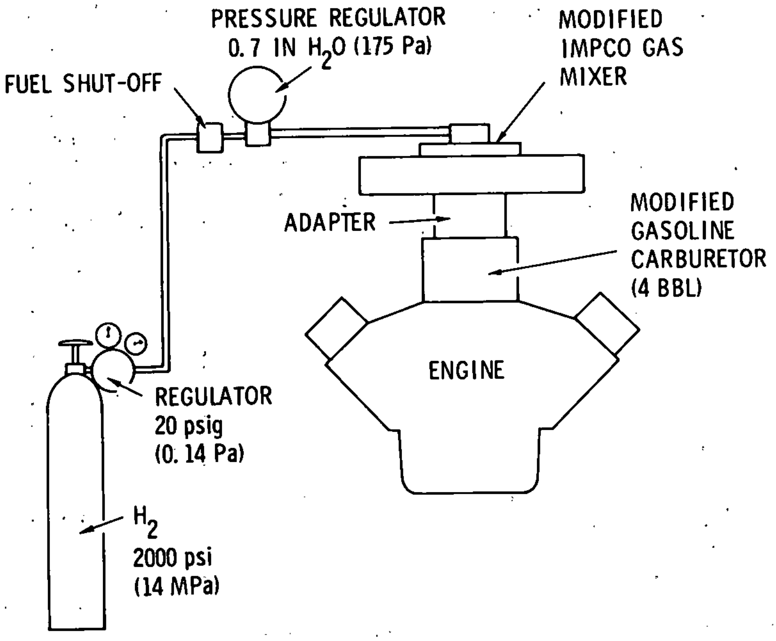
[a] Refs. 3-3 through 3-6.

[b] 18 gal of gasoline with supplementary H_2 of 10% by weight.

[c] Fuel plus tankage.



(a) CONSTANT FLOW HYDROGEN SYSTEM



(b) CONSTANT HYDROGEN FRACTION SYSTEM

Figure 3-1. Constant flow hydrogen systems: (a) basic constant flow hydrogen system; (b) constant hydrogen fraction system (Ref. 3-2)

When the hydrogen is stored as a liquid, it must be maintained at a very low temperature (-423°F) in a vacuum-insulated container. The density of liquid hydrogen is 4.4 lb/ft^3, resulting in an energy density of about 160,000 Btu/ft^3. It is clear from Table 3-1 that the volume and weight of the liquid hydrogen system are much lower than in the compressed gas system but are, nevertheless, quite high when it is considered that the hydrogen is only a supplementary fuel. Other questions concern the durability, safety, and cost of liquid hydrogen storage for automotive applications. These topics are discussed in a recent paper (Ref. 3-3) from which it can be concluded that a liquid hydrogen system is likely to be expensive and that little data are available concerning the durability and safety questions. In addition, storage and distribution of liquid hydrogen to and from central refueling stations are even more difficult than the handling of compressed hydrogen gas. In summary, it appears that for several reasons storage of the hydrogen as a cryogenic liquid onboard the vehicle does not seem practical in the foreseeable future.

3.1.2 Metal Hydride Systems

Hydrogen can be stored onboard a vehicle as a metal hydride within a packed bed of porous, granular material impregnated with various metals and metal alloys. The most attractive combinations developed to date (Refs. 3-4 and 3-5) are magnesium and nickel (MgNi) and iron and titanium (FeTi), which absorb a significant atom ratio of hydrogen at pressures near or slightly above atmospheric. The important characteristics of MgNi and FeTi beds are listed in Table 3-2. Both of the hydride systems are quite heavy when compared with gasoline storage; the FeTi system is much the heavier of the two hydride systems. Also, the operating temperature of the FeTi bed is much lower than that of the MgNi bed. For automotive applications in which the relatively hot exhaust gases are available to heat the bed, the MgNi system appears to be the most attractive because of its lighter weight. FeTi beds could prove to be useful for supplying hydrogen during vehicle warmup periods.

Table 3-2. Metal Hydride Bed Properties[a]

Property	FeTi	MgNi
Bed pressure, atm	3	1
Bed temperature, °F	160	800
Maximum hydrogen content, %	1.5	7.3
Bulk density, lb/ft^3	150	56
Heat of dissociation, Btu/lb H_2	7,250	16,650
Thermal conductivity, Btu/hr °F ft	1.0	0.3
Energy density,		
Btu/lb[b]	772	3,750
Btu/ft^3[c]	115,900	210,000

[a] Refs. 3-4 and 3-5.

[b] Corresponding value for gasoline is 19,000.

[c] Corresponding value for gasoline is 870,000.

The hydrogen stored in a metal hydride can be released (evolved) by heating the bed above the equilibrium (operating) temperature corresponding to the storage pressure. As indicated in Table 3-2 a MgNi bed must be heated to about 800°F. The heat of dissociation of the hydride in a MgNi bed is 16,650 Btu/lb H_2 or about one-third the heat of combustion of the evolved hydrogen. The required heat can be transferred to the bed from the engine exhaust gases. When the bed is being loaded with hydrogen the reverse process (recombination) occurs, and at least part of the heat of formation of the hydride must be removed from the system by a coolant. The pressure and temperature at which the loading takes place depend on the thermodynamics and kinetics of the hydride formation process and the loading time deemed to be practical. The thermal and heat transfer aspects of the design of metal hydride beds are discussed in Ref. 3-5.

A schematic sketch of the MgNi hydride system for storage of the supplementary hydrogen is shown in Figure 3-2. The volume and weight characteristics of the hydride bed are compared in Table 3-1 with the compressed gas and cryogenic liquid storage systems previously discussed. The hydride system is comparable in weight and volume to the liquid hydrogen system. As evidenced by Figure 3-2, the overall hydride system onboard the vehicle is considerably more complex than the corresponding liquid hydrogen system in that the hydride system requires at least two heat exchangers and a means of circulating hot hydrogen through the bed. The control of the flow of the hydrogen in the liquid hydrogen storage system requires only a rather simple vaporizer-regulator much like that currently

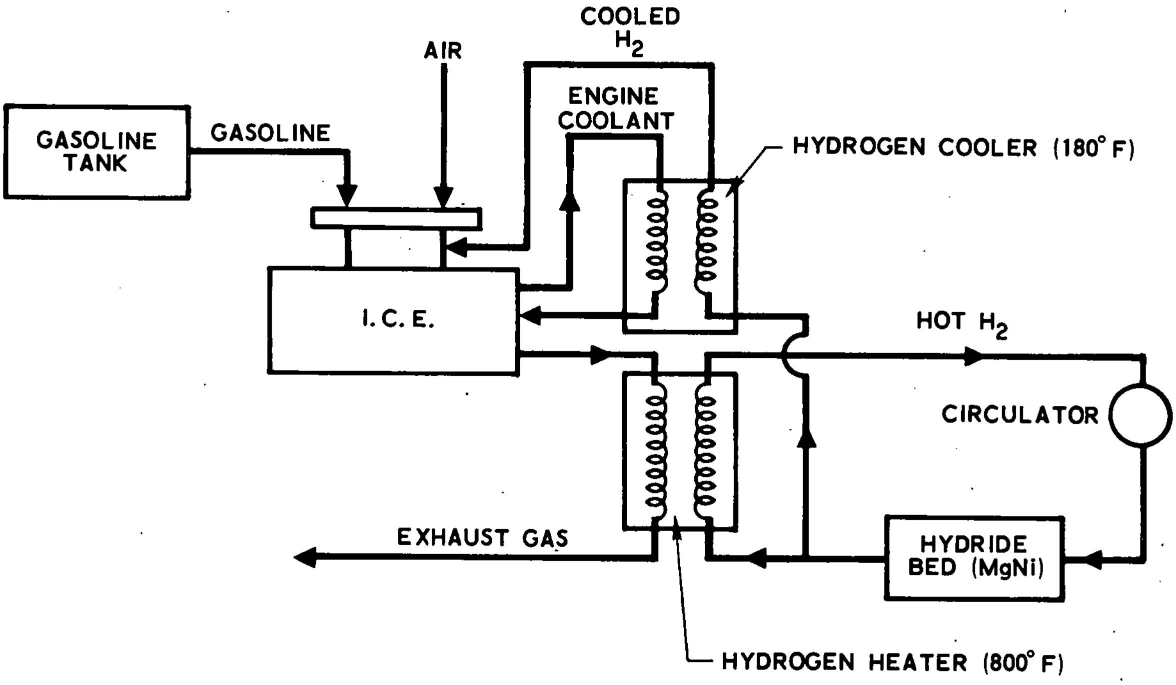


Figure 3-2. Ultralean combustion system using a metal hydride bed for supplementary hydrogen storage (Ref. 3-4)

being used in LP-Gas (propane) fueled vehicles. Loading of hydrogen into the hydride bed probably presents less difficulty than loading liquid hydrogen, once a high pressure source of hydrogen is available. Much additional research on metal hydrides and development of the associated systems are needed before metal hydride storage of the supplementary hydrogen onboard a vehicle can be considered a practical alternative to the fuel reformer approaches discussed in the next sections.

3.2 ONBOARD FUEL REFORMER SYSTEMS

3.2.1 Introduction and General Considerations

A convenient means of obtaining the hydrogen needed to sustain ultralean combustion in spark ignition engines is to convert a portion of the primary fuel (gasoline) to gaseous products which include the needed hydrogen. This process is termed fuel reforming, and the general technology needed to develop an onboard fuel reformer for automotive use has been well developed over the years in the petroleum and synthetic natural gas industries (Refs. 3-7 and 3-8). Recently, considerable work has been done by various groups to develop automotive fuel reformers both in connection with the ultralean combustion project of present interest and the more general use of gaseous fuels such as methane and hydrogen in spark ignition engines to reduce emissions and improve vehicle fuel economy. The variety of approaches that have been pursued in the reformer development programs can be divided into two categories: (1) partial oxidation and (2) steam reforming. In the case of partial oxidation, the gasoline is burnt (reacted) in an external combustor using a limited volume of air so that the combustion (or oxidation) is incomplete. A portion of the chemical energy of the gasoline is released to heat the mixture to the reaction temperature, but the resultant gaseous products contain species such as hydrogen, methane, and carbon monoxide which are themselves fuels and hence can be burnt in the engine. In steam reforming, the gasoline is reacted with steam at a relatively high temperature (1000-2000°F). The reaction products in this case contain the same gaseous fuels cited previously for the partial oxidation case. The steam reforming reactions are endothermic; therefore, the heat required must be obtained by burning a portion of the product fuel gas or by using the waste heat from the engine. In both reformer approaches, the gaseous products are hot and must be cooled before they can be injected into the engine. In comparing the attractiveness of the different reformer systems for the ultralean combustion application, the following characteristics are of prime importance: (1) the mass of hydrogen produced per pound of gasoline processed and (2) the overall thermal efficiency referenced to the heat of combustion of the input gasoline. The thermal efficiency depends both on the heat of combustion of the product gases and the extent to which waste heat from the reformer itself and the engine can be used to supply the heat needed by the reforming process. Since the steam reforming reactions are endothermic and, in principle, most of the required process heat can be obtained from the engine waste heat, it is possible in the steam reforming type of reformer for the overall thermal efficiency to be greater than 100%. Almost by definition, since the referenced thermal efficiency is based on the chemical energy of the reformer products, the partial oxidation-type reformers have a thermal efficiency of less than one, usually in the range of 80% to 90%.

Another broad distinction between various reformers is determined by whether a catalyst is used to promote the oxidation or reforming reactions or whether only control of the thermal environment is used. In general, reformers that use a catalyst are smaller in size and operate at lower temperatures than those that do not. These considerations can be very important for the automotive application for which space is usually limited and the temperature of the engine exhaust gases are relatively low (less than 1500°F in most cases). Another aspect of the use of catalysts in automotive applications is that gasolines often contain compounds (lead, sulfur, and phosphorous) which can deactivate catalysts after a relatively short period of use. Removal or elimination of these compounds from gasoline might be necessary if a catalyst is used in the fuel reformer.

One of the major problems encountered in the development of fuel reformers is the tendency of such devices to coke internally and/or to generate an exhaust stream containing objectionable levels of solid carbon or soot. For automotive applications, the soot levels must be low both because the product gases are fed into the engine and associated pollution control systems which have a low tolerance for soot and because the particulate pollution (smoke) standards for motor vehicles are severe, requiring at the very least a nonvisible exhaust. In most cases, the sooting problem determines or fixes the internal configuration and reformer operating conditions such as temperature, pressure, equivalence ratio, and steam carbon ratio. Particular attention is given to the sooting problem in the discussions of reformer development presented in later sections.

Various reformer development programs are discussed in the following sections. In each case, the reformer system is described in terms of hardware configuration, complexity, size, and operating conditions. Available test results concerning the composition of the product gases and the overall thermal efficiency of the reformer system are summarized. Particular problems, such as sooting, low thermal efficiency, and thermal lag, have been delineated. The application of the system to ultralean combustion is assessed.

3.2.2 Jet Propulsion Laboratory (JPL) Concept

A program to develop a compact onboard hydrogen generator for use with a hydrogen-enriched gasoline internal combustion engine has been under way at JPL, California Institute of Technology, for the last several years (Refs. 3-9 and 3-10). The partial oxidation approach has been taken, with work being done on both thermal and catalytic reactors. Most recent designs use catalysts, and one such type has been tested in conjunction with a V-8 engine on an engine dynamometer. Figure 3-3 illustrates the overall system approach, and Figure 3-4 depicts the basic reformer concept.

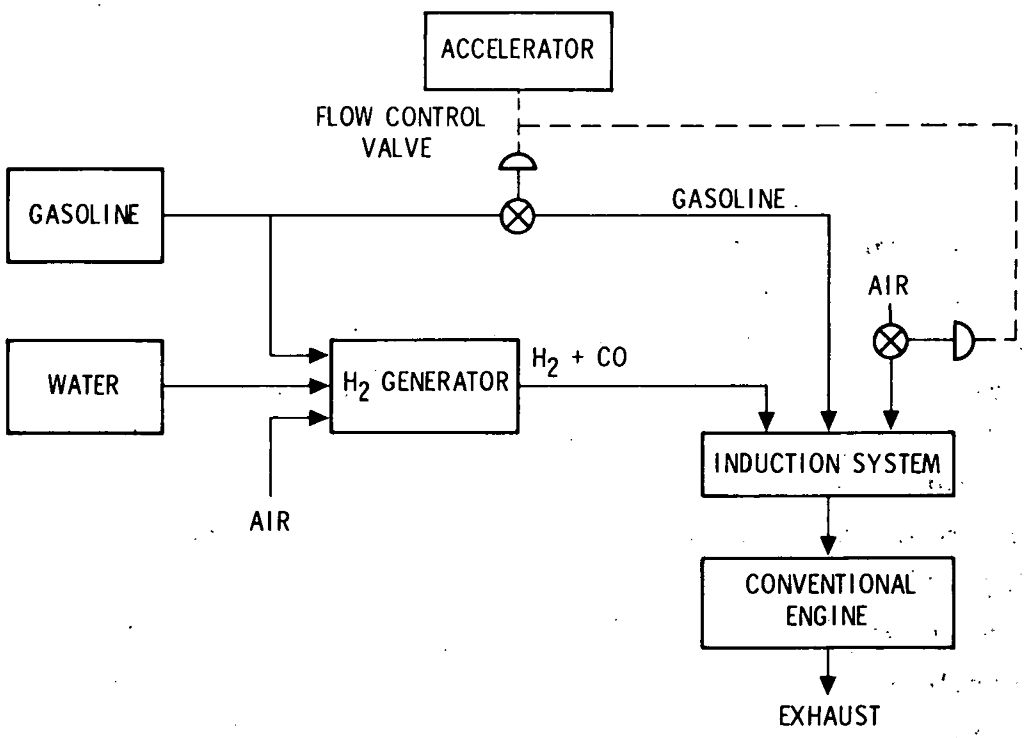


Figure 3-3. JPL system schematic

$$C_8H_{18} + 4O_2 \longrightarrow 8\,CO + 9\,H_2$$

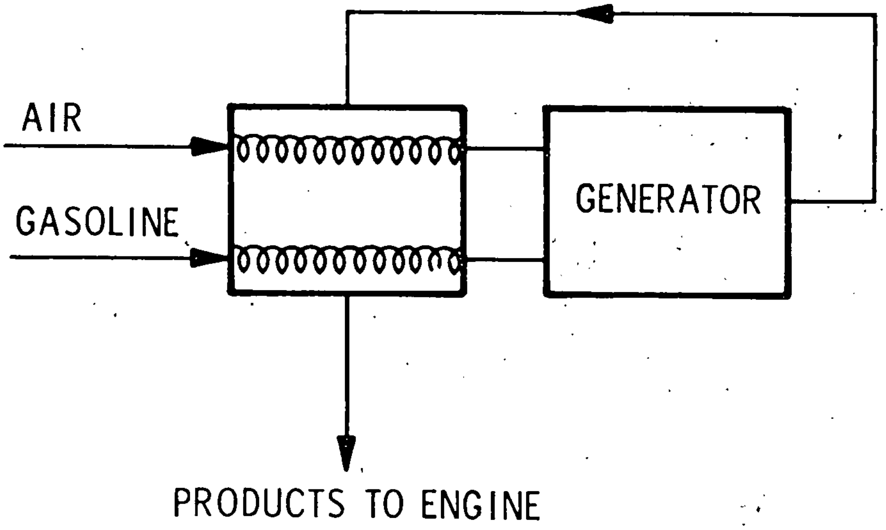


Figure 3-4. Partial oxidation hydrogen generator

3.2.2.1 Description of Reformer Hardware

A sketch of a JPL hydrogen generator (fuel reformer) is shown in Figure 3-5. The reactor portion of the device is about 10 in. long and 8 in. in diameter including the ceramic insulating liner and metal case. The most recent JPL reformer designs utilize a catalyst bed which is about 6 in. long and 3 in. in diameter (Figure 3-6). Commercially available nickel steam reforming catalysts have been used to promote partial oxidation, and their performance has been found to be satisfactory. Earlier work without a catalyst was abandoned because of a sooting problem and slow oxidation kinetics at practical reactor temperatures. At normal operating temperatures, the gasoline is vaporized before being mixed with preheated air and injected into the reactor. The reaction (partial oxidation of the gasoline) takes place in a flameless manner in the catalytic bed. During the warmup period before the system reaches its operating temperature, the gasoline is injected directly as a spray into the zone ahead of the catalyst where the air-fuel mixture is ignited. The reaction product gases

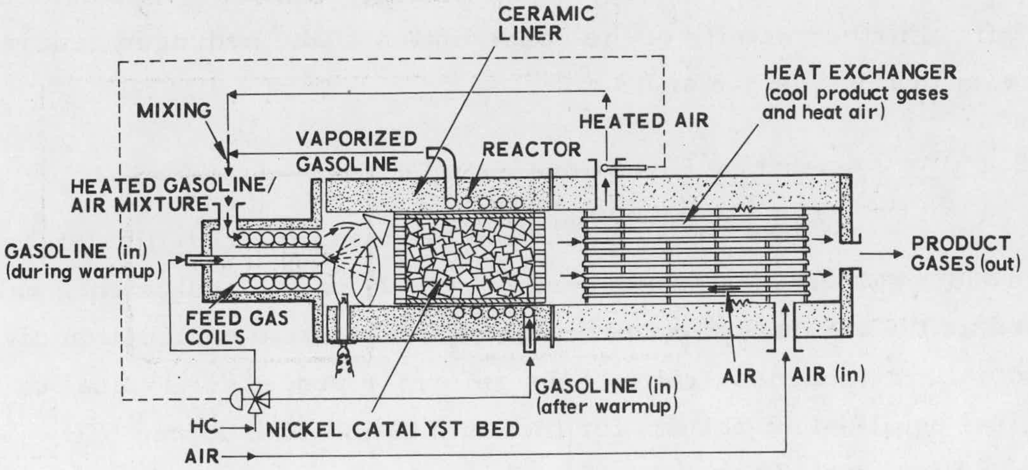


Figure 3-5. JPL catalytic hydrogen generator (Ref. 3-9)

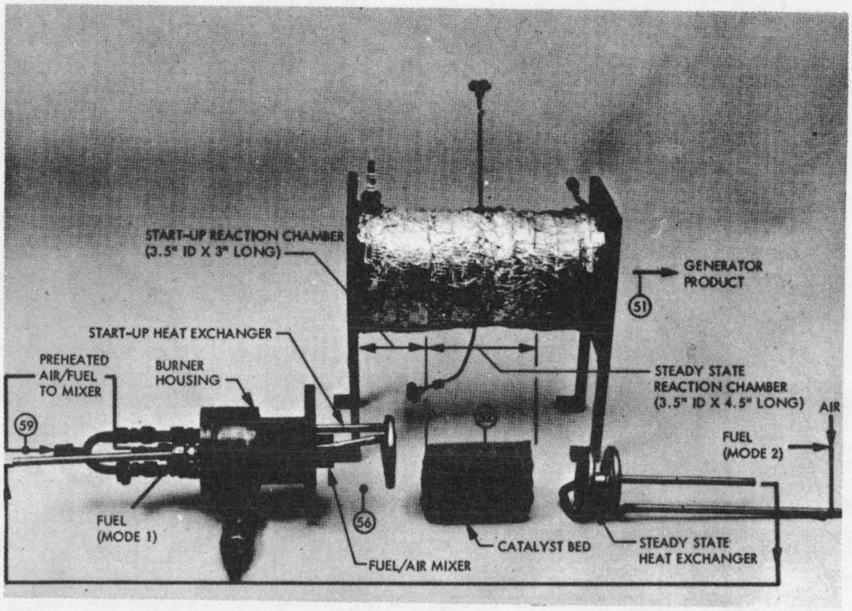
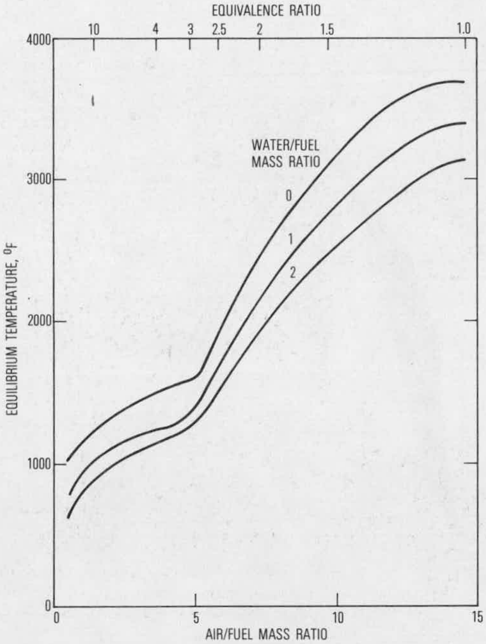


Figure 3-6. Compact catalytic hydrogen generator (Ref. 3-18)

are cooled by the incoming air in a double-wall tubular heat exchanger positioned just downstream of the reactor. The air is, in turn, preheated to about 450°F. The total length of the reactor and heat exchanger is about 2-1/2 ft. Further details of the construction of the hydrogen generator are indicated in Figures 3-5 and 3-6.

3.2.2.2 Operating Conditions

For the partial oxidation process, the reaction temperature and product gas composition depend primarily on the equivalence ratio, defined as the air-fuel mass ratio for stoichiometric combustion divided by the actual air-fuel mass ratio of the reformer process. Typical calculated chemical equilibrium results for the combustion of indolene ($CH_{1.92}$) are given in Figures 3-7 through 3-9. Figures 3-8 and 3-9 show that the maximum hydrogen yield (lb H_2/lb gasoline) and the onset of soot formation both occur at an air-fuel ratio of about 5, which is an equivalence ratio of 3. Figure 3-8 also indicates that at least for chemical equilibrium the hydrogen yield can be increased and the onset of soot formation delayed to lower air-fuel ratios by the addition of water to the reactants. JPL did not choose to pursue the water addition approach to handle the soot problem for several



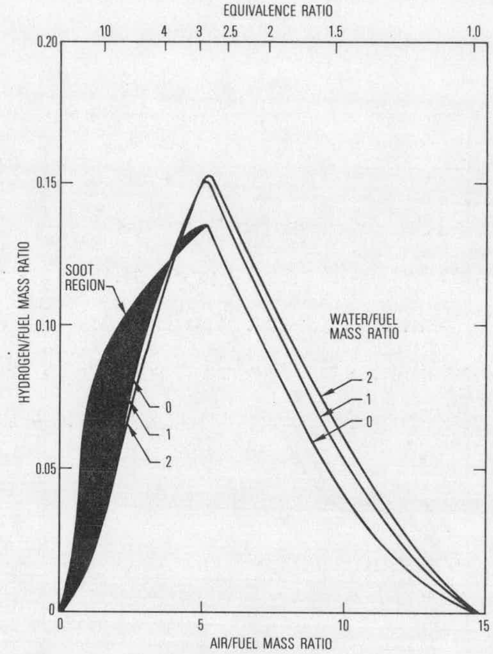
THEORETICAL EQUILIBRIUM, ADIABATIC COMBUSTION
INDOLENE ($CH_{1.92}$) - WATER-AIR AT 80°F, 44 psia

Figure 3-7. Equilibrium temperatures (Ref. 3-9)

reasons: (1) The resultant increase in hydrogen yield from the partial oxidation process was small; (2) use of a catalyst markedly lessened the soot problem and, in addition, significantly increased the hydrogen yield by accelerating the kinetics; and (3) adding water increases the overall complexity of the system and leads to problems with the water freezing at low ambient temperatures. It would be desirable to operate the reformer at an air-fuel ratio significantly lower than 5 in order to improve the overall thermal efficiency of the system. However, even with using a catalyst, JPL has not found it possible to do that.

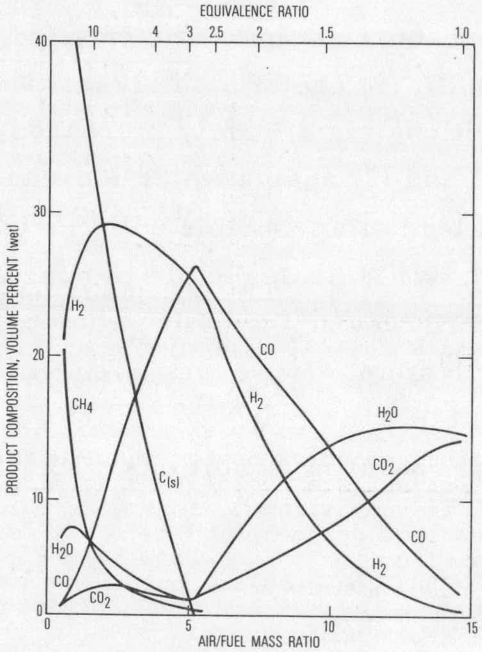
The typical operating conditions of the JPL hydrogen generator are as follows:

Air flow rate, lb/hr	45.6
Fuel flow, lb/hr	8.9
Air-fuel ratio	5.15
Equivalence ratio	2.83
Pressure, psig	1.40
Catalyst temperature, °F	1770



THEORETICAL EQUILIBRIUM, ADIABATIC COMBUSTION
INDOLENE ($CH_{1.92}$) - WATER-AIR AT 80°F, 44 psia

Figure 3-8. Hydrogen-to-fuel mass ratio and conditions for soot formation (Ref. 3-9)



THEORETICAL EQUILIBRIUM, ADIABATIC COMBUSTION
INDOLENE ($CH_{1.92}$) - AIR AT 80°F, 44 psia

Figure 3-9. Product gas composition (Ref. 3-9)

3.2.2.3 Test Results

The JPL hydrogen generator has been thoroughly tested and the product gas concentrations measured for a range of input gasoline flow rates. Typical mole and mass fraction concentrations of the various chemical species in the product stream are given in Table 3-3, which shows that the principal components on a mass basis are nitrogen and carbon monoxide.

Table 3-3. Product Mole and Mass Fractions for the JPL Gas Generator[a]

Output Conditions	Mole Fraction	Mass Fraction	Mass Output, lbm/hr
H_2	0.2160	0.0194	1.06
CO	0.2360	0.296	16.19
CH_4	0.0103	0.0074	0.404
C_2H_4	0.0009	0.0011	0.062
CO_2	0.0123	0.024	1.326
H_2O	0.0120	0.0097	0.529
N_2	0.5125	0.642	35.15
Total	1.0000	1.000	54.72
Mean molecular weight	22.33		
H_2/fuel	0.12		
H_2/C	1.925		
Exit pressure, psig	1.0		
Exit temperature, °F	1527.0		
Generator efficiency	0.785		
Efficiency/theoretical efficiency	0.957		
Fuel input to reformer	8.8 lb/hr		

[a] Ref. 3-10.

The output flow rates (lb/hr) of the fuel components (H_2, CH_4, and CO) in the product stream are shown in Figure 3-10 for input gasoline flow rates up to 16 lb/hr (~2.6 gal/hr). Over the range of gasoline flow rates tested, the generator produced between 0.12- and 0.13-lb H_2/lb gasoline input. The generator also produced a large amount of CO, 1.6- to 1.7-lb CO/lb gasoline input. The stoichiometry of the simple partial oxidation reaction for $\Phi \approx 3$

$$C_7H_{13.44} + 7/2\ O_2 \rightarrow 7CO + 6.72\ H_2$$

indicates that CO and H_2 appear in the product stream in about the concentrations measured. Chemical equilibrium calculations including all the species listed in Table 3-3 are, of course, required to predict in detail the product stream composition for a particular equivalence ratio Φ. Some or all of the CO could be shifted to H_2 and CO_2 by cooling of the product gas and the addition of steam, but this approach was not taken in the JPL work because of increased system complexity and the need to carry significant quantities of water onboard the vehicle.

The thermal efficiency of the JPL hydrogen generator was found to be 78.5%; i.e., the chemical energy content of the product gases were 78.5% of the chemical energy of the gasoline used to produce them. This represented 96% of the theoretical value based on chemical equilibrium calculations. As the CO shift reaction is exothermic, using that approach to obtain more H_2 slightly reduces the thermal efficiency of the partial oxidation process. Inherent in the partial oxidation process is a thermal efficiency of significantly less than one because part of the chemical energy in the gasoline is released to heat the mixture in the reactor. This energy cannot be recovered in a form useful to the internal combustion engine.

Tests (Ref. 3-9) were also made by JPL to study the poisoning effect on the catalyst of lead in the gasoline fed to the hydrogen generator. No deactivation of the reforming catalyst was noted in 100 hr of using

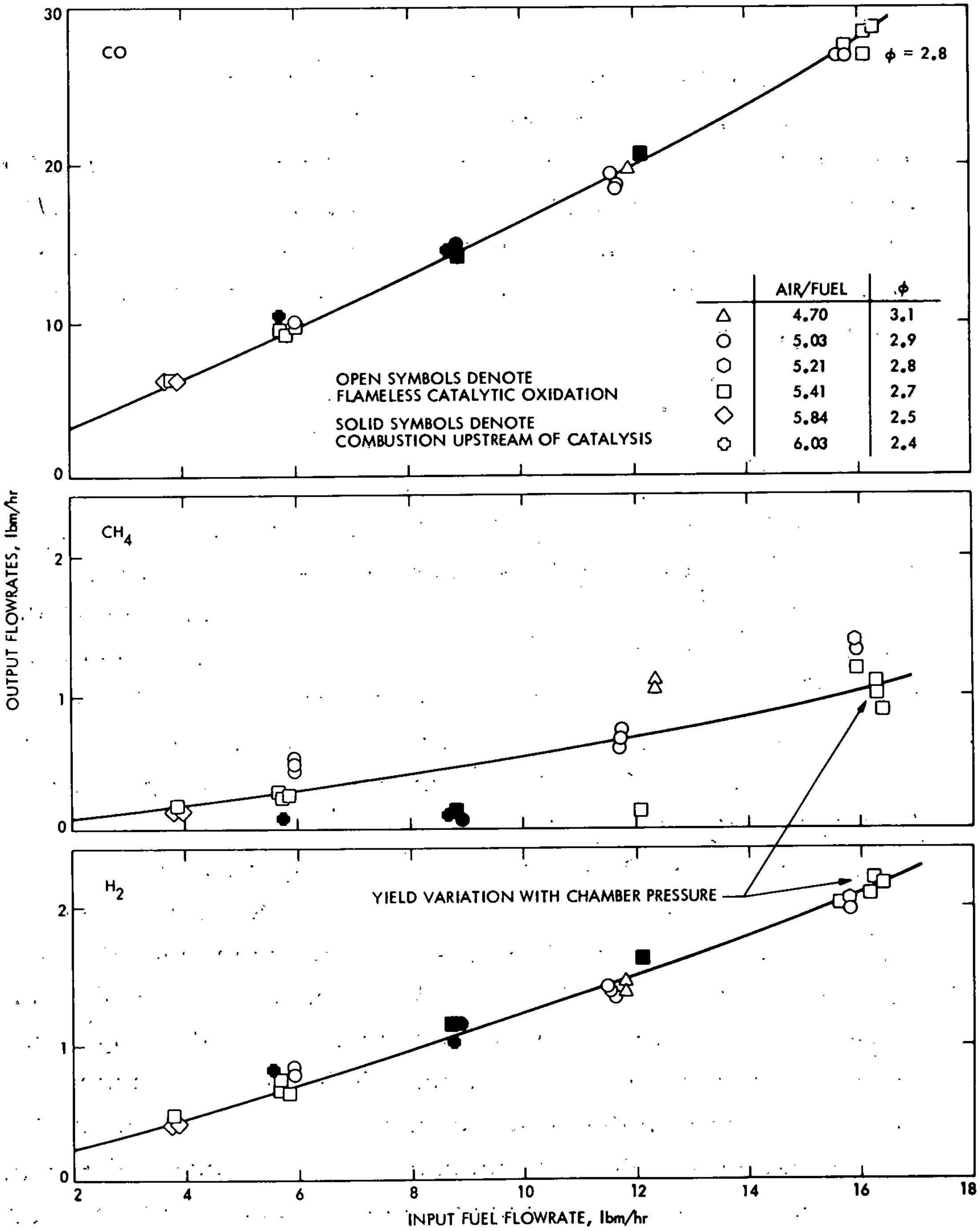


Figure 3-10. Product output flow rates for the JPL gas generator (Ref. 3-10)

indolene containing 3-gm Pb/gal. This surprising result may be due to the relatively high operating temperature (1800°F) of the catalyst. Shrinkage and settling of the catalyst pellets in the reactor were experienced in early work at JPL, but subsequent operation with vertical downward flow in the reactor seems to have eliminated this problem.

Tests with the catalytic hydrogen generator at equivalence ratios of 3 or slightly lower showed close to equilibrium yields of hydrogen with only a trace[1] of soot (Figure 3-11) once the reformer reached the

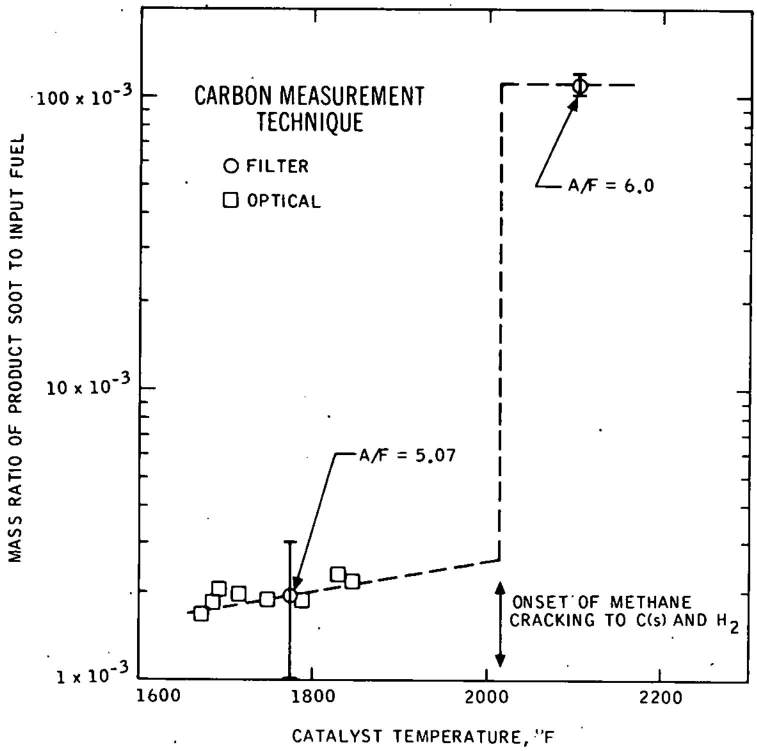


Figure 3-11. Measured soot content in the JPL gas generator product stream (Ref. 3-10)

---

[1] A soot-mass ratio (soot-to-input gasoline) of 2×10^{-3} results in a particulate emission of 0.3 gm/mi for a gasoline flow of 9 lb/hr to the reformer. Particulate emission from a car using leaded fuel is currently about 0.2 gm/mi.

operating temperature of 1800°F. The use of a catalyst and vaporization of the gasoline before injection into the reactor was necessary to achieve this soot-free operation at A/F = 5. During the warmup period before it is possible to vaporize the fuel, the reactor was run at A/F = 9 to avoid soot formation. No information is available concerning soot problems during transient operation of the hydrogen generator. All of the JPL tests to date have involved steady-state operation.

3.2.3 International Materials Corporation Concept

The design and development of an experimental vehicle operating on gaseous fuel (primarily hydrogen) obtained by reforming onboard the vehicle gasoline or some other liquid hydrocarbon was undertaken by the International Materials Corporation (Ref. 3-11). The vehicle was called the "Boston Reformed Fuel Car." A significant part of the International Materials program dealt with the design of an onboard fuel reformer for generating hydrogen; the results are pertinent to the present study. The reformer approach taken by International Materials was thermal steam reforming (no catalyst) by a two-step process, (1) partial reforming to CO and H_2 and (2) CO shifting to H_2 and CO_2, with part of the product gas supplying the heat required by the reforming processes. Much of the work done by International Materials involved conceptual designs and engineering calculations with only limited actual test data being reported.

3.2.3.1 Description of the Reformer System and Hardware

A description of the reformer approach taken by the International Materials Corporation is given in Refs. 3-11 and 3-12. A sketch of a reformer utilizing that general approach is shown in Figure 3-12. The steam reforming of the liquid hydrocarbon fuel is performed in two steps, and the hardware is arranged accordingly. The first step, converting the hydrocarbon fuel to carbon monoxide and hydrogen, is done in a cylindrically wound tube-type reactor heated by hot combustion gases from a burner fueled with product gases from the reformer. Water is vaporized

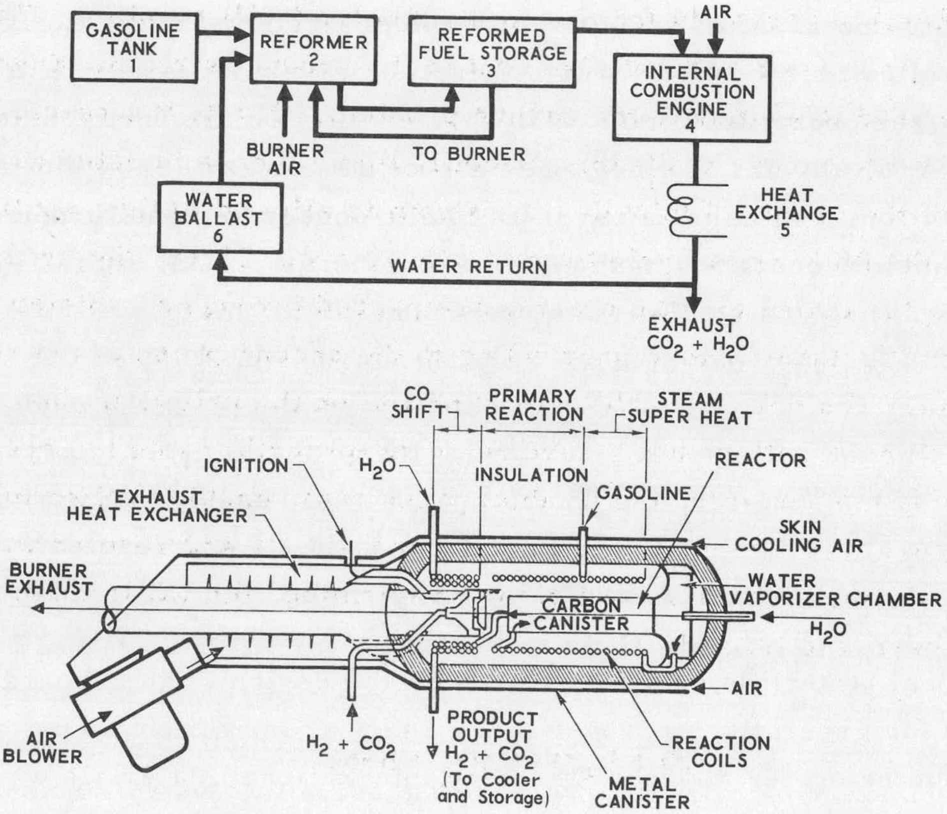


Figure 3-12. International Materials fuel reformer (Ref. 3-11)

in a chamber heated by the burner gases and is superheated in the tubular reactor before mixing with vaporized hydrocarbon fuel. The chemical reactions in the first phase of the process are

$$C_7H_{16} + 7\,H_2O \rightarrow 7\,CO + 15\,H_2 \text{ - Heat}$$

$$C_7\,H_{16} \rightarrow 7\,C + 8\,H_2 \text{ - Heat}$$

$$C + H_2O \rightarrow CO + H_2 \text{ - Heat}$$

The first phase of the fuel reforming takes place at a temperature of 2200°F and a pressure of 800 psi. It is clear from the reaction mechanisms cited that considerable carbon is formed in the tubular (coil) reactor. This can lead to coking in the tubes and also soot in the product stream. Several means are used to eliminate the carbon problem. First, the reactor is run with a large excess of steam. Second, liquid fuel is injected during only part of the cycle. Typically, the steam is allowed to flow six times as long as the fuel with a cycle time of 6 to 10 sec. Third, the product gases from the coiled tubular reactor are passed through a canister containing alumina beads before proceeding to the second phase of the reforming process. The coke particles are captured on the pellets in the canister and reacted there with steam before being swept farther into the system. After leaving the canister, the product gases enter another coiled tube-type reactor where they are cooled by the burner inlet air and reacted with additional steam. In this second step of reforming, the CO in the product gas is converted to H_2 and CO_2:

$$CO + H_2O \rightarrow CO_2 + \text{heat}$$

In order for nearly all the CO to be shifted, the temperature must be less than 800°F. The product gases after the complete reforming process contain primarily H_2 and CO_2, but significant concentrations of CH_4 may also be present. Since a large excess of steam is used to control coking in the process, most of the water in the product stream is condensed out and recycled to the reformer. A block diagram of the complete reformer process is given in Figure 3-12.

Several experimental versions of the reformer have been built and tested in the laboratory by International Materials, but detailed information concerning the tests is not available in the literature. Since the reformer approach taken involves high temperatures and moderately high pressures, the materials problems associated with this type of design are severe.

3.2.3.2 Product Gas Composition and Thermal Efficiency

Experimental data on the composition of the product stream from the International Materials Corporation reformer are not given in Ref. 3-11. Most of the information given in Ref. 3-11 was inferred from the stoichiometry of the simple steam reforming reaction

$$C_7H_{13.44} + 14\,H_2O \rightarrow 7\,CO_2 + 20.72\,H_2$$

with the possibility of CO and CH_4 in the product stream being ignored. The final composition of the product gas after cooling and CO shifting is determined by equilibrium between two interacting reactions:

$$CO + H_2O \rightleftarrows CO_2 + H_2 + \text{heat}$$

$$CO + 3\,H_2 \rightleftarrows CH_4 + H_2O + \text{heat}$$

Thus, significant concentrations of both CO and CH_4 are likely to be found in the product gas. The hydrogen yield (lb H_2/lb liquid fuel) can be expressed as

$$\frac{\text{lb } H_2}{\text{lb liquid fuel}} = \frac{20.72\ (2/98)}{1 + 4r_{CH_4} + r_{CO}}$$

where r_{CH_4} and r_{CO} are the ratios of the molefractions of CH_4 and CO, respectively, to the molefraction of H_2. Chemical equilibrium calculations for poststeam reforming gas mixtures indicate that for the CO shifting conditions of the International Materials reformer (1000°F, 800 psi), the mole fractions of the fuel gas components[2] are 1.25% CO, 15% H_2, and 26% CH_4.

---

[2] A steam-to-carbon ratio (molar) of 2.5 is assumed.

The resulting hydrogen yield of only 0.05 compares with 0.42 if it is assumed that the concentrations of CO and CH_4 in the product gas are negligible. The primary reason for the high CH_4 concentration is the high pressure at which the reformer operated. If the CO shifting takes place at 40 psi, 1300°F, then the equilibrium fuel gas component concentrations are 10% CO, 50% H_2, and 4% CH_4, resulting in a hydrogen yield of 0.28. Hence, it is clear that the hydrogen yield from a thermal equilibrium-type reformer depends critically on the final pressure and temperature of the process; simple stoichiometry neglecting CO and CH_4 is not adequate for predicting the product gas composition.

A simple analysis of the reformer thermal efficiency is given in Ref. 3-11. It was concluded that the thermal efficiency is about 60%, based on the chemical heat content of the reformer product stream, since about 40% of the product gas is needed to supply the heat needed in the process itself. A more detailed thermal analysis of the reformer system than that given in Ref. 3-11 is needed for two reasons: (1) The chemical heat content (heat of combustion) of the product stream depends on the composition, particularly the concentration of H_2 and CH_4, and simple stoichiometry is not adequate for determining these concentrations; (2) the amount of process heat which must be produced with the product fuel gas depends on the way in which waste heat from the reformer itself and the vehicle engine are used. For example, the heats of combustion of the product stream are 1050, 1199, and 1270 kcal/mole liquid fuel for the cases of hydrogen yield of 0.05, 0.28, and 0.42, respectively. The heat of combustion of the liquid fuel (Indolene) is 1050 kcal/mole. Assuming that all the process heat (about 500 kcal/mole fuel) is obtained by burning product fuel gas, the thermal efficiency, based on the chemical energy content of the liquid fuel, corresponding to the three product stream H_2 concentrations are 51%, 66%, and 73%, respectively. If, because of better use of waste heat from the reformer and engine, only 250 kcal/mole fuel of process heat is required from burning product fuel gas, the corresponding thermal efficiency values become 75%, 90%, and 97%, respectively. Hence, depending on the conditions under which the steam reforming takes place and the way in which waste heat is used, the thermal efficiency can vary from 50% to nearly 100%.

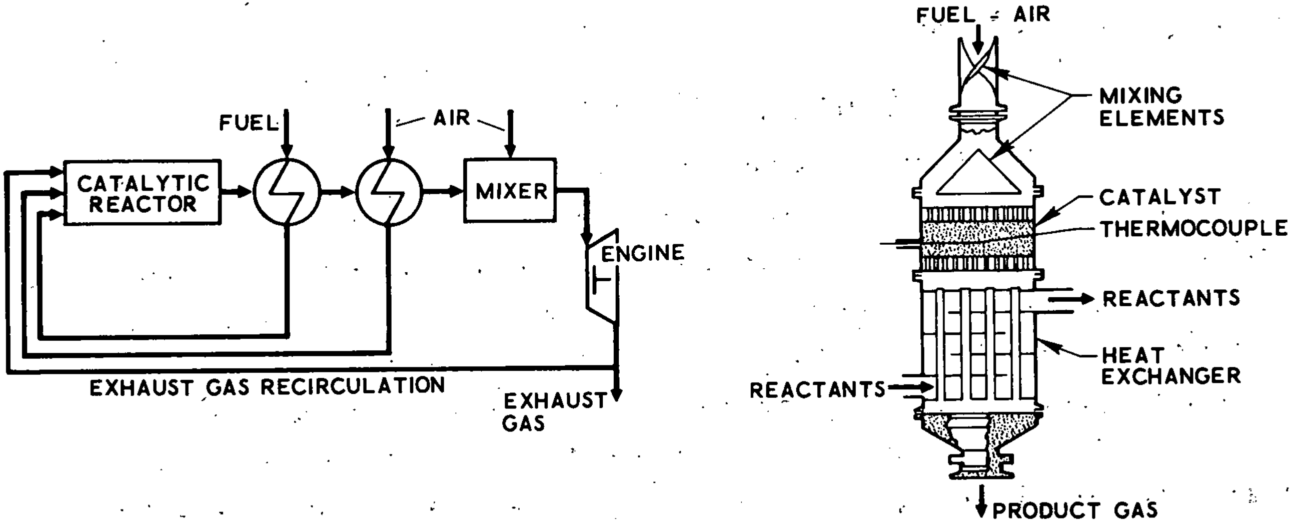
3.2.4 Siemens Corporation Concept

The Siemens Corporation in Germany has been developing over the last several years what it terms a catalytic carburetor to be used with the conventional internal combustion engine (Refs. 3-13 and 3-14). The catalytic carburetor system converts liquid hydrocarbon fuel such as gasoline to a fuel gas (CO, CH_4, and H_2), which is then mixed with air and burnt in the spark ignition engine. The Siemens gas generator was not designed to generate hydrogen preferentially but rather to produce a fuel gas at as high a thermal efficiency as possible. Nonetheless, the Siemens gas generator work is relevant to the present study.

Siemens has taken the partial oxidation approach with a catalyst, but its work differs in several important respects from the JPL work discussed previously. First, Siemens has by design limited the residence time of the reactants in the reformer so that chemical equilibrium is not approached. In this way, it has been able to attain soot-free operation at equivalence ratios as high as 10 ($A/F \approx 1.5$) compared with about 3 for JPL. Secondly, Siemens has used engine EGR as a means of adding small, but significant, amounts of water to the mixture being processed by the gas generator. The presence of the water permits some of the endothermal steam reforming reactions to take place and that combined with the low air-fuel ratio yields a reformer thermal efficiency much higher than is possible with the JPL equilibrium approach to catalytic partial oxidation fuel reforming.

3.2.4.1 Description of the Reformer System and Operating Conditions

The Siemens reformer system (Figure 3-13) utilizes partial oxidation with a catalyst. Air and fuel are preheated and mixed with



PROCESS SCHEMATIC OF AUTOTHERMIC FUEL GASIFICATION BY PARTIAL OXIDATION

CATALYTIC GAS GENERATOR

Figure 3-13. Siemens Corporation concept (Ref. 3-13)

recirculated engine exhaust gases before they are fed to the catalytic reactor. Since by design the reactor operates at conditions far from chemical equilibrium, the composition of the product stream depends critically on the kinetics of the reactions occurring in the reactor. Thus, the residence time or the contact time of the reactants with the catalyst becomes an important design parameter, in addition to temperature and equivalence ratio. The reactor contact time is usually expressed in terms of the space velocity v_{sp}, which is defined as

$$v_{sp} = \text{space velocity} = \frac{\text{volume flow of reactants/unit time}}{\text{volume of catalyst}}$$

In the Siemens work (Ref. 3-14), the space velocity is defined in terms of the gasoline processed per unit time divided by the volume of catalyst used. If the amount of gasoline processed is given as the weight, the units of the space velocity parameters are kg/ℓ hr (ℓ = liter). The design parameters for the Siemens system are then v_{sp}, T, and ϕ.[3] The system operates at slightly below atmospheric pressure.

The sequence of reactions taking place in the reactor involves cracking of the complex hydrocarbon fuel (e.g., C_7H_{16}) into simpler hydrocarbon molecules and their partial oxidation to CO and H_2. This process is illustrated in Figure 3-14. The cracking number is roughly proportional

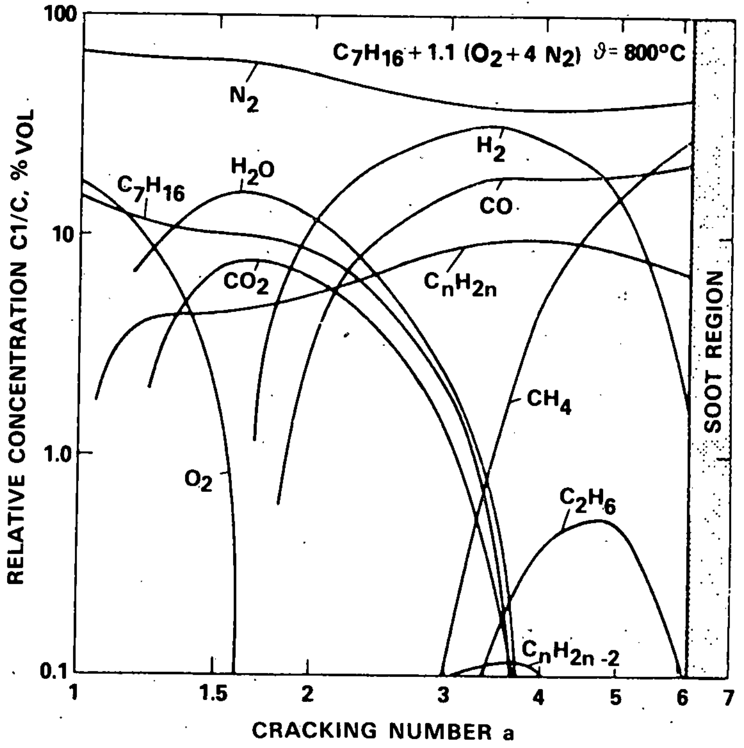


Figure 3-14. Gas composition vs cracking number (Ref. 3-13)

---

[3] As is common in Europe, Siemens uses λ equal to $1/\phi$ rather than ϕ. Rich operation results in $\lambda < 1$.

to residence time. Soot formation does not occur until the cracking and partial oxidation steps are completed. The Siemens catalytic reactors operate in the nonequilibrium region to the left of the soot formation line. Hence, if the residence time is selected properly, catalytic reactors can be operated soot-free at equivalence ratios of 10 to 20 ($\lambda = 0.05 - 0.1$), which is far into the soot region for equilibrium chemistry (Figure 3-8). Successful operation of the reactor also requires use of catalysts which have the desired conversion characteristics. Siemens has developed such catalysts, but their chemical composition and method of preparation are proprietary information.

Experimental data on the conversion characteristics of the Siemens catalysts are shown in Figures 3-15 and 3-16 for temperatures between 1100 and 1500°F and very fuel-rich operation ($\lambda = 0.05, 0.1$). As expected, the fraction of the fuel reacted (m/m_0) increases with temperature and residence time (smaller values of kg/ℓ hr). From Figure 3-15, the size of a catalytic reactor for a given application can be computed. For example, for the case of 15-lb gasoline/hr (2.5 gal/hr) and operation at 1500°F (800°C), 15 in.^{3} of catalyst is required for 90% conversion at an equivalence ratio of 10 ($\lambda = 0.1$). This catalyst volume is about one-fifth that used by JPL in their equilibrium catalytic reactor operating at an equivalence ratio of about 3.

In principle, the Siemens system is relatively simple in that it requires only control of the reactor air-fuel ratio and the volume of recirculated engine exhaust gases. This control should be possible with presently available automotive components (advanced carburetor systems and EGR valves). However, since the conversion fraction and product stream composition depend strongly on residence time (i.e., gasoline flow rate), transient effects could be more important and more difficult to control in the Siemens reactor than in some of the other types of fuel reformers which are less sensitive to residence time in the reactor.

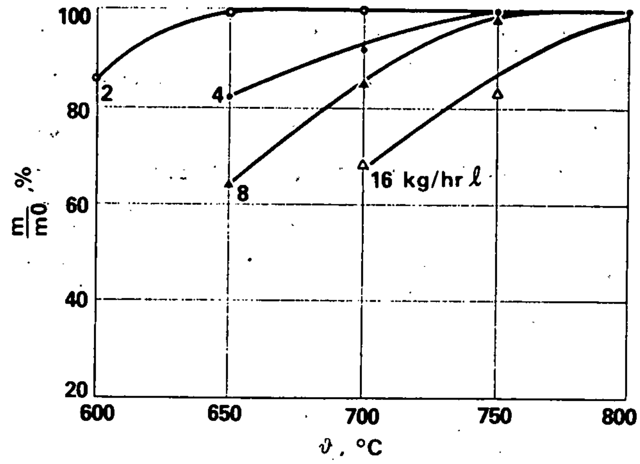


Figure 3-15. Conversion of n-heptane vs reaction temperature and different loads (heptane throughput in kg/hr ℓ per catalyst volume in liters [ℓ]): (a) λ = 0.1; (b) λ = 0.05 (Ref. 3-14)

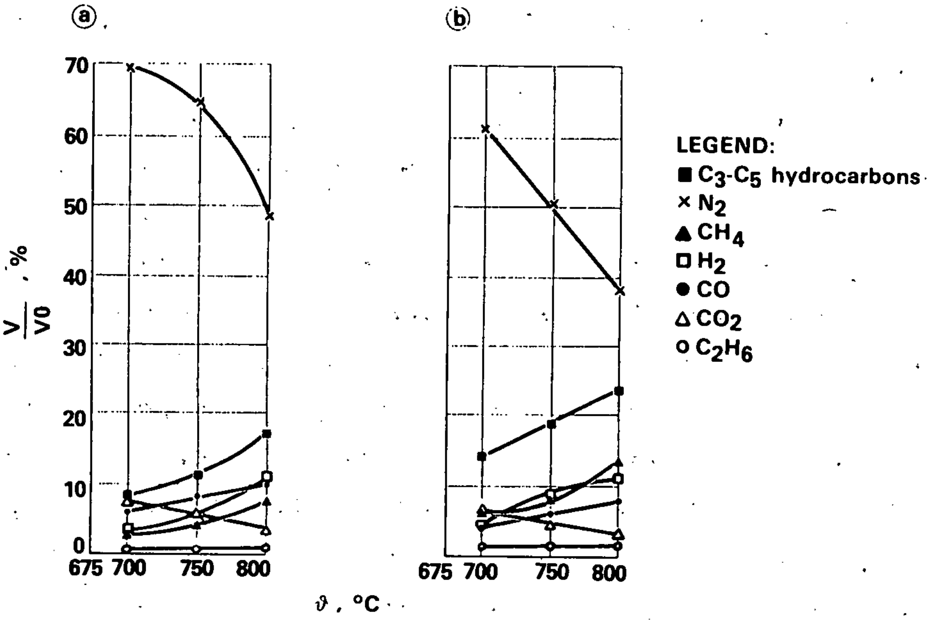


Figure 3-16. Gas composition as a function of the reaction temperature at a load of 16 kg/hr ℓ: (a) λ = 0.1; (b) λ = 0.05 (Ref. 3-14)

3.2.4.2 Product Composition and Thermal Efficiency

Experimental data are available on the composition of the product stream from the Siemens fuel reformer (Ref. 3-13). As shown in Figure 3-17, the concentration of the fuel gas components (CO, H_2, and CH_4) are relatively insensitive to the fraction of the engine exhaust gases recirculated when the fraction is greater than 20%. In the present application, the hydrogen yield is of prime importance. Using a simple mass balance of carbon, oxygen, and hydrogen atoms, it can be shown that the hydrogen yield is given by

$$\frac{\text{lb } H_2}{\text{lb gasoline}} = \frac{21(1 - \lambda)\,(2/98)}{1 + 4r_{CH_4} + r_{CO}}$$

where, as before, r_{CH_4} and r_{CO} are the ratio of the mole fractions of CH_4 and CO, respectively, to the molefraction of H_2 in the product stream. The effect of the unoxidized portion of the fuel has been neglected. For the case shown in Figure 3-17, CO ~ 12.75%, H_2 ~ 12.75%, and CH_4 ~ 4%, the calculated hydrogen yield is 0.118-lb H_2/lb gasoline (C_7H_{14}). This is essentially the same as that of the JPL gas generator. The CO yields of the Siemens and JPL reformers are also about the same, 1.6 (lb CO/lb gasoline). Presumably, both reformer designs will operate soot-free. The coking characteristics of the Siemens catalyst are shown in Figure 3-18. Apparently, the coking is significant during the first 5 to 10 hr of operation, but the situation stabilizes for additional operation up to at least 300 hr.

In the partial oxidation process, most of the required process heat comes from the exothermic reactions taking place within the process. The thermal efficiency for a partial oxidation process depends primarily on the equivalence ratio of the process and is given approximately by $\eta_{th} \approx 1 - \lambda$ for λ up to 0.3 (Ref. 3-14). If water is added to the reactants, higher thermal efficiency is possible because the presence of the water

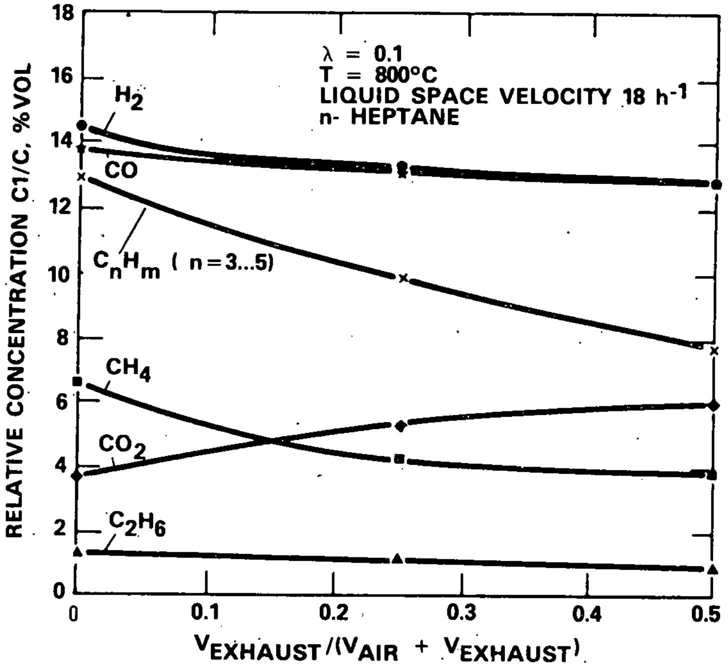


Figure 3-17. Gas composition vs gas fraction of the oxidant (Ref. 3-13)

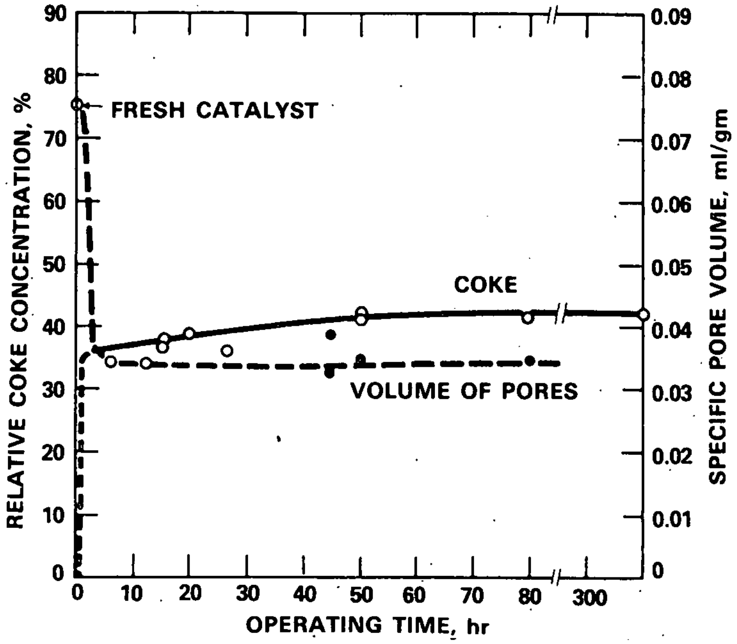


Figure 3-18. Relative coke concentration and specific pore volume vs operating time (Ref. 3-13)

permits the endothermic stream reforming reaction to occur in the reactor. Hence, one can expect the thermal efficiency of the Siemens process using $\lambda = 0.1$ and EGR to be significantly higher than that of the JPL process using $\lambda = 0.3$ and no water addition. That this is the case can be shown by calculating the heat of combustion of the fuel gases in the product stream from the Siemens and JPL reformers. Such a calculation shows that the heat of combustion of the product stream from the Siemens reformer is ~1000 kcal/gm-mole gasoline while the corresponding value for the JPL reformer is ~850 kcal/gm-mole gasoline. The primary reason for this difference is that the stream from the Siemens reformer contains much more CH_4 than that from the JPL reactor. The thermal efficiency of the Siemens process is then about 95%, while that of the JPL process is about 80%. For a partial oxidation process, the system thermal efficiency is not strongly dependent on the way in which the waste heat from the reformer and engine processes are used. As stated previously, however, thermal efficiency is somewhat dependent on the use of some water with the reactants.

3.2.5 Phillips Petroleum Company Concept

The Phillips Petroleum Company has proposed a fuel reformer to convert liquid fuel to a fuel gas as a means of reducing exhaust emissions from a spark ignition engine (Ref. 3-15). The approach taken is that of catalytic steam reforming, using the engine waste heat to supply the process heat needed. The system involves primarily a conceptual design of the gas generator and limited single-cylinder emissions testing using synthetic reformer fuel gas. The Phillips work is discussed briefly in this report because it illustrates the use of the moderate temperature (1000°F) catalytic steam reforming approach to generate hydrogen onboard a vehicle.

3.2.5.1 Description of the System and Operating Conditions

A schematic drawing of the proposed Phillips fuel reformer system is given in Figure 3-19. The main special feature of the system is the use of a catalyst at a sufficiently low temperature (1000°F) that the process heat needed to drive the steam reforming reactions can be obtained entirely from the engine exhaust gases. Such use of engine waste heat yields a reformer system thermal efficiency of greater than 100%.

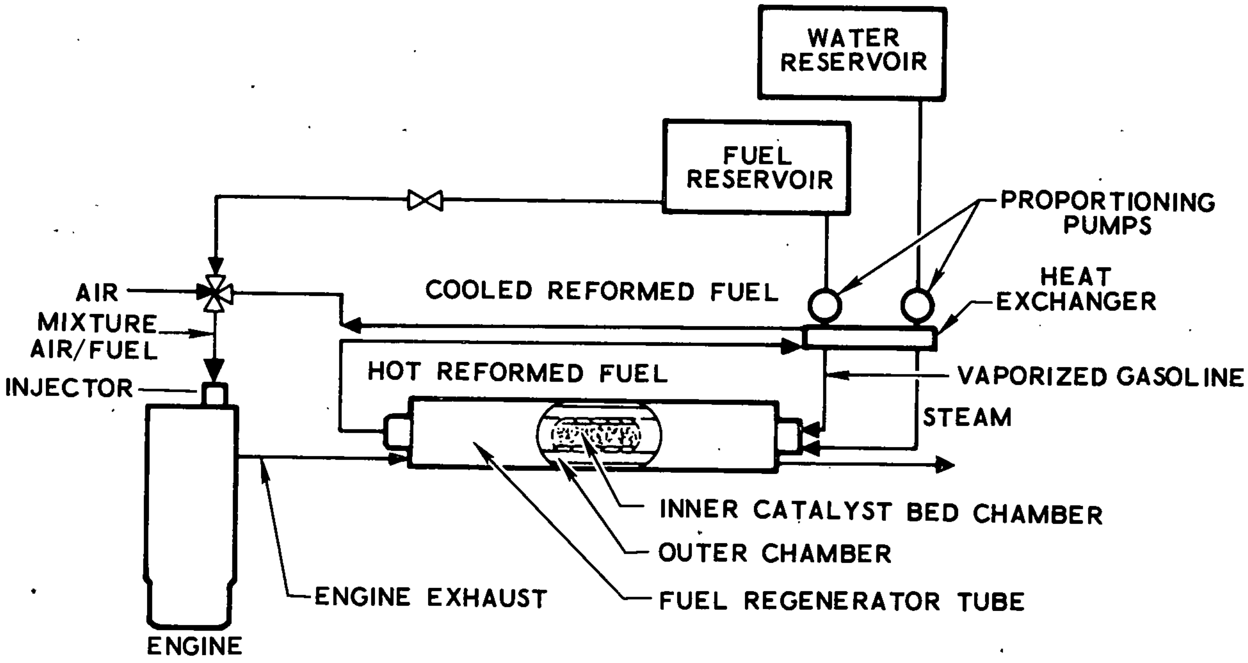


Figure 3-19. Phillips Petroleum fuel reformer (Ref. 3-15)

Proper selection of the catalyst is necessary to attain practical liquid fuel processing rates and to avoid soot formation and coking problems. These aspects of the reformer development were not pursued experimentally by Phillips and are critical to the success of its proposed design. Sooting and coking problems are of particular importance in automotive applications. In order to minimize sooting problems, steam reforming requires an excess of steam; i.e., the steam-to-carbon molar ratio of the reactants is greater than one. This ratio is usually between 2 and 3. Chemicals (such as alkalis) which catalyze the steam oxidation of carbon can be added to the standard nickel steam reforming catalysts, and the steam-to-carbon ratio thus can be reduced to about 1.5. The coking problem and catalyst selection are discussed in detail in Ref. 3-16.

3.2.5.2 Product Composition and Thermal Efficiency

As discussed previously in connection with other fuel reformer approaches, the hydrogen yield and thermal efficiency of the reformer are of prime interest for automotive applications. Measured product stream concentrations of the fuel gas components are not given in Ref. 3-15. Since the intent of the reformer design is to generate a product stream in chemical equilibrium, product concentrations (assuming chemical equilibrium is attained in the reactor) are used in the following considerations. For a reformer operating at 1000°F, 40 psi, and steam-to-carbon ratio (molar) of 1.5, the equilibrium product concentrations (Ref. 3-17) are ~4% CO, ~30% H_2, and ~23% CH_4. The resulting hydrogen yield is

$$\frac{\text{lb } H_2}{\text{lb gasoline}} = 0.102$$

The hydrogen yield can be increased by using more excess steam, by reducing the pressure, and by increasing the operating temperature of the reformer. In general, for a maximum hydrogen yield, the methane (CH_4) concentration in the product stream should be kept to a minimum. Increasing the reformer temperature makes it more difficult to obtain all the process heat from the engine exhaust gases, and using a greater excess of steam means that a larger quantity of water must be condensed from the product stream before injection of the fuel gas into the engine induction system. Hence, it may not be practical to attain hydrogen yields much greater than 0.1 for automotive applications.

The thermal efficiency of the reformer system follows directly from the heat of combustion of the product gases. For the product fuel gas concentration cited previously for the proposed Phillips reformer, the lower heat of combustion of the product stream is 1066 kcal/mole gasoline. Taking the heat of combustion of the gasoline (indolene) as 1050 kcal/mole, the thermal efficiency of the reformer is 101.5%. Hence, with a moderate-temperature catalytic reformer, it is possible to increase the heating value of the fuel. This, of course, is done at the expense of considerable additional fuel system complexity.

3.2.6 University of Arizona Studies

A study of catalytic steam reforming of gasoline (hexane) for automotive applications was conducted at the University of Arizona (Ref. 3-17). The study included equilibrium chemistry calculations of reformer product gas concentrations for a range of temperature, pressure, and steam-to-carbon ratio values. A laboratory reactor was built and tested, using commercially available steam reforming catalysts. Measurements of product stream gas concentrations indicated that chemical equilibrium was approached in the reactor for practical catalyst volumes, with the product concentration agreeing well with the theoretical calculations. Carbon formation occurred during the experimental runs, with the results indicating that sooting and coking problems were potentially serious for such catalytic reforming reactors.

For purposes of easy reference, selected equilibrium composition results (Figures 3-20 to 3-22) taken from Ref. 3-17 are included with this report. The composition plots clearly show the composition trends with temperature, pressure, and steam-to-carbon ratio cited previously in connection with the International Materials and Phillips Petroleum catalytic steam fuel reformers.

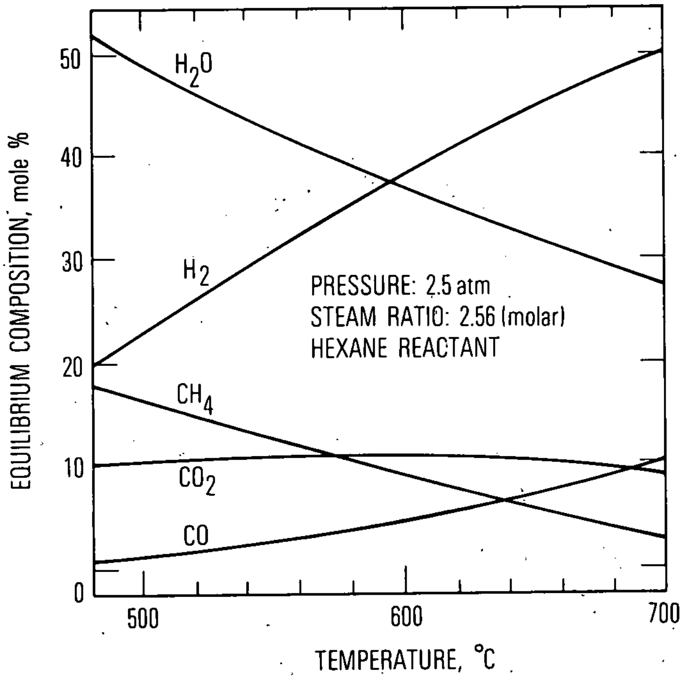


Figure 3-20. Equilibrium composition for steam reforming of hexane: effect of varying temperature (Ref. 3-17)

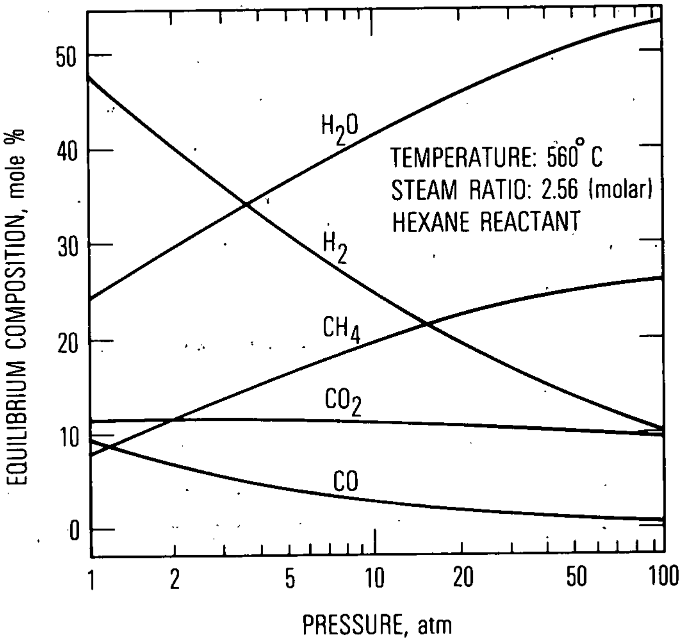


Figure 3-21. Equilibrium composition for steam reforming of hexane: effect of varying pressure (Ref. 3-17)

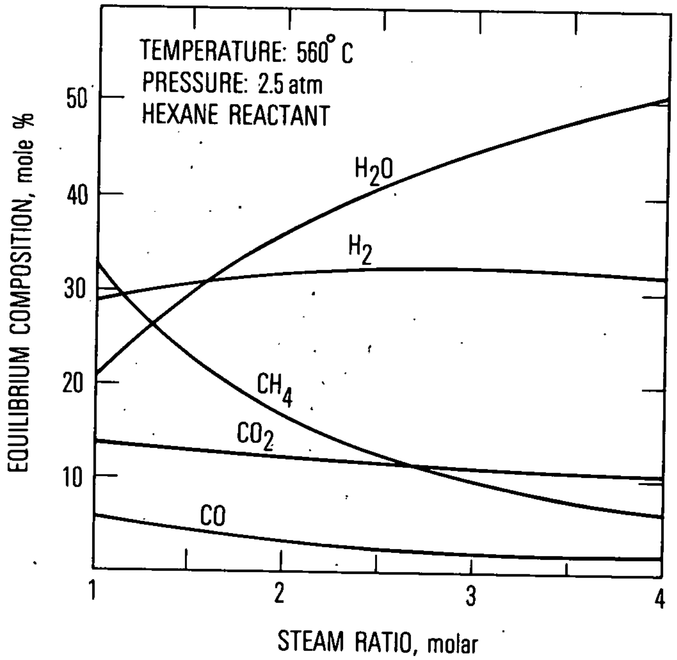


Figure 3-22. Equilibrium composition for steam reforming of hexane: effect of varying steam ratio (Ref. 3-17)

REFERENCES FOR SECTION 3

3-1. F. W. Hoehn and M. W. Dowdy, "Feasibility Demonstration of a Road Vehicle Fueled With Hydrogen-Enriched Gasoline," Paper presented at the 1974 Energy Conversion Conference, San Francisco.

3-2. R. F. Stebar and F. B. Parks, "Emission Control With Lean Operation Using Hydrogen-Supplemented Fuel," SAE Paper No. 740187, Detroit (March 1974).

3-3. R. E. Billings, "Hydrogen Storage in Automobiles Using Cryogenics and Metal Hydrides," Paper presented at the Hydrogen Economy Miami Energy Conference (March 1974).

3-4. C. H. Waide, J. J. Reilly, and R. H. Wiswall, "The Application of Metal Hydrides to Ground Transportation," Paper presented at the Hydrogen Economy Miami Energy Conference (March 1974).

3-5. W. S. Yu, E. Suuberg, and C. Waide, "Modeling Studies of Fixed-Bed Metal Hydride Storage Systems," Paper presented at the Hydrogen Economy Miami Energy Conference (March 1974).

3-6. Current Status of Alternative Automotive Power Systems and Fuels. Volume III: Alternative Nonpetroleum-Based Automotive Fuels, ATR-74(7325)-1, Vol. III, The Aerospace Corporation, El Segundo, Calif. (July 1974).

3-7. D. K. Nicholas and E. R. Ward, "The Production of Fuel Gases from Oil by Continuous Catalytic Reforming and Hydrogasification," *Proceedings of the Seventh World Petroleum Congress, Mexico City*, Elsevier Publishing Company, Amsterdam, pp. 157-173 (1967).

3-8. G. W. Bridger and G. C. Chinchen, "Hydrocarbon-Reforming Catalysts," *Catalyst Handbook*, Springer-Verlag, Berlin (1970).

3-9. J. Houseman and D. J. Cerini, Onboard Hydrogen Generator for a Partial Hydrogen Injection Internal Combustion Engine," SAE Paper No. 740600, SAE Meeting, Anaheim, Calif. (1974).

3-10. *JPL Final Report on EPA-Supported Studies Related to Ultralean Combustion with Hydrogen Addition*, Jet Propulsion Laboratory, Pasadena, Calif. (November 1974).

3-11. M. S. Newkirk and J. Abel, "The Boston Reformed Fuel Car," SAE Paper No. 720670 (November 1971).

3-12. M. S. Newkirk, "Method and Means for Generating Hydrogen and a Motive Source Incorporating Same," U. S. Patent Disclosure 3,682,142 (August 1972).

3-13. W. Frie et al., "Compact Gas Generator for the Economic Operation of Internal Combustion Engines with Low Pollutant Emissions," Paper presented at Second Symposium on Low Pollution Power System Development, Dusseldorf, Germany (November 1974).

3-14. H. J. Henkel et al., *Compact Gas Generator for Combustion with Low Level of Noxious Products*, Vol. 2, No. 1 (in English), Siemens Scientific Publications, Germany (1973).

3-15. H. M. Fox, "Method and Apparatus for Reducing Engine Exhaust Pollutants," U. S. Patent Disclosure No. 3,717,129, Phillips Petroleum Company (February 1973).

3-16. S. P. S. Andrew, *Catalysts and Catalytic Processes in the Steam Reforming of Naphtha*, Vol. 8, No. 3, I&EC Product Research and Development (September 1969).

3-17. G. A. Lorton, "Steam-Hydrocarbon Reforming for Lower Polluting Automotive Fuels," M.S. Thesis, Dept. of Chemical Engineering, University of Arizona (July 1973).

3-18. "Hydrogen Enrichment for Internal Combustion Engines," Briefing presented to *Advanced Automotive Power System*, Division of the Environmental Protection Agency, by Jet Propulsion Laboratory (3 December 1974).

SECTION 4

EFFECTS OF HYDROGEN ADDITION ON ENGINE AND VEHICLE CHARACTERISTICS

In this section, the effects of hydrogen addition on various operating characteristics of gasoline-fueled spark ignition engines and vehicles are analyzed and discussed. From a variety of sources, three types of data have been collected and compared: (1) CFR engine data, (2) vehicle and engine dynamometer data obtained using a standard V-8 engine and bottled hydrogen, and (3) engine dynamometer data obtained using a standard V-8 engine and a prototype fuel reformer to generate hydrogen. Vehicle road or chassis dynamometer data in which a prototype fuel reformer was integrated with the vehicle system are not yet available. Hence, the operating characteristics, e.g., fuel economy, emissions, and driveability, of the final integrated vehicle-reformer system are calculated or inferred from subsystem data rather than measured directly.

4.1 LEAN OPERATING LIMITS

Various studies have been done of lean operation of spark ignition engines using gasoline, gaseous fuels, and mixtures of gasoline and gaseous fuels, particularly hydrogen (Refs. 4-1 through 4-3). A prime objective of the studies has been to determine the minimum equivalence ratio at which the engine-fuel system being tested can operate satisfactorily. The minimum equivalence ratio is termed the lean (operating or misfire) limit of the system.

Much of the available lean limit data has been obtained from CFR engine systems. A cross section of such data for various fuels and fuel mixtures, presented in Figure 4-1, shows that a wide range of results have been found. This is due both to differences in the definition and method employed to determine the lean limit and to differences in the fuel system (fuel

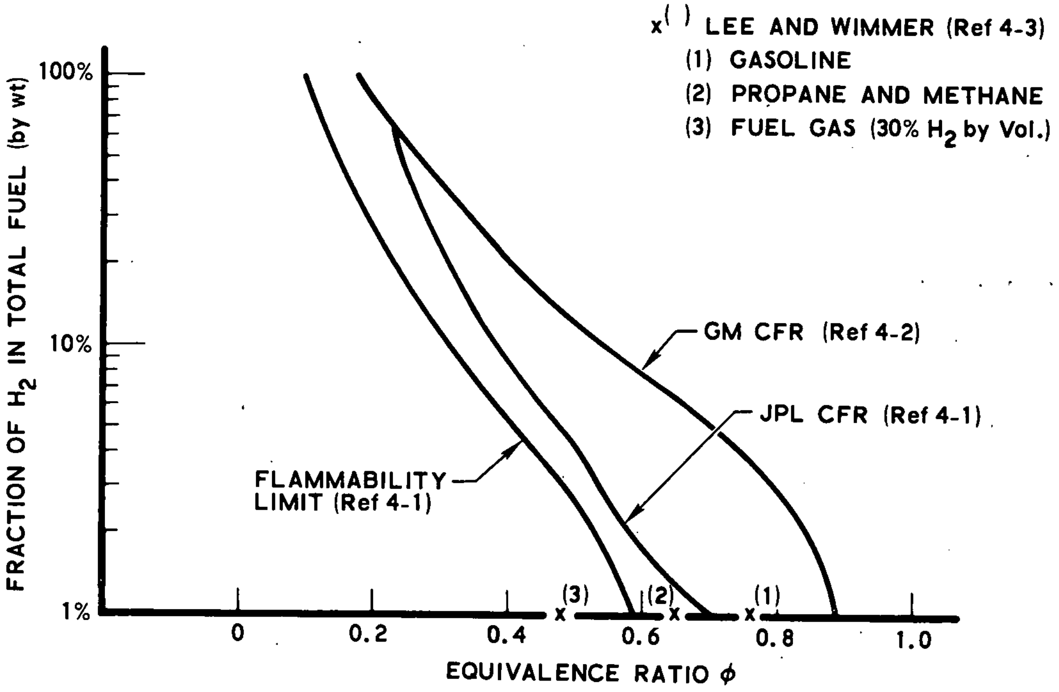


Figure 4-1. CFR lean limit data

metering, atomizing, vaporizing, mixing, etc.) used in the tests. Since the characteristics and uniformity of the fuel-air charge to the engine are particularly critical in lean limit studies, differences in the fuel systems used in the various studies are likely the primary reason for the variation in the results found. For example, in the GM studies (Ref. 4-2) a relatively standard manifold injection system was used, while in the JPL work (Ref. 4-1) a pneumatic atomizer was employed and special care was taken to provide a uniform, completely gaseous charge to the engine. Hence, the JPL CFR engine system operated much leaner on gasoline alone and required less hydrogen to operate at low equivalence ratios than the GM CFR system.

The large difference in the JPL and GM CFR engine lean limit results would make it difficult to estimate the hydrogen requirements for ultralean ($\phi \approx 0.5$) engine operation if that were the only data available. Fortunately, as shown in Figure 4-2, relevant V-8 engine data are available and can provide a much better estimate of hydrogen requirements. Both engine and chassis dynamometer tests have been made. In the GM chassis dynamometer tests (Ref. 4-2), the hydrogen was supplied from high pressure bottles, while for the JPL engine dynamometer tests (Refs. 4-4 and 4-5) hydrogen was generated from a fuel reformer and was supplied also from high pressure bottles. The results shown for bottled hydrogen are not at the lean limit but are probably reasonably close to it, while the curve for the V-8 engine and fuel reformer combination corresponds to the loci of maximum system thermal efficiency points for various engine and reformer operating conditions. From the viewpoint of maximum thermal efficiency, this later

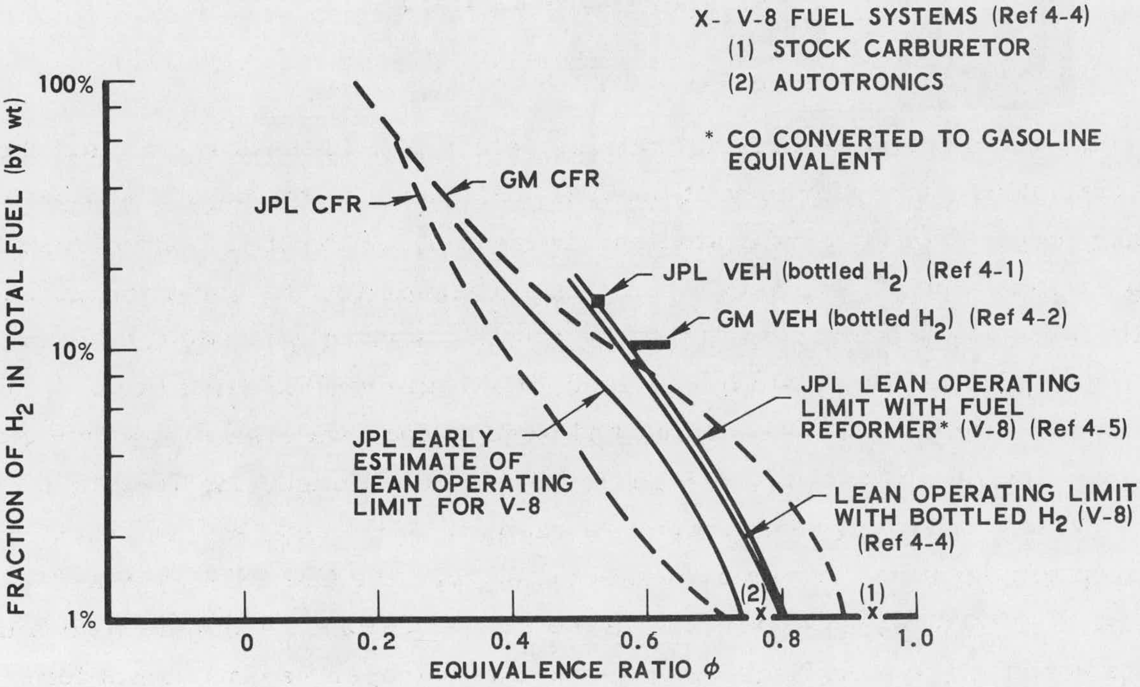


Figure 4-2. Lean operating limits for vehicles using hydrogen addition

definition is particuarly appropriate for determining the lean limit for vehicle applications.

The lean operating limit curves (fraction of H_2 vs ϕ) shown in Figure 4-2 for bottled hydrogen and fuel reformer systems are the current best estimates of the hydrogen requirements for vehicles utilizing hydrogen addition as a means of attaining ultralean combustion in spark ignition engines. The lean operating limit for gasoline alone depends on the fuel system used. For an advanced fuel system such as the Autotronics system (Figure 4-3) used by JPL, a V-8 engine can be operated on gasoline at equivalence ratios as low as 0.75 to 0.78. Less complex gasoline fuel systems result in a lean limit of 0.85 to 0.95.

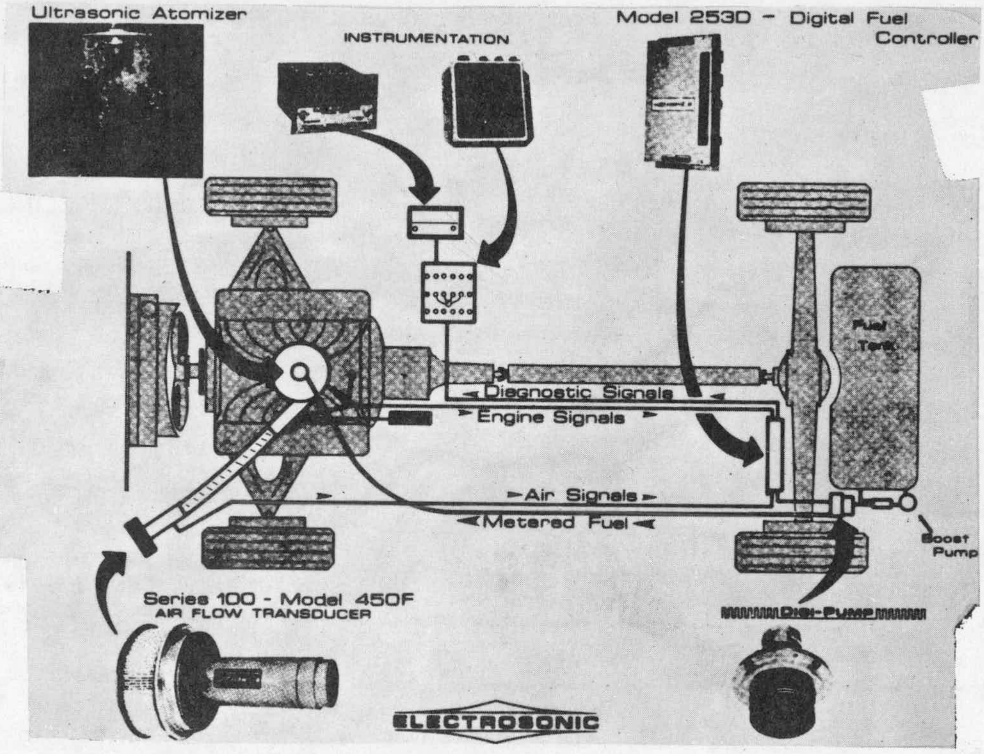


Figure 4-3. Autotronics fuel induction system (Ref. 4-1)

4.2 EMISSIONS

The effect of hydrogen addition on emissions is discussed in terms of three test configurations: (1) the CFR engine system using bottled hydrogen, (2) a V-8 engine using bottled hydrogen, and (3) a V-8 engine combined with a prototype fuel reformer (JPL).

4.2.1 Emissions from a CFR Engine

Measurements of exhaust emissions[4] from CFR engines with hydrogen addition have been made by both GM and JPL. The data reported by GM (Figure 4-4) are for CFR engine operation at the lean limit corresponding to a range of hydrogen weight fractions, while those reported by

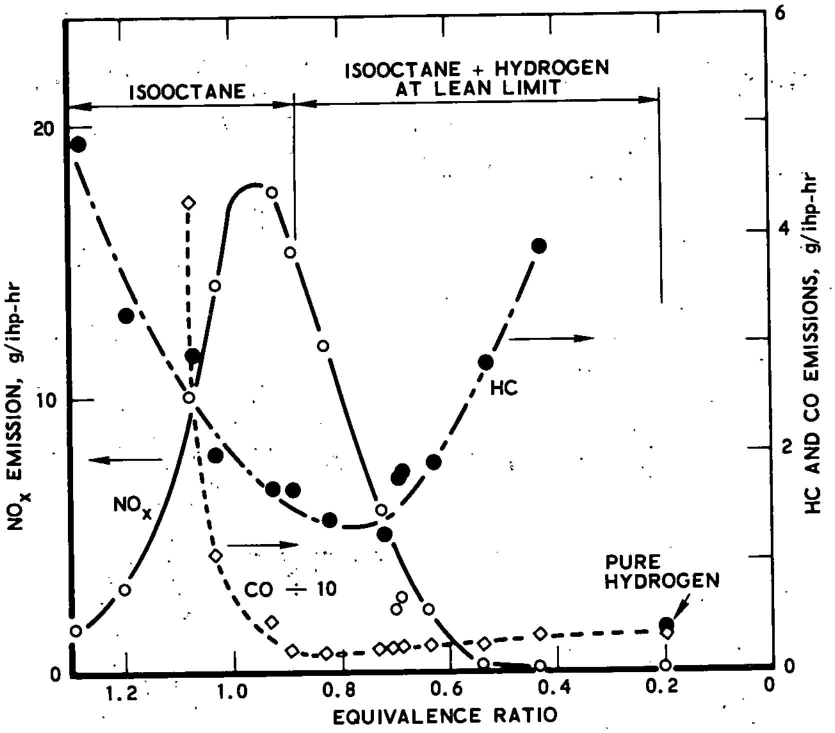


Figure 4-4. Emissions characteristics of isooctane-hydrogen fuel mixtures: GM CFR test data (Ref. 4-2)

---

[4] All CFR measurements taken at spark advance MBT (minimum for best torque).

JPL (Figures 4-5 to 4-7) are for a wide range of hydrogen, gasoline, and air flows representing engine operating conditions in the vicinity of the lean limit for a similar range of hydrogen weight fractions. From Figure 4-4, it is seen that the NO_x emissions (gm/ihp-hr) decrease rapidly as the lean limit is reduced, but that both HC and CO emissions increase as the engine is operated at progressively leaner conditions. The increase in HC emissions is particularly marked beyond the minimum.

The variations of CO and HC emissions with equivalence ratio for fixed weight fractions of hydrogen are shown in Figures 4-6 and 4-7. The lean limit for each hydrogen weight fraction is indicated on the curves. Even though the minimum CO and HC emissions attainable decrease with increasing hydrogen weight fraction, the CO and HC emissions at the lean

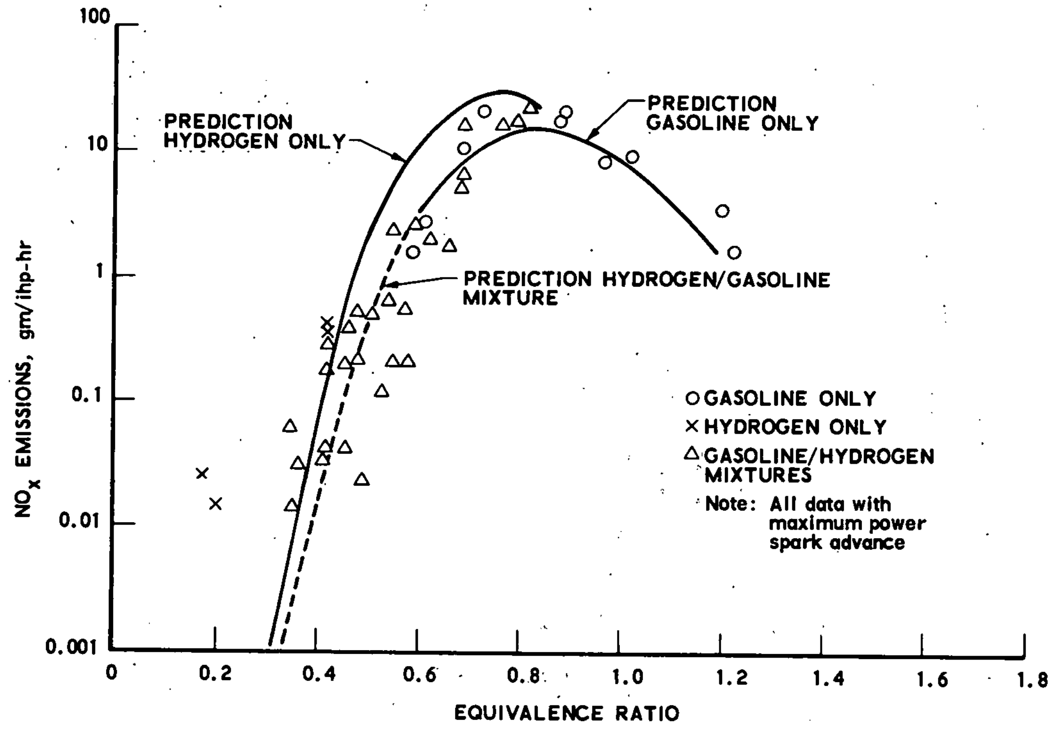


Figure 4-5. NO_x emissions: CFR engine data (Ref. 4-1)

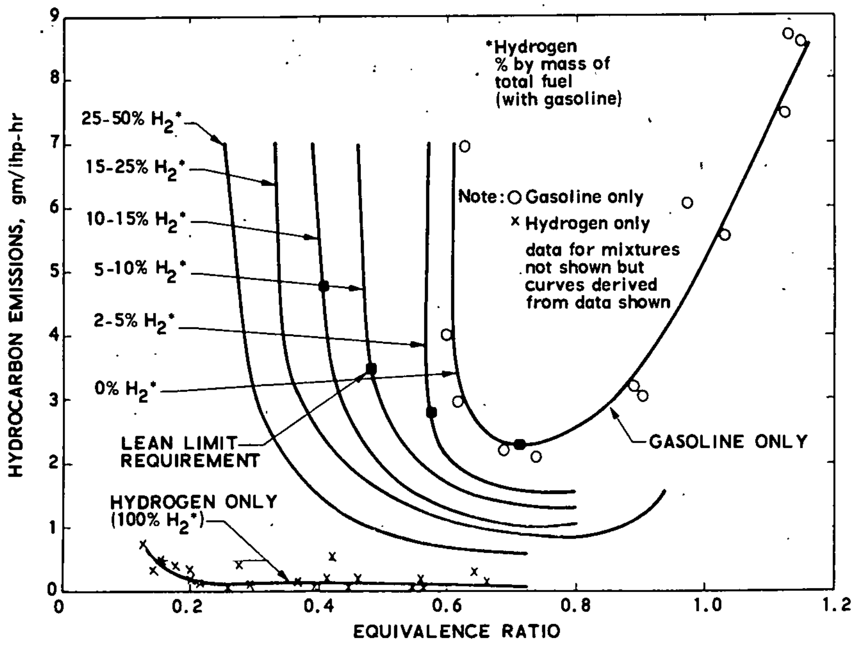


Figure 4-6. HC emissions: CFR engine data (Ref. 4-1)

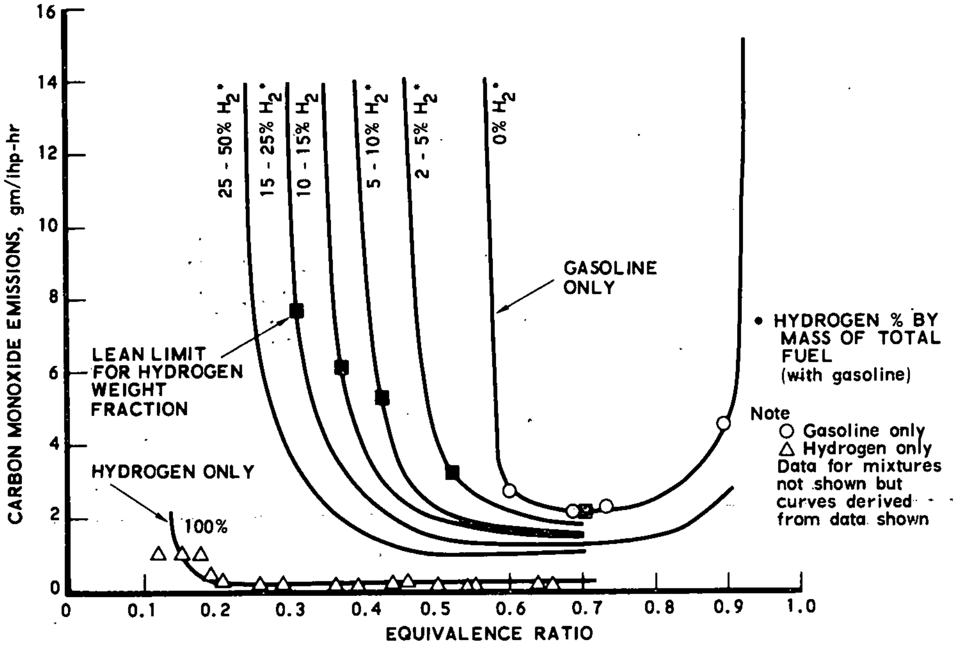


Figure 4-7. CO emissions: CFR engine data (Ref. 4-1)

limit for progressively greater hydrogen weight fractions increase significantly. This behavior results because, as shown in Figure 4-8 for $\phi = 0.55$ and 0.65, at a given lean equivalence ratio the CO and HC emissions increase as the hydrogen weight flow is reduced, with the maximum HC and CO emissions occurring at the minimum hydrogen weight fraction (the lean limit hydrogen requirement given in Figure 4-1) for which the engine operates satisfactorily. The minimum HC and CO emissions using gasoline alone as the fuel are also indicated in Figure 4-8. If a hydrogen weight fraction considerably greater than the minimum lean limit requirement is used, both the HC and CO emissions of the CFR engine at ultralean conditions can be reduced significantly below the minimum attainable using gasoline alone. Hence the high HC and CO emissions at ultralean conditions from hydrogen addition are not inherent to the approach, but can be avoided by use of hydrogen flow rates greater than the lean limit minimum.

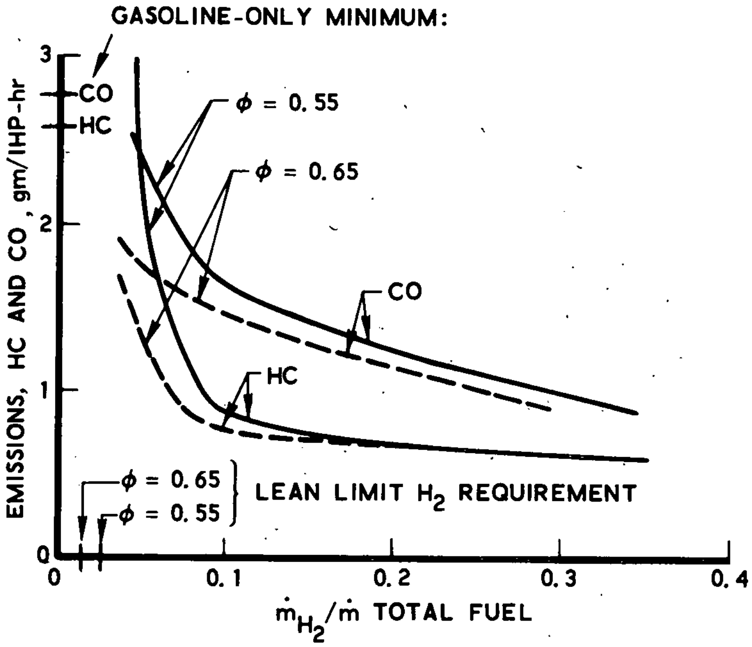


Figure 4-8. Effect of hydrogen weight fraction on HC and CO emissions: CFR engine operating at lean conditions

4.2.2 V-8 Engine Emissions from Bottled Hydrogen and Gasoline Mixtures

The first experimental studies of emissions from V-8 engines utilizing hydrogen addition as a means of attaining ultralean operation used bottled hydrogen. Such studies have been conducted by JPL and GM. In the GM studies (Ref. 4-2), a 1973 intermediate-size vehicle was modified[5] so that it operated on a mixture of gasoline and hydrogen; testing was done on a chassis dynamometer. The JPL studies included both chassis dynamometer tests of a 1973 standard-size vehicle modified to operate on gasoline-hydrogen mixtures and engine dynamometer tests of a V-8 engine using bottled gases to simulate the products from a fuel reformer (Ref. 4-6). The fuel system of the GM test vehicle (Figure 3-1 in Section 3.1) incorporated a gas mixer, which metered hydrogen proportional to the air flow, along with minor modifications of the standard four-barrel carburetor originally on the vehicle. The GM fuel system maintains a relatively constant hydrogen weight fraction (about 10%), and the operating equivalence ratio increased slowly with engine load and rpm (0.55 at idle to about 0.65 at high loads). In the JPL vehicle,[6] the standard carburetor system was replaced by the Autotronics fuel induction system (see Figure 4-3), with the hydrogen being metered as shown in Figure 4-9 by a flow control valve driven by a motor actuated by signals from the air flow meter. The JPL test vehicle idled on hydrogen alone, and the gasoline and hydrogen flows at loaded conditions were metered so that a hydrogen weight fraction of about 15% is maintained, permitting engine operation at $\Phi \approx 0.53$. When the hydrogen demand exceeded 3 lb H_2/hr or when full throttle was reached, the equivalence ratio was increased to 0.63 and finally to 0.94 at maximum acceleration conditions.

---

[5] EGR was removed, but spark timing was not changed.

[6] EGR was removed and sparking timing was advanced to 50° BTDC.

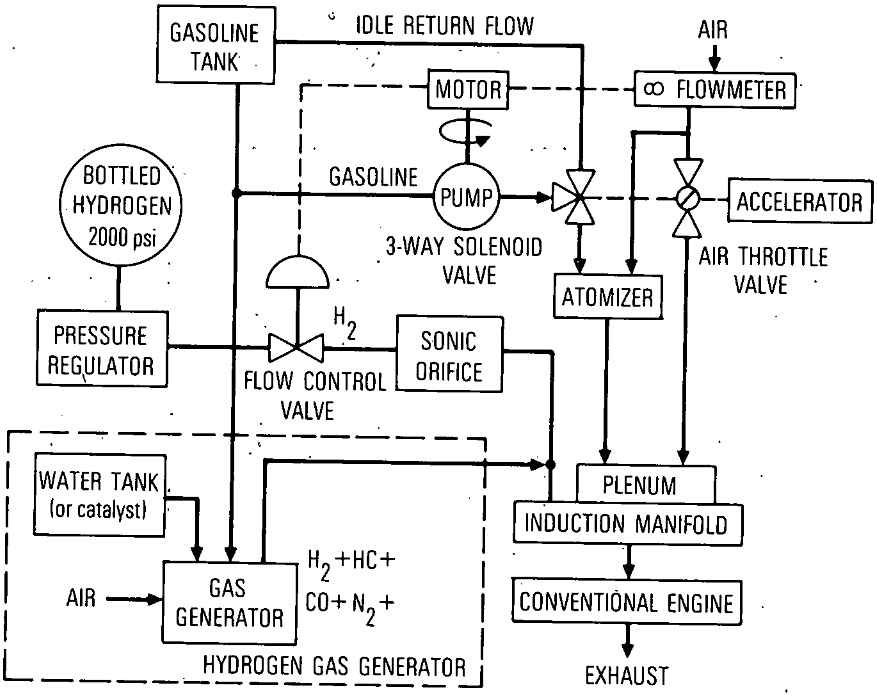


Figure 4-9. Fuel-induction system for JPL hydrogen-enriched gasoline vehicle (Ref. 4-6)

Vehicle emission (1975 FTP) results obtained from testing the GM and JPL test vehicles are summarized in Table 4-1. The two sets of emissions data are comparable in that both vehicles had low NO_x and CO emissions (0.3-0.5 and 2-3 gm/mi, respectively) and high HC emissions (2-3 gm/mi). In light of the previously discussed CFR engine emission results, the high HC emissions are not surprising since both the JPL and GM test vehicles operated much of the time at equivalence ratios of 0.55 to 0.65 at hydrogen weight flows near the lean limit requirements.

Further insight into the effect of hydrogen addition on emissions from multicylinder engines can be gained from the engine dynamometer tests done by JPL (Ref. 4-5). The same engine and fuel system used in the JPL test vehicle was tested at various combinations of rpm and load (level road load conditions for steady-state vehicle operation). In addition, data were taken for the stock engine (Chevrolet 350 CID, V-8) and emission control

γ

Table 4-1. Summary of Vehicle Emissions Results Using Bottled Hydrogen[a]

Vehicle	1975 FTP, gm/mi		
	HC	CO	NO_x
JPL, stock 1973 Chevrolet Impala, V-8, 350-CID engine	2.3	44	1.8
JPL, modified 1973 Chevrolet Impala, V-8, 350-CID engine, Autotronics fuel system, bottled hydrogen, 15% H_2 at $\phi = 0.53$ with 3-lb H_2/hr max. flow, idle on H_2 (1.2 lb/hr) at $\phi = 0.32$, two power enrichment steps: $\phi = 0.63$ and $\phi = 0.94$	2.6 (3.1)[b]	1.5 (2.2)[b]	0.5 (0.31)[b]
GM, 1973 Intermediate, V-8, 455-CID engine, dual fuel system, bottled hydrogen, nominal 10% H_2, $\phi = 0.55$ at idle and $\phi = 0.65$ at 60 mph, increasing ϕ with load and gasoline flow.	3.1	3.3	0.39

[a] Refs. 4-1 and 4-2.

[b] Later tests with leaner operation.

systems. The emission characteristics of the JPL test vehicle engine and fuel system, the Autotronics fuel induction system operating on gasoline alone, and the stock engine emission control system are compared in Figures 4-10 to 4-12 for level road loads up to 80 hp. It is seen in Figure 4-10 that the HC emissions of the engine operating at ultralean conditions ($\phi = 0.53$) with hydrogen addition (13%) are much higher than those for the stock engine and Autotronics systems at the same level road horsepower. Since the FTP HC emissions of the GM and JPL test vehicles are only slightly higher than those of the stock vehicle, improved cold-start and idle emissions of the test

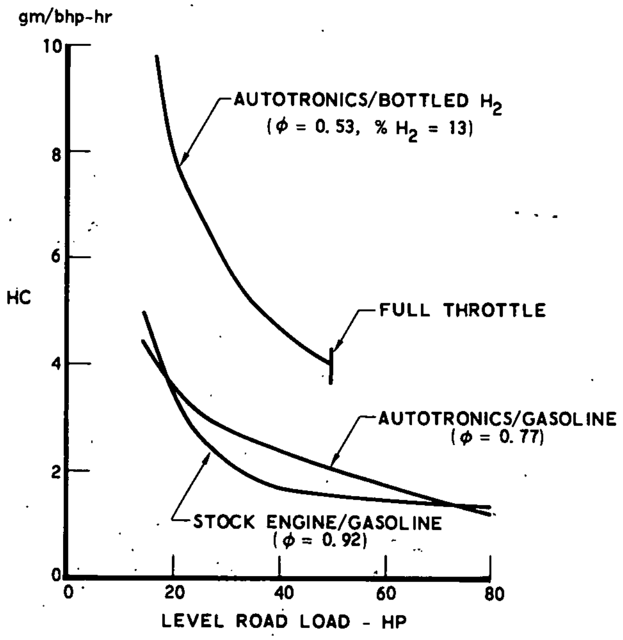


Figure 4-10. V-8 engine HC emissions using bottled hydrogen (Ref. 4-4)

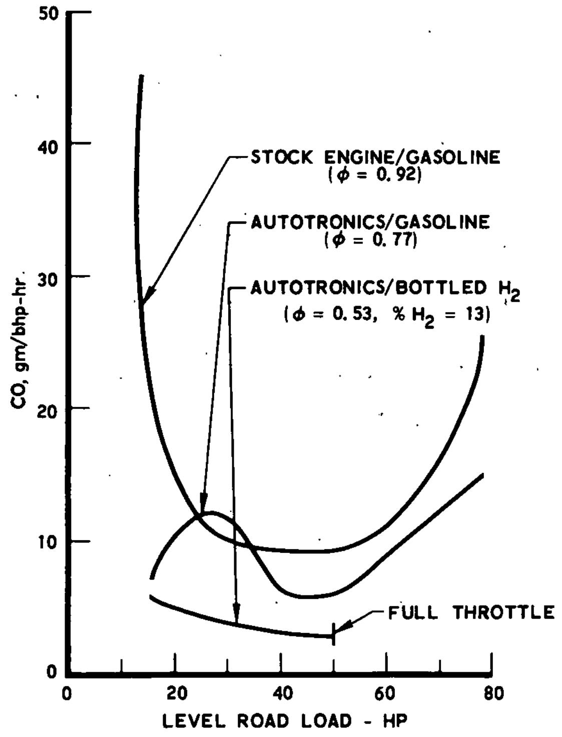


Figure 4-11. V-8 engine CO emissions using bottled hydrogen (Ref. 4-4)

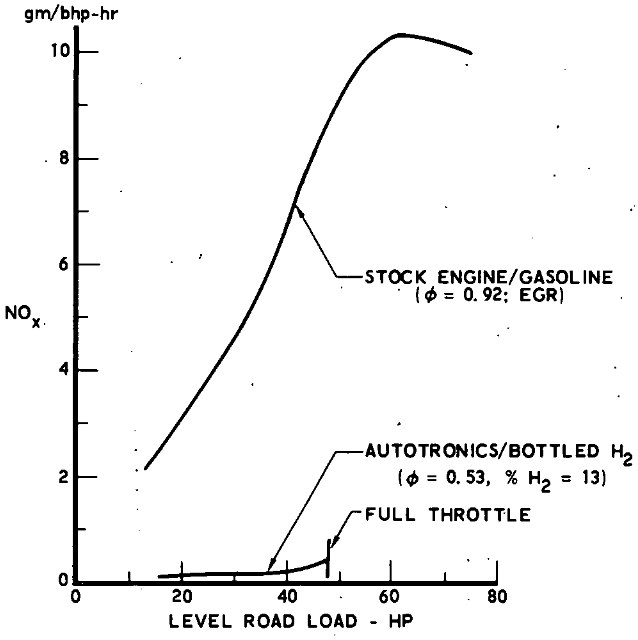


Figure 4-12. V-8 engine NO_x emissions using bottled hydrogen (Ref. 4-4)

vehicles apparently partially compensate for their high emissions at loaded engine conditions. Figures 4-11 and 4-12 indicate that both the CO and NO_x emissions are greatly reduced by operation at low equivalence ratios. This is consistent with the low CO and NO_x emissions found in the FTP vehicle tests. The FTP CO emissions of the test vehicles are even lower than those inferred from the dynamometer tests, again indicating that the hydrogen-enriched fuel system has superior idle and cold start emissions. As expected, the test vehicle FTP NO_x emissions are not reduced by as large a factor as would be inferred from a comparison of the stock engine and hydrogen-enriched fuel system emission results at level road horsepower because hydrogen flow rate and peak power limitations in the gasoline-hydrogen system mandate operation at higher than the minimum equivalence ratios during part of the federal test cycle.

The effect of fuel reformer (gas generator) products (hydrogen as well as inert dilutents) on engine emissions was also studied by JPL using bottled hydrogen and nitrogen. The simulation of the effect of the reformer products was only partial in that fuel components other than hydrogen, especially CO, were not included. Data obtained for the case of a gas generator operating at an equivalence ratio of 3 ($\Phi_{GG} = 3.0$) are shown in Figure 4-13 for a range of hydrogen flow rates at an engine operating point (2000 rpm, 40 bhp) corresponding to a level road speed of 55 mph. Test results are shown for engine operation at equivalence ratios of 0.68 to 0.65 as well as at the lean limit (0.6-0.5). The high HC and CO emissions near the lean limit using minimum hydrogen flow rates are again evident. Also, as was the case for the CFR engine, the NO_x emissions are not sensitive to hydrogen flow rate near the lean limit.

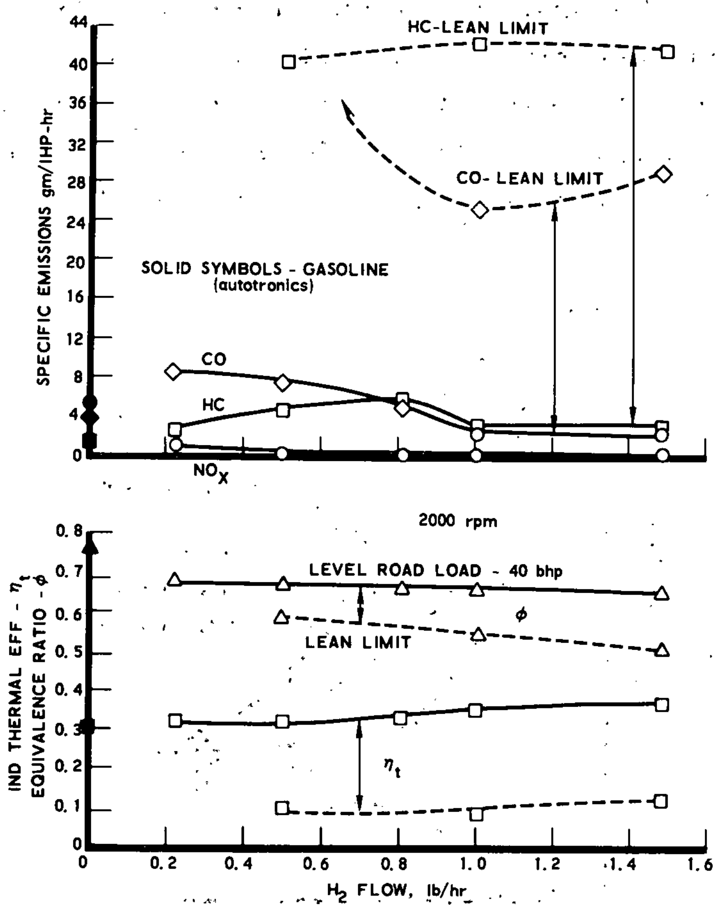


Figure 4-13. Simulation of gas generator products using bottled gases: V-8 engine test; $\phi_{GG} = 3.0$ (Ref. 4-5)

4.2.3 V-8 Engine with Fuel Reformer Emissions

Extensive mapping of the emissions characteristics of a V-8 engine and fuel reformer combination has been done by JPL for a wide range of engine and reformer operating conditions. The complete emission maps are given in Ref. 4-1, but their complexity makes it difficult to compare the emissions resulting from the various engine-fuel system configurations. Hence, as was done in the previous section, the emission comparisons are made at various level road (steady-state) engine operating points. This is done in Figures 4-14 to 4-16 for the stock engine and stock engine modified with the Autotronics fuel induction system operating on gasoline alone and the V-8 engine- and fuel reformer combination with hydrogen flow rates of 0.5, 1.0, 1.5 lb/hr. Comparisons of Figures 4-14 to 4-16 with Figures 4-10 to 4-12 indicate that a vehicle using the V-8 engine and fuel reformer combination with a maximum hydrogen flow of 1.5 lb/hr would have significantly

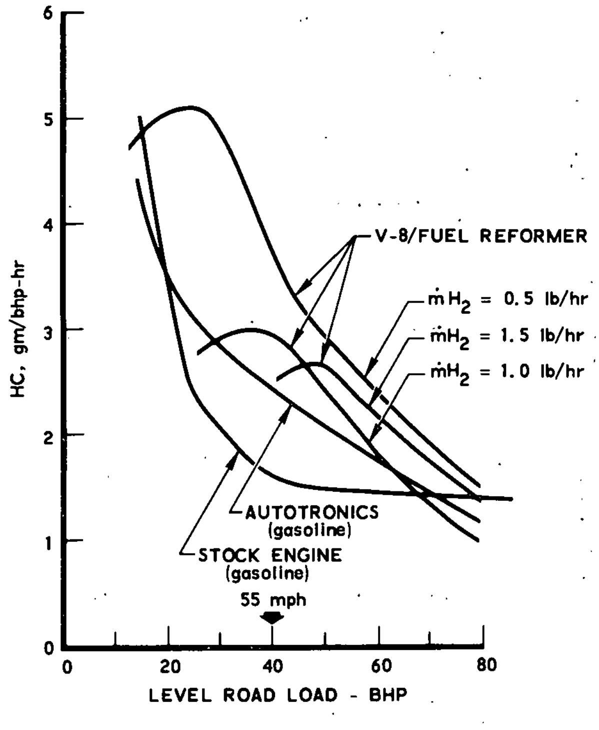


Figure 4-14. HC emissions for V-8 engine gas generator system (Ref. 4-1)

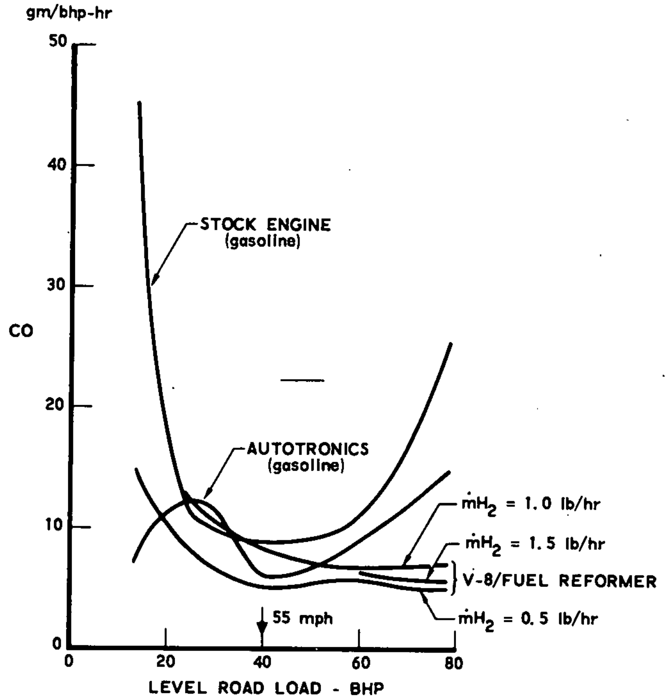


Figure 4-15. CO emissions for V-8 engine gas generator system (Ref. 4-1)

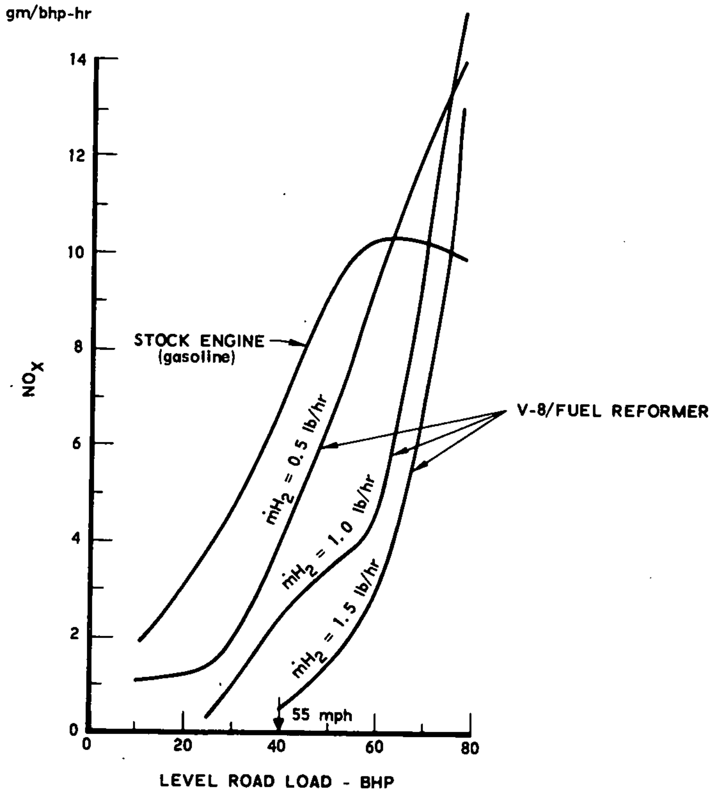


Figure 4-16. NO_x emissions for V-8 engine gas generator system (Ref. 4-1)

lower HC emissions, significantly higher CO emissions, and somewhat higher NO_x emissions on the FDC than the GM and JPL test vehicles previously discussed (Table 4-1). In making these qualitative statements, it was assumed that the cold start emissions of the engine and fuel reformer combination would be similar to those of the bottled gas test vehicles.

Detailed interpretation of the engine and fuel reformer emissions is not straightforward because of the complex way in which the operating equivalence ratio Φ changes with engine load and hydrogen flow rate. In general, Φ can be expected to increase with engine load for a fixed hydrogen flow rate and to decrease for a fixed engine load with increasing hydrogen flow rate. These trends occur because, in order to minimize NO_x emissions and maximize fuel economy, it is desirable to operate the engine as lean as possible consistent with lean operating limit hydrogen requirements, the availability of hydrogen, and the peak power (full throttle) attainable at a given rpm, equivalence ratio, and hydrogen flow rate. For a given engine CID and a fixed maximum hydrogen flow rate, this means that at loads and rpm below certain values it will be possible to maintain ultralean operation ($\Phi \simeq 0.5$), but at higher loads and rpm it will be necessary to allow Φ to increase if engine misfire is to be avoided and the power needed to operate the vehicle is to be achieved. This is illustrated rather pointedly in Figure 4-16 where increasing the hydrogen flow rates permits ultralean operation (NO_x emissions less than 0.5 gm/bhp-hr) up to progressively higher road loads (hp). A rapid increase in NO_x emissions is a good indication of the engine load above which it is not possible to maintain ultralean operation.

Since the primary reason for adding hydrogen to the fuel is to permit ultralean engine operation, it is of interest to correlate emissions with equivalence ratio for various hydrogen flow rates and fuel system configurations. Figures 4-17 to 4-19 show that NO_x emissions decrease rapidly with equivalence ratio, while both HC and CO emissions increase with ϕ beyond a minimum which occurs at $\phi = 0.65$ to 0.75. The minimum level of HC and CO emissions does not appear to be a strong function of hydrogen flow rate at least for hydrogen flow rates up to 2.5 lb/hr. The rate at which

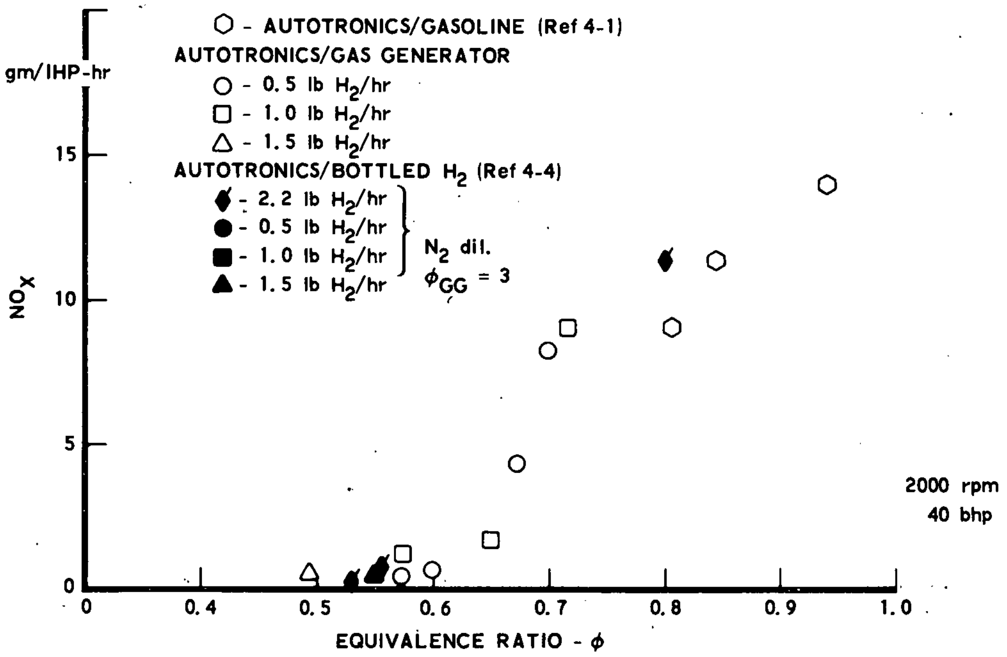


Figure 4-17. Comparison of NO_x emission results at 2000 rpm

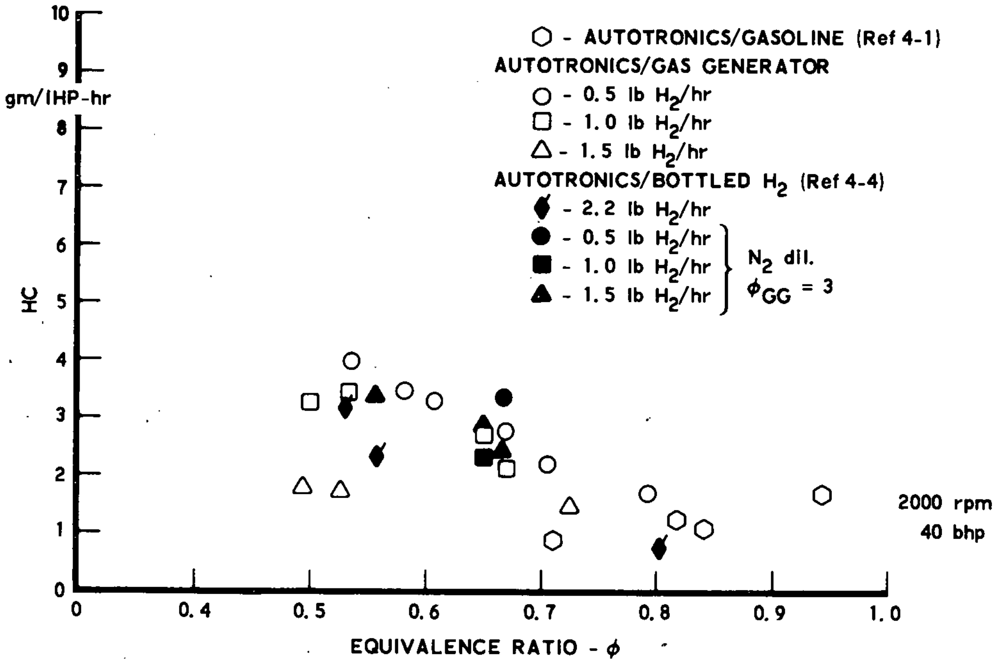


Figure 4-18. Comparison of HC emission results at 2000 rpm

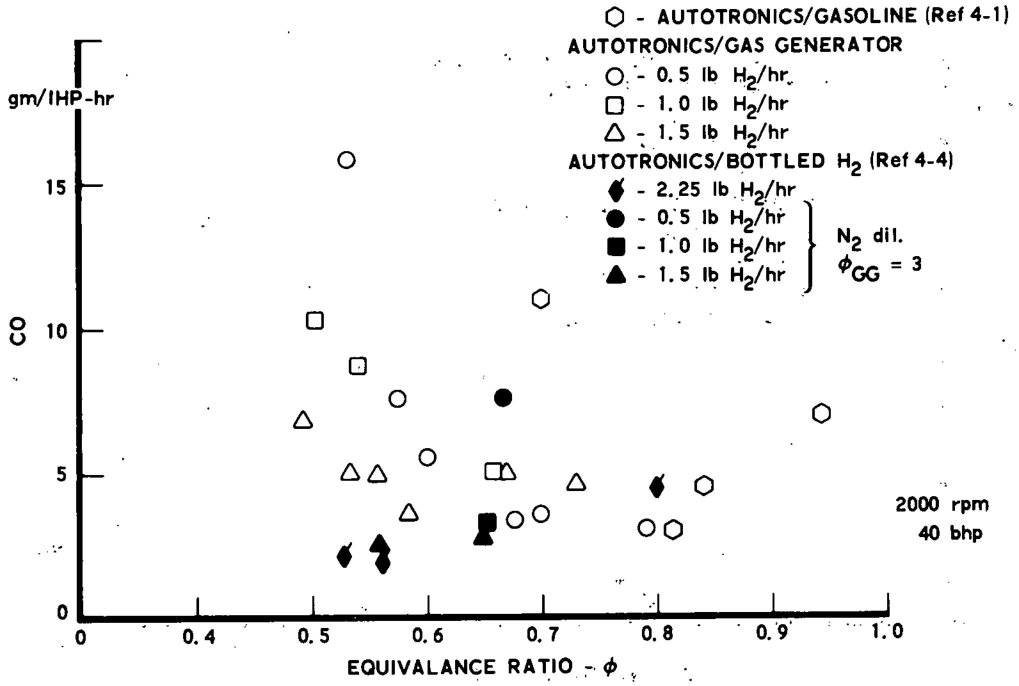


Figure 4-19. Comparison of CO emission results at 2000 rpm

the HC and CO emissions increase beyond the minimum does, however, appear to be a reasonably strong function of hydrogen flow rate. As noted previously, increasing the hydrogen flow rate significantly above the lean limit requirements for particular equivalence ratios reduces the rate at which the HC and CO emissions increase beyond the minimum. It is also of interest to note from Figures 4-17 to 4-19 that the emission levels measured from bottled hydrogen use and those resulting from the use of a fuel reformer to generate the hydrogen are essentially the same at comparable hydrogen flow rates. This indicates that the effect on emissions of the other fuel components (carbon monoxide and methane) and diluents (nitrogen and carbon dioxide) in the reformer products is not significant, and bottled gas simulation of reformer products is valid.

4.2.4 Siemens Emission Data

Limited emissions data with gas generator operation are also available for the Siemens gas generator concept (Ref. 4-13). In this case, comparative test runs were made with a 1.6 liter VW engine with both the stock carburetor and the Siemens gas generator. The emissions results are shown in Figure 4-20 for the 2000-rpm test condition, including variances in equivalence ratio and ignition timing.

With regard to HC, the gas generator had less than one-half the emission level of the stock carburetor over a broad range of equivalence ratios. The HC rise at low equivalence ratios with gas generator operation was not sharp until $\Phi \approx 0.7$. Similar results were found for CO. In the case of NO_x, gas generator operation reduced NO_x steadily as Φ decreased. At $\Phi = 0.7$, the NO_x emissions were approximately one-tenth the level of carburetor NO_x levels. The NO_x emissions increased as the spark was advanced, as in the case of stock carburetor operation.

4.3 EXHAUST MANIFOLD TEMPERATURE

As noted previously, the HC emissions can be quite high when an engine is operated at ultralean conditions. Hence, it is likely that an

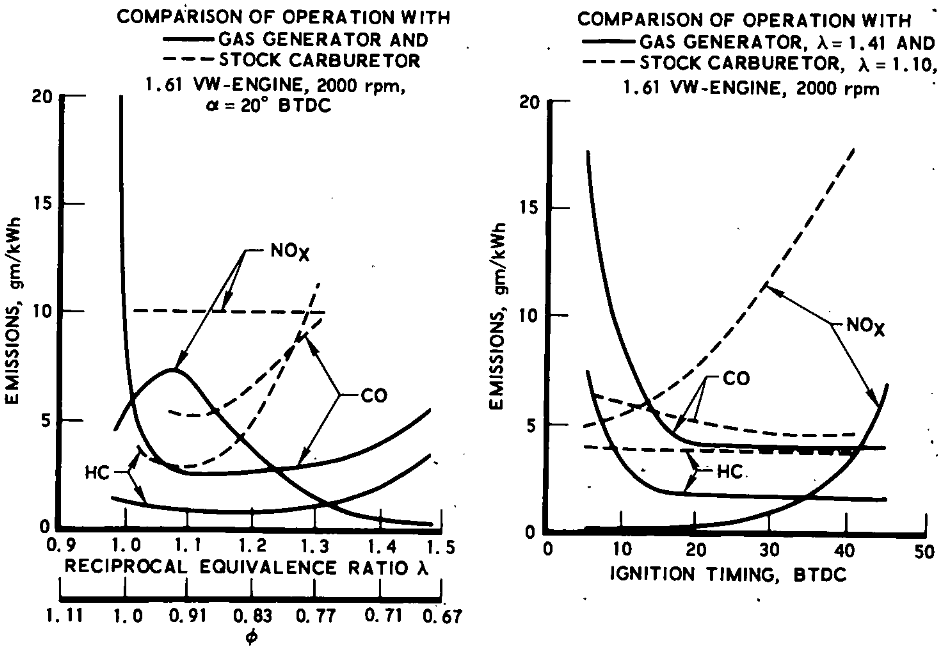


Figure 4-20. Siemens emissions data
(Ref. 4-12)

exhaust gas treatment system (e.g., a thermal reactor or catalytic converter) is needed to reduce HC emissions to levels which will comply with the 1977 Federal Emission Standard (0.4 gm/mi). The conversion efficiency of both the catalytic and thermal reactor systems depends in part on the temperature of the exhaust gas entering the reactor. Thus, of considerable interest is the effect of lean operation on exhaust gas temperature. Measurements of exhaust gas temperature have been made by JPL using thermocouples placed just downstream of the exhaust ports in the exhaust manifold (Ref. 4-7). The temperatures determined represent steady-state time-averaged values over many engine cycles.

Exhaust manifold temperature data are shown in Figure 4-21 for the modified engine and fuel reformer combination operated over a range

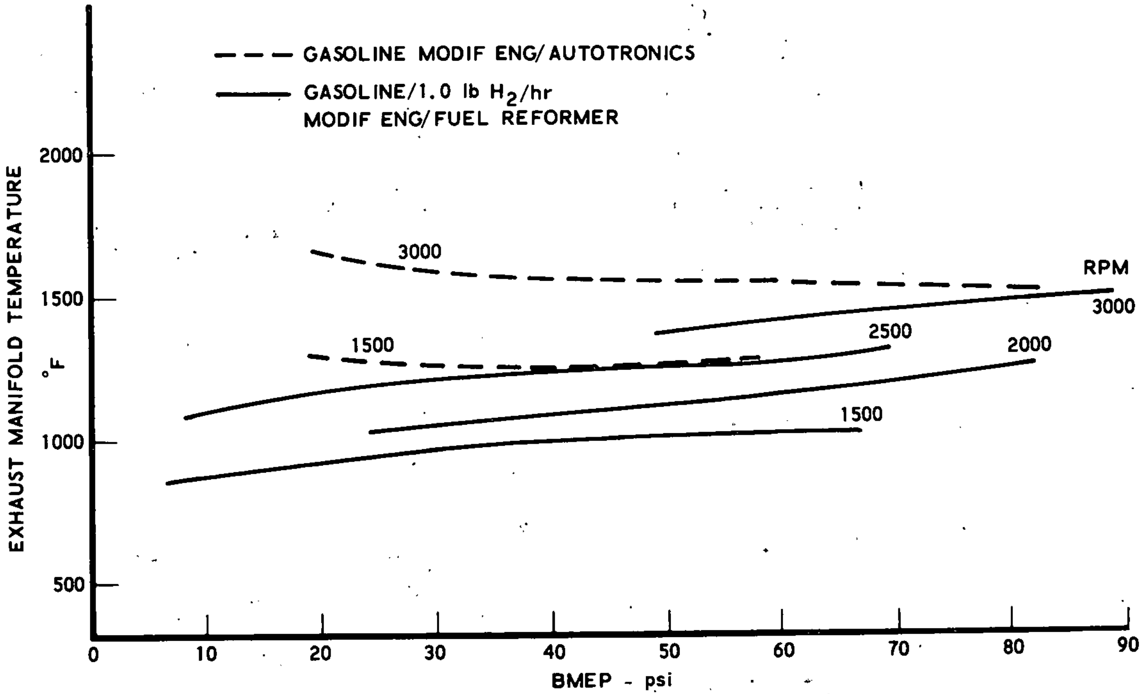


Figure 4-21. Exhaust gas temperature: JPL data (Ref. 4-7).

of engine rpm and throttle setting at brake mean effective pressure (BMEP). Data are given for a hydrogen flow rate of 1.0 lb/hr. Also shown in Figure 4-21 for comparison are the corresponding exhaust temperatures for the modified engine (Autotronics fuel induction system) operating on gasoline alone at 1500 and 3000 rpm. Several observations can be made from Figure 21. First, the exhaust temperatures are relatively independent of BMEP (throttle position) but increase significantly with rpm for both engine configurations. The increase with rpm is probably due to a small increase in the number of crank angle degrees, during which heat release (combustion) is significant, and to reduced heat loss to the coolant at the higher rpm. The spread in the exhaust temperature curves with rpm is greater with hydrogen addition than for operation with gasoline alone. Second, the exhaust temperatures for the modified engine and fuel reformer configuration are significantly lower than for the modified engine operating at near-stoichiometric equivalence ratios ($\phi \approx 1$) on gasoline. Typically, at 1500 rpm the ultralean operating configuration has exhaust temperatures 250 to 350°F lower, while at 3000 rpm the difference is 200 to 300°F. The lower exhaust gas temperatures at lean conditions are due primarily to the lower peak cycle temperatures resulting from less energy release per unit weight of charge air (less fuel added). The exhaust temperatures at lean operation show a greater increase with BMEP than those for gasoline alone because at the higher engine loads and rpm the hydrogen flow rates required to maintain ultralean operation exceed the capability of the fuel reformer to generate hydrogen and it thus becomes necessary to operate at progressively higher equivalence ratios to avoid lean limit misfire. As shown in Figure 4-22, at a given rpm the exhaust temperature depends strongly on equivalence ratio and the effect of fuel system characteristics (operation on gasoline or gasoline-hydrogen mixtures) is of minor importance. Hence, the reduced exhaust temperatures are a result of lean operation, and the magnitude of the reduction can be predicted quite well once the operating equivalence ratio of the engine is known.

JPL conducted vehicle tests with and without an oxidation exhaust catalyst to determine if the use of a catalyst could assist in the

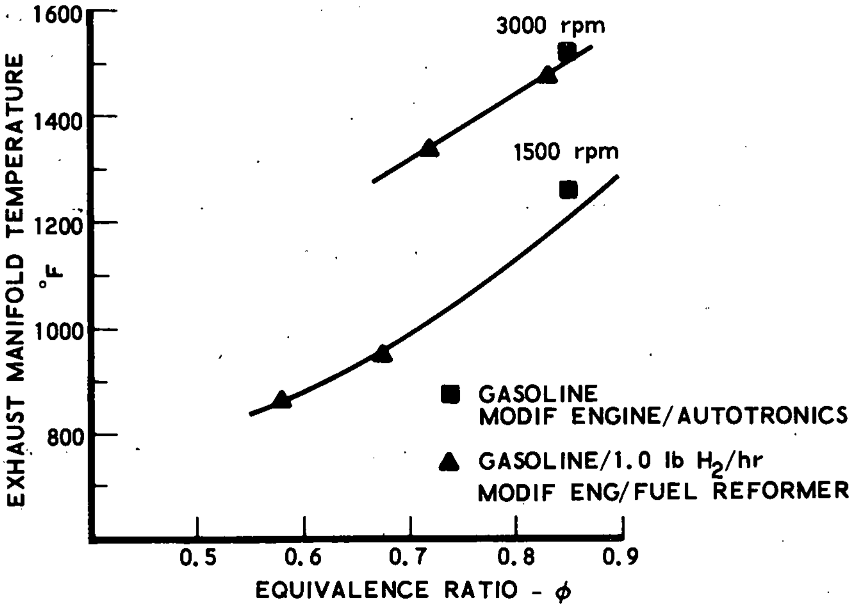


Figure 4-22. Exhaust gas temperature: JPL data (Ref. 4-7)

reduction of the high HC levels associated with ultralean operation. The test results are shown in Table 4-2. The exhaust catalyst was a stock GM 1975 model year oxidation catalytic converter, and the HC reduction efficiency was approximately 77% (from 3.1 to 0.72 gm/mi), with the lower exhaust temperatures resulting from the lean operation of the bottled gas car. The mpg (Btu equivalent) figure of 13 for the bottled gas car is not representative of gas generator operation as the fuel reformer efficiency is not accounted for.

4.4 ENGINE THERMAL EFFICIENCY

The effect of lean operation on engine thermal efficiency is considered in this section. In particular, engine operation at ultralean

Table 4-2. Bottled Gas Car Test Results[a]

Parameter	Unmodified Vehicle	Bottled Gas Car Without Exhaust Catalyst	Bottled Gas Car With[c] Exhaust Catalyst	EPA 1978 Standard
HC emissions, gm/mi	2.29	3.1	0.72	0.41
CO emissions, gm/mi	43.91	2.2	0.1	3.4
NO_x emissions, gm/mi	1.75	0.31	0.28	0.4
Fuel, Btu/mi[b]	12,700	8,850	8,850	
Fuel, mpg (Btu equiv.)	9.5	13	13	

[a] Ref. 4-13.

[b] Total gasoline and hydrogen Btus.

[c] First test with 1975 GM exhaust catalyst, 11-21-75.

equivalence ratios using hydrogen addition is discussed. As was the case for emissions, much of the detailed engine efficiency data are for a CFR engine. Fortunately, considerable multicylinder (V-8) engine data with hydrogen addition are available for both bottled hydrogen and fuel reform systems. The CFR engine efficiency results are presented in terms of indicated thermal efficiency η_i, while the V-8 engine data are given in terms of brake thermal efficiency η_b. The two efficiencies are related through the engine friction or mechanical efficiency η_m by the simple relation

$$\eta_b = \eta_m \eta_i \qquad (4-1)$$

where

$$\eta_m = \frac{bhp}{ihp} = \frac{1}{1 + fhp/bhp}$$

bhp = brake horsepower

ihp = indicated horsepower

fhp = friction horsepower

The friction horsepower includes friction and pumping losses as well as the power needed to drive various engine auxiliary systems. For a particular engine, fhp[7] is primarily a function of engine rpm since the variation of pumping losses with throttle position is relatively small.

Thermal efficiency data (η_i) taken by JPL (Ref. 4-1) and GM (Ref. 4-2) on CFR engines are summarized in Figures 4-23 and 4-24. Results are shown for a wide range of equivalence ratio and various hydrogen weight fractions. The following conclusions relative to the effect of equivalence ratio and hydrogen addition on engine efficiency can be drawn from the cited data:

1. The maximum (peak) thermal efficiency for a given gasoline-hydrogen mixture increases as the lean limit of the engine and fuel system decreases, but the incremental improvement in peak efficiency becomes small for lean limit equivalence ratios less than ~0.6.
2. The improvement in thermal efficiency beyond that possible with gasoline operation at its lean limit is relatively small (~10-15%), and the hydrogen weight fraction needed to attain essentially all the improvement is less than 15%.
3. The maximum thermal efficiency for a given gasoline-hydrogen mixture occurs at an equivalence ratio very near the lean limit for that mixture.

A summary of V-8 engine brake thermal efficiency data[8] (Refs. 4-1 and 4-4) taken by JPL using the 1973 Chevrolet 350 CID engine

---

[7] In this study, wherever required, the friction horsepower was calculated with a curvefit polynomial expression developed by JPL and based on experimental data from GM.

[8] The brake thermal efficiency was calculated in some cases from the brake specific fuel consumption (BSFC) data given in Ref. 4-1.

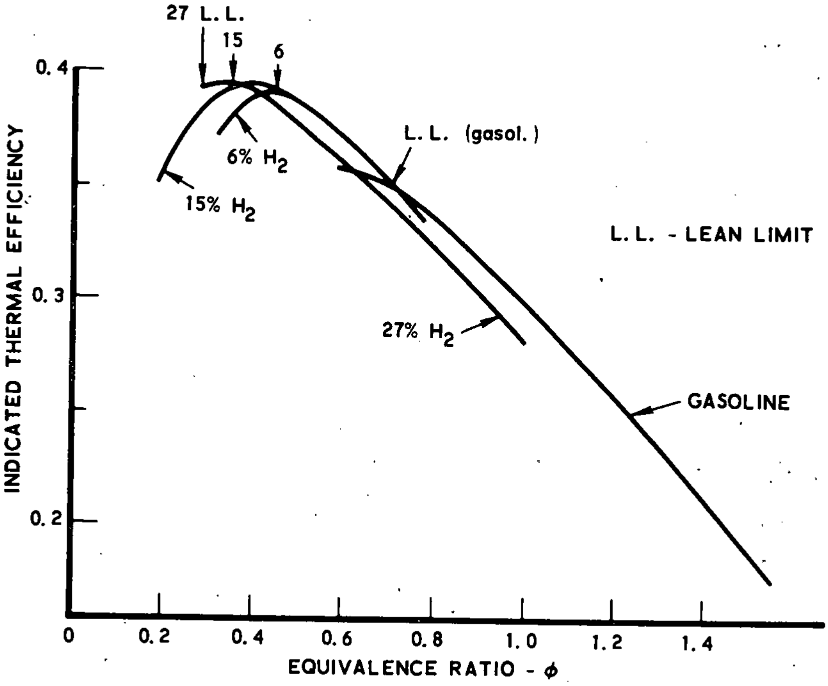


Figure 4-23. CFR engine thermal efficiency trends (Ref. 4-1)

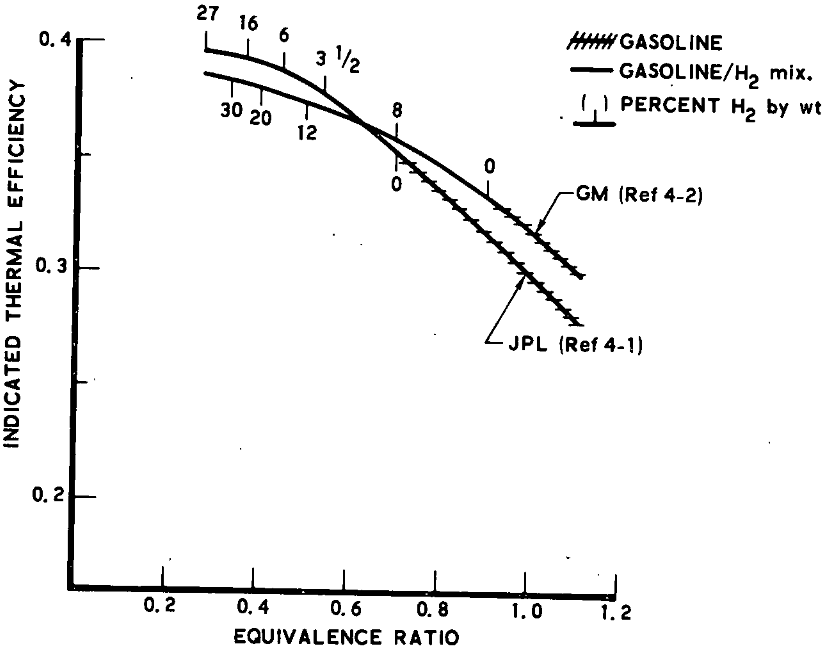


Figure 4-24. CFR engine thermal efficiency trends: GM and JPL

and its modifications is given in Figure 4-25. Data are shown for the stock engine, the modified engine using the Autotronic fuel induction system operated on gasoline alone, and the modified engine operating on gasoline-hydrogen mixtures for a range of level road loads. All the data for the modified engine system were taken for MBT spark timing. The stock engine was run at the manufacturer's specified timing, which was necessarily retarded in order to meet the 1973 HC and NO_x exhaust emission standards. The timing change between the stock and modified engine systems accounts for a significant fraction of the difference in thermal efficiency found when the engine systems were operated on gasoline. The remainder of the difference in thermal efficiency is due to the leaner operation (ϕ = 0.77 compared with 0.92) of the modified engine. The results labeled "JPL bottled H_2 vehicle" probably represent an upper bound for the improvement in thermal efficiency to be expected for ultralean operation using gasoline hydrogen mixtures since the JPL test vehicle system was operated at a leaner condition ($\phi \simeq 0.5$) and had a greater hydrogen flow rate capability (up to 3 lb/hr)

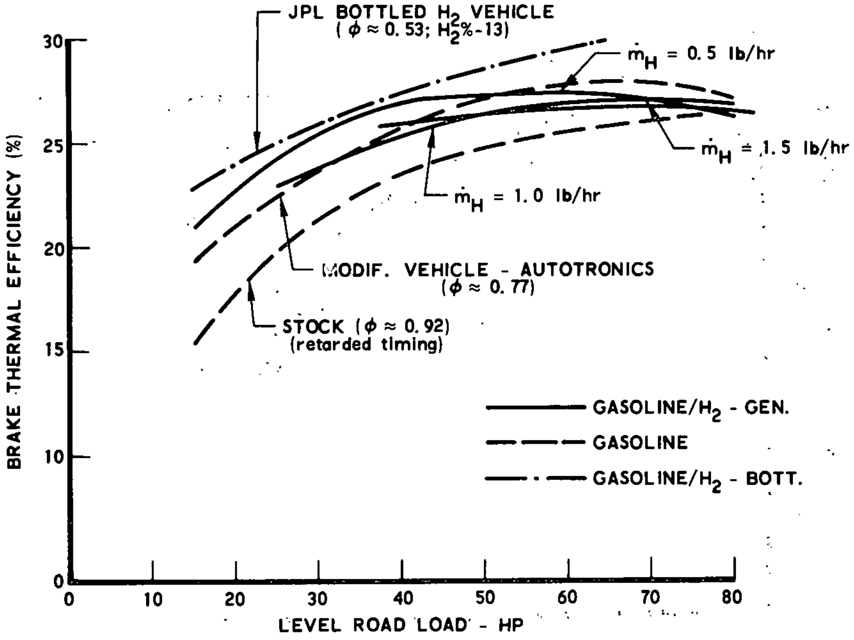


Figure 4-25. V-8 engine thermal efficiency data (Refs. 4-1 and 4-4)

than is likely to be possible in systems using a fuel reformer. Brake thermal efficiency data for the JPL V-8 engine fuel reformer system are shown in Figure 4-25 for hydrogen flow rates of 0.5, 1.0, and 1.5 lb/hr. All the V-8 engine fuel reformer system results fall within a narrow band. This indicates that the system losses associated with passing a larger fraction of the gasoline through the partial oxidation reformer (having a thermal efficiency of ~80%) at higher hydrogen flow rates essentially just balance the effect of the slightly improved engine thermal efficiency at increased hydrogen flow capacity and the resultant maintenance of ultralean operation up to higher road loads.

The magnitude of the improvement in the thermal efficiency of the V-8 engine (Figure 4-25) for lean operation with gasoline and ultralean operation with gasoline-hydrogen mixtures is consistent with those inferred from the CFR engine data given in Figure 4-23 and 4-24. First, for gasoline operation, the CFR engine data indicate that lowering the equivalence ratio from 0.92 to 0.77 increases the thermal efficiency by ~8%, which compares reasonably well with the 10% to 11% difference in thermal efficiency found for the stock and modified engine systems (Figure 4-25) at moderate road loads (30-60 hp) especially since significant changes in timing were also made. Secondly, the maximum incremental improvement in thermal efficiency resulting from ultralean operation with hydrogen addition in the CFR engine was ~15%. The comparable value of 10% was found for the JPL test vehicle using bottled hydrogen. Further evidence to support the cited 10% improvement in thermal efficiency due to hydrogen addition can be obtained from the steady-state level road load results when the V-8 engine reformer system was used. JPL found that the brake thermal efficiency of that system was essentially the same as that of the Autotronics system operating on gasoline alone. For the case of a hydrogen flow rate of 1.0 lb/hr, which requires that ~40% of the total gasoline flow be passed through the JPL fuel reformer, an improvement in engine thermal efficiency of 8.9% is needed just to balance the effect of the 80% thermal efficiency of the fuel reformer.

4.5 VEHICLE FUEL ECONOMY

In the previous section, the effect of lean operation and hydrogen addition on engine thermal efficiency at various steady state operating conditions was considered. The results yield some insight into similar effects on vehicle fuel consumption on a driving cycle such as the LA-4, but the transient nature of the engine-vehicle operation needed to follow the FDC makes it difficult to accurately calculate vehicle fuel economy on a driving cycle using steady-state engine dynamometer data. Hence, the only reliable way of determining the effect of lean operation and hydrogen addition (with or without a fuel reformer) on vehicle fuel economy is to actually measure the fuel consumption on a chassis dynamometer or to road test the vehicle-engine-fuel reformer system. Unfortunately, no such data for the system including a fuel reformer are presently available, and only limited data for bottled hydrogen have been taken. In the absence of appropriate test data, recourse can be made to calculations of vehicle fuel economy if the uncertain accuracy of the results is recognized and calculated differences between the systems are assessed accordingly. JPL has made fuel economy calculations (Ref. 4-1) in which the FDC was simulated on the computer by a series of quasi-steady-state steps. The JPL results are used in the subsequent discussions of fuel economy.

The available vehicle fuel economy results (both test data and calculations) pertinent to the present study are summarized in Table 4-3. Fuel economy values are given for each of the four engine configurations discussed previously in the section on engine thermal efficiency. Each of the engine configurations are discussed in turn, and the effects of lean operation and hydrogen addition on fuel economy are delineated.

The stock or reference vehicle has a measured fuel economy of 10 to 10.5 mpg.[9] The corresponding fuel economy predicted using the JPL computer program and engine dynamometer data is 12.1 mpg. When the

---

[9] All fuel economy values will be given in terms of mpg gasoline equivalent.

Table 4-3. Summary of Vehicle Fuel Economy Test and Calculation Results

Engine Configuration	Vehicle Fuel Economy, mpg	
	Measured[a]	Predicted[b]
Stock	10.6 10.1[c]	12.1
Modified, Autotronic carburetor (gasoline only)[d]	12.8	13.8
Modified, dual fuel reformer (gasoline and bottled hydrogen)	13.5[c]	--
Modified, dual fuel reformer (gasoline and 0.5 H_2 lb/hr)	--	14.1

[a] Chassis dynamometer, 1975 FTP constant volume sampling (CVS) (Refs. 4-1 and 4-8).

[b] Prediction based on JPL simulation of FDC and engine dynamometer performance data (Ref. 4-1).

[c] Calculated from Btu/ton-mi test values given in Ref. 4-16.

[d] Timing advanced to MBT.

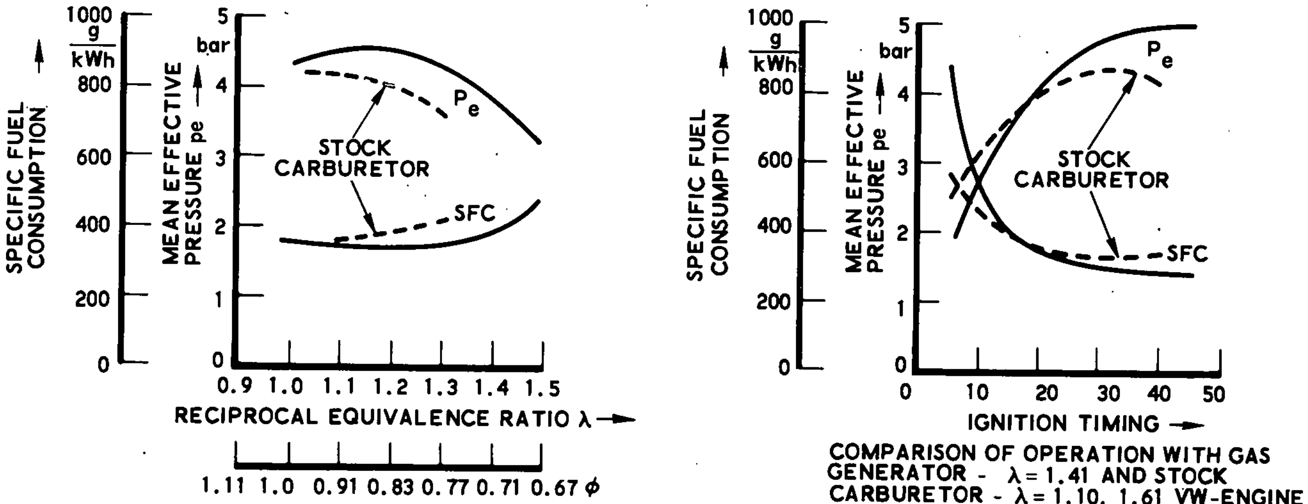
stock engine system is modified by replacing the standard carburetor and induction system with the Autotronics fuel induction system, removing the EGR system and air pumps, and advancing the spark timing to MBT, the measured fuel economy of the vehicle using the modified engine was 12.8 mpg. The 20% improvement in fuel economy is due to three factors: (1) leaner operation, (2) advanced spark timing, and (3) no EGR. The stock engine using the standard carburetor and induction system would have certainly yielded better fuel economy than 10 mpg had the timing been advanced to MBT and the EGR system been removed. Hence, only a part (probably less than one half) of the fuel economy improvement of the modified engine can be credited to leaner operation. It should also be noted that the modified engine vehicle had about double the HC and NO_x emissions of the stock vehicle. The JPL computer program again predicted a higher fuel economy than measured for the modified vehicle, i.e., 13.8 mpg compared with the measured value of 12.8 mpg.

Next consider the engine configurations utilizing hydrogen addition to attain ultralean combustion. Tests (Ref. 4-6) of the JPL test vehicle operating on a gasoline-hydrogen mixture with the hydrogen being supplied from high pressure bottles showed a fuel economy of 13.5 mpg (gasoline equivalent) on the FDC. This is only ~5% better than the modified vehicle attained when it was operated on gasoline alone. Both vehicles had the timing set at MBT, and neither had an EGR system for NO_x control. Hence, the 5% improvement in fuel economy is due only to lean operation made possible by hydrogen addition. The modified vehicle operating on a gasoline-hydrogen mixture and utilizing a fuel reformer to generate the hydrogen has not been tested; only calculated fuel economy results are available for that system. For the modified engine and fuel reformer configuration, the JPL computer program (Ref. 4-1) predicts a vehicle fuel economy of 14.1, which is essentially the same as that predicted for the modified vehicle operating on gasoline alone. Both the JPL test vehicle (bottled hydrogen) fuel economy data and computer results for the modified engine with fuel reformer configuration indicate that the effect of hydrogen addition on fuel economy will be small, less than 10%. This is consistent with the conclusion reached previously on the basis of engine thermal efficiency data (Figures 4-23 and 4-25.

The effect of the fuel reformer on the vehicle fuel economy depends both on the fraction of the gasoline f_G passed through the reformer and the thermal efficiency η_{Rth} of the reformer. At a given hydrogen flow rate, the presence of the fuel reformer in the system reduces the fuel economy by a factor $[1 - f_G(1 - \eta_{Rth})]$ from that which would have resulted from the use of bottled hydrogen, because part of the heat content of the gasoline is used in the reforming process to produce gaseous products (including hydrogen) before burning in the engine. If, for example, 40% of the total gasoline to the engine is passed through a reformer having a thermal efficiency of 80%, the effect of the reformer would be to reduce the fuel economy by 8%. Hence, in that case the improvement in average engine efficiency over the driving cycle due to hydrogen addition would have to exceed 8% before a net gain in vehicle fuel economy would be realized.

4.5.1 Siemens Fuel Consumption Data

Specific fuel consumption data with gas generator operation are available for the Siemens gas generator concept (Ref. 4-13). Comparative test runs, both with the stock carburetor and the Siemens gas generator, were made with a 1.6-liter engine from Volkswagon (VW). The specific fuel consumption (SFC) results are shown in Figure 4-26 for the 2000-rpm test condition, including variances in equivalence ratio and ignition timing.



COMPARISON OF OPERATION WITH GAS GENERATOR - AND STOCK CARBURETOR 1.61 VW-ENGINE, 2000 rpm, ∝ = 20° BTOC

COMPARISON OF OPERATION WITH GAS GENERATOR - λ = 1.41 AND STOCK CARBURETOR - λ = 1.10, 1.61 VW-ENGINE 2000 rpm

Figure 4-26. Seimens fuel consumption data (Ref. 4-12)

When spark advance is limited to 20 deg, the SFC of the gas generator-equipped engine in the 0.7 to 0.75 equivalence ratio Φ range was approximately the same as the stock carburetor in the Φ = 0.9 to 1.0 range. At the leanest condition shown (Φ = 0.65), however, the gas generator SFC was ~40% poorer than the stock carburetor at Φ = 0.9 to 1.0. With increased spark advance, to 45° BTDC, the gas generator minimum SFC becomes ~13% better at Φ = 0.65 than that of the stock carburetor at Φ = 0.91. This shows rather graphically that lean operation does not result in improved fuel economy unless it is accompanied by an increase in spark advance.

4.6 ENGINE POWER

The maximum power that can be produced by an engine at full throttle depends on the fuel flow rate, heat content of the fuel, and the thermal efficiency of the engine. The fuel flow rate depends on the weight flow of air (and hence oxygen) through the engine and the equivalence ratio Φ at which the engine is being operated. At a given rpm, the maximum total volumetric gas flow (air, fuel, generator products, etc.) is set by the size (CID) of the engine. Any factor or system characteristic which reduces the weight of air to the engine necessarily reduces the maximum power that can be generated by the engine at a given rpm and Φ. Thus, gaseous fuels and the presence of fuel reformer products in the charge to the engine reduce maximum horsepower because those gases occupy volume which can be occupied by charge air if gasoline alone is used as the fuel. In addition, any heating of the air-fuel mixture by the reformer product gases further reduces the mass flow of air to the engine. Gaseous hydrogen has a low density; therefore, its presence reduces the air flow more than an equal weight flow of a heavier fuel gas.

For a given air flow to the engine, the amount of fuel added is proportional to the operating equivalence ratio. Hence, the maximum power is reduced more or less proportional to Φ, and lean operation always results in reduced maximum power from an engine regardless of how the lean operation is accomplished. The small improvement in engine thermal efficiency at these conditions compensates for only a small part of the reduction in fuel flow required to maintain a lean Φ. These characteristics are exemplified by the GM CFR engine data shown in Figure 4-27.

Measurements of full throttle power for the modified engine and fuel reformer configuration have been made by JPL (Ref. 4-1) for hydrogen flow rates of 0.5, 1.0, and 1.5 lb/hr. Data taken at 2500 rpm are given in Figure 4-28 in terms of indicated horsepower as a function of engine equivalence ratio. Indicated horsepower is used rather than brake horsepower because the decrease in mechanical efficiency with bhp tends to exaggerate the reduction in maximum power which occurs at low equivalence ratios. For 2500 rpm, the mechanical efficiency is 79% at 100 bhp, and it decreases to 69% at 60 bhp. Also shown in Figure 4-28 are the maximum indicated horsepower values for the stock engine operating at Φ ≅ 1

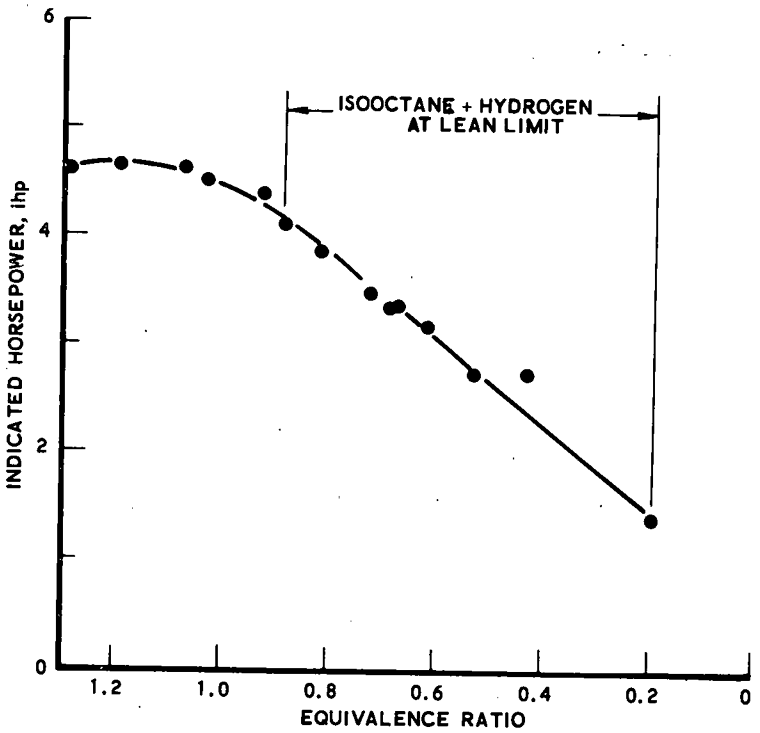


Figure 4-27. Effect of ultralean operation on engine power with hydrogen-supplemented fuel: GM CFR engine (Ref. 4-2)

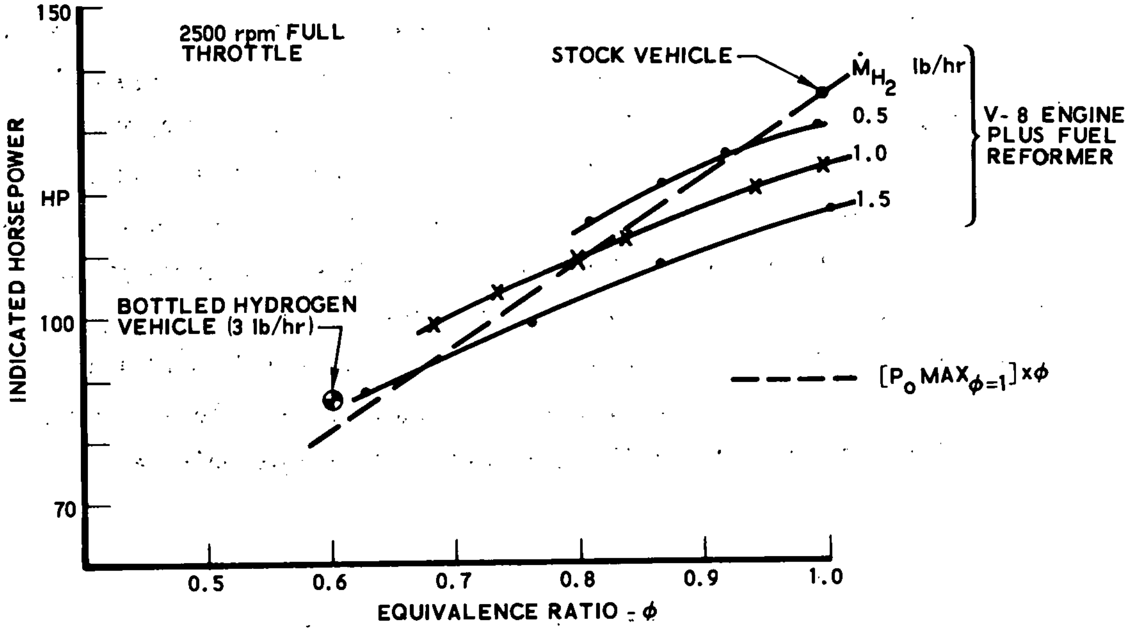


Figure 4-28. Engine power output vs equivalence ratio

and the JPL test vehicle (bottled hydrogen storage) operating at $\Phi = 0.6$ (Ref. 4-9). As expected, the maximum horsepower is reduced significantly by lean operation, and the magnitude of the reduction at a given equivalence ratio increases as the hydrogen flow rate is increased. Typically, for a hydrogen flow rate of 1.0 lb/hr, the maximum horsepower is reduced by about 10% at $\Phi = 1$ and by about 35% at $\Phi = 0.65$.

4.7 FUEL REQUIREMENTS AND ENGINE KNOCK

The effect on fuel requirements and engine knock of a fuel reformer used to process a significant portion of the gasoline fed to an engine is considered in this section. When a reformer is used, the fuel mixture fed to the engine includes hydrogen, carbon monoxide, and methane in addition to gasoline. The knock and autoignition characteristics of such a fuel mixture can be expected to be significantly different than gasoline alone. A systematic study of the characteristics of a mixture of gasoline and reformer products in a multicylinder engine has not been done to date, but there have been studies (Ref. 4-1) of gasoline-hydrogen mixtures and mixtures of various gaseous fuels (Refs. 4-10 and 4-11) in CFR engines. The results of these latter studies can provide some insight into how the reformer products are likely to affect engine knock resistance in a multicylinder engine.

JPL has conducted a series of tests (Ref. 4-1) in a CFR engine in which the equivalence ratio, hydrogen weight fraction, and compression ratio were varied and the knock characteristics of the engine operation measured. Engine knock was characterized from cylinder pressure measurements in terms of a knock magnitude index which depends on both the fraction of the engine cycles in which pressure fluctuations occur and the average magnitude of the fluctuations. A summary of the CFR engine knock results are given in Figure 4-29 for various fuel mixtures and a range of equivalence ratios. The data shown indicate that the critical compression ratio for the onset of significant knock increases with leaner operation on gasoline alone, and at a given lean equivalence ratio becomes larger as the hydrogen weight fraction is increased. The JPL data suggest that adding hydrogen to a gasoline mixture significantly increases the knock resistance (i.e., effective octane rating) of the mixture, permitting the use of a lower octane gasoline at a fixed engine compression ratio or an increase in allowable engine compression ratio for a fixed octane gasoline. This result is initially somewhat surprising in that the addition of hydrogen increases the flame propagation rate and thus might be expected to make the mixture detonate (and hence knock) more easily. The increased burning rate, however, also limits the residence time of the end gas in the cylinder, and this reduces the chances for detonation and knock. Evidently, this second effect of hydrogen addition is dominant, at least for the CFR engine tests. As shown in

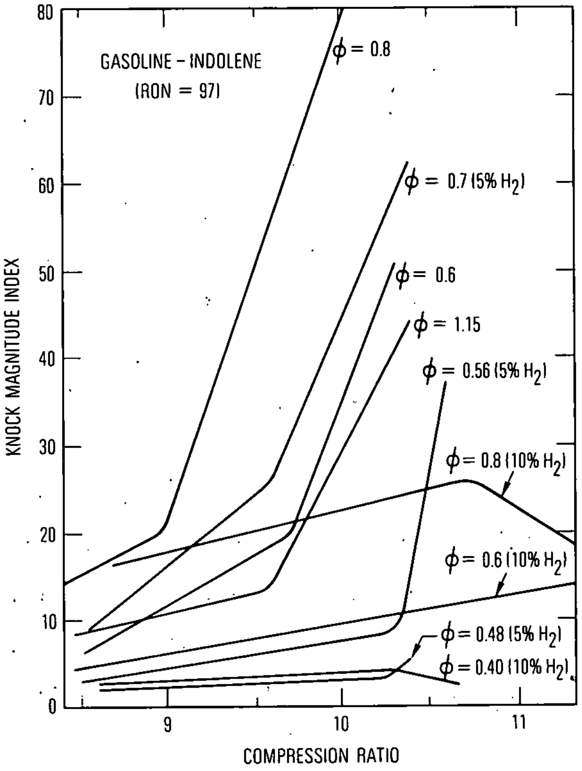


Figure 4-29. CFR engine knock results vs compression ratio, hydrogen weight, percent, and equivalence ratio (Ref. 4-1)

Figure 4-30, the spark advance used in the CFR engine tests was much less than that found necessary in the multicylinder (V-8) engine studies to produce maximum torque and thermal efficiency. The effect of this difference in spark timing on knock characteristics of the various fuel mixtures is not known; hence, some caution should be followed in using the CFR results directly without further studies with V-8 engines. Preliminary indications from the JPL tests with the V-8 engine- and fuel reformer system are that combustion is rapid with hydrogen addition, but ignition delay is significant, which accounts for the need for a 40 to 50 deg spark advance.

CFR studies (Refs. 4-10 and 4-11) have also been made of the knock characteristics of various mixtures of gaseous fuels. Methane has been found to be the most knock-resistant of the gaseous fuels, having an effective octane rating between 115 and 120. As indicated in Figure 4-31, the knock resistance of hydrogen is only slightly less than iso-octane (which

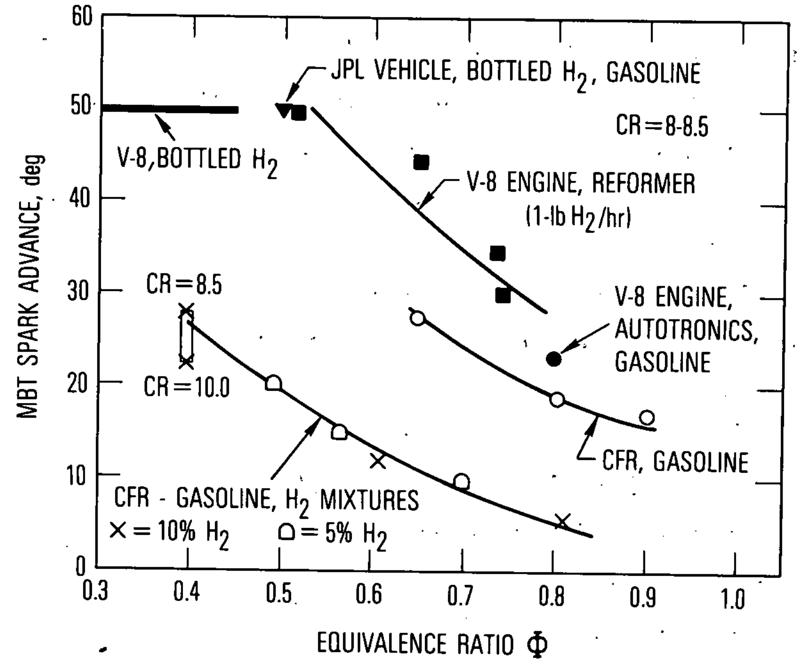


Figure 4-30. Summary of MBT spark advance settings for various engine configurations and fuel mixtures (Refs. 4-1, 4-4, and 4-5).

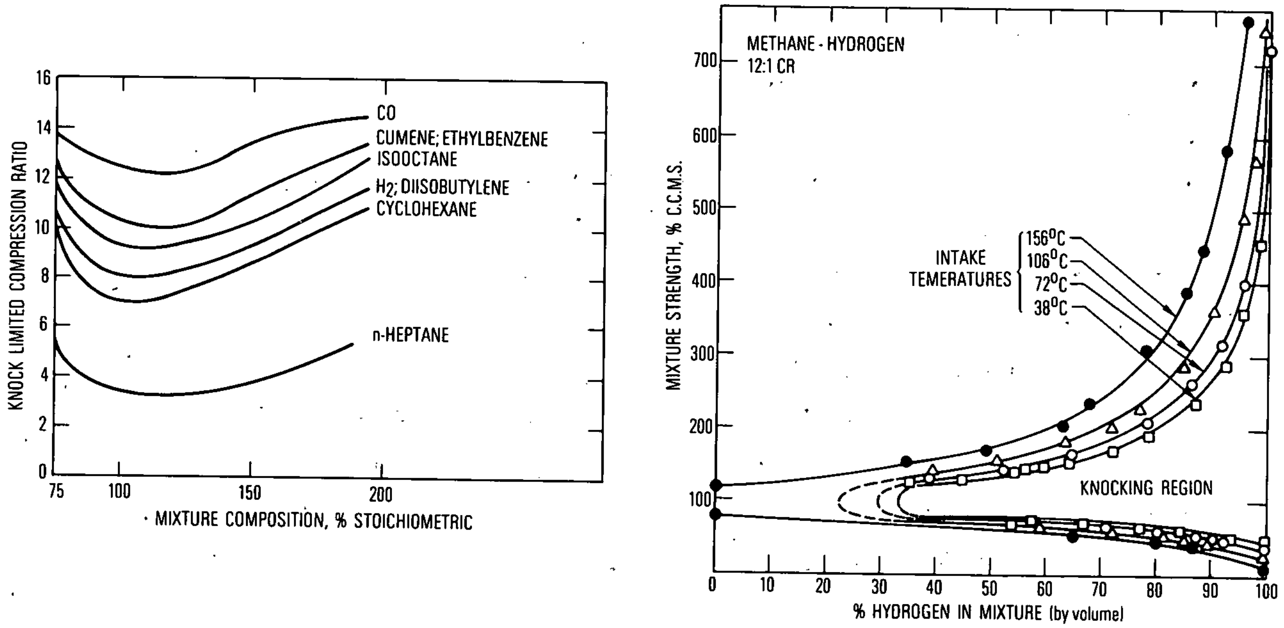


Figure 4-31. Knock characteristics of various gaseous fuels (Refs. 4-10 and 4-11)

has an octane rating of 100 by definition), and that of CO is significantly greater than iso-octane. Hence, one would expect that fuel mixtures (gasoline and reformer products) encountered in an engine-reformer system such as that developed by JPL would have an effective octane rating considerably greater than the gasoline.

The gaseous fuel results, combined with the JPL CFR engine data with gasoline-H_2 mixtures, give a strong indication that the fuel octane requirements are less severe using the engine-reformer system and, if all the liquid fuel were processed by the reformer rather than only a portion of it, the octane rating of the liquid fuel would be of little significance. In either case, it is reasonable to assume that the compression ratio of engines used with a fuel reformer can be increased above that customarily used for gasoline alone, unless the overall system approach includes the operational mode of all-gasoline operation.

4.8 DRIVEABILITY OF HYDROGEN-ENRICHED GASOLINE VEHICLES

The effect on vehicle driveability of hydrogen addition and the use of a fuel reformer in the fuel system is discussed in this section. As indicated in Table 4-4, driveability considerations can be divided into two groups, (1) vehicle startup and (2) vehicle performance. Detailed information on the effect of hydrogen addition per se on driveability is available as both JPL and GM have road tested vehicles utilizing onboard high pressure storage of hydrogen. The effect on driveability of the fuel reformer is not known at the present time because fuel reformer development work is still in progress and a total vehicle system utilizing even a prototype fuel reformer to generate the hydrogen has not been assembled and road tested to date. Startup, transient operation, and control of the fuel reformer is currently an important part of the continuing JPL reformer project. Until that work is completed, testing of the complete vehicle system is not possible.

Table 4-4. Driveability Considerations

Vehicle Startup

- Time-to-start after key turned on
  - Reformer startup characteristics
  - Effects of cold weather
  - Fuel mixture
- Time to warmup
- Idling and stall
  - Fuel mixture
  - Throttling (air-fuel)
  - rpm

Vehicle Performance

- Acceleration times
  - Changes in air-fuel with power required
  - Loss of power
- Smoothness of accleration
  - Transient fuel reformer operation
  - Control of fuel flow to engine and reformer
- Top speed

4.8.1 Vehicles Using Bottled Hydrogen Storage

In the GM and JPL tests (Refs. 4-2 and 4-6) of vehicles operating on gasoline-hydrogen mixtures, good driveability was experienced, and no unexpected problems were encountered. Engine starting at both cold and warm ambient conditions was satisfactory. Vehicle startup and idle on hydrogen alone was smoother than with a gasoline-H_2 mixture, but either approach was workable. No difficulties with preignition were encountered because the hydrogen-air mixtures at idle and hydrogen-gasoline-air mixtures at load were sufficiently lean to prevent it. Vehicle acceleration times were longer with use of hydrogen addition than for the stock gasoline engine because of leaner operation of the modified vehicle even at full throttle. For example, elapsed times to accelerate from 0 to 65 mph were 16.2 sec for the stock gasoline vehicle and 25.2 sec for the JPL modified vehicle (Ref. 4-6). Full throttle accelerations were smooth, but a mild surge was noticeable for part-throttle accelerations in the JPL vehicle due to programmed power-enriching transitions of the air-fuel ratio.

On the basis of the testing of the GM and JPL vehicles utilizing bottled hydrogen storage, hydrogen addition per se to gasoline presents no significant driveability problems, and any such problems encountered with the engine-fuel reformer system result from the startup and transient characteristics of the fuel reformer.

4.8.2 Vehicles Using a Fuel Reformer

Most of the questions concerning the effects of the fuel reformer on the driveability of hydrogen-enriched gasoline vehicles are related to the startup and transient operating and control characteristics of the reformer. Unfortunately, only limited information is available about these aspects of the reformer operation as most of the work done by JPL was concerned with steady-state conditions. Data are given in Ref. 4-1 on the startup characteristics of the most compact JPL reformer design. As indicated in Figures 4-32 and 4-33, the time required for the reformer to reach steady-state operation from a cold start is about 1-1/2 to 2 min, with the energy content of the reformer products being high enough after ~20 sec to idle the V-8 engine. Details of the reformer startup procedures are also given in Ref. 4-1.

Detailed investigations of various approaches to starting vehicles with an engine-reformer system and controlling the gasoline flow to the reformer during transient vehicle operation have not been performed to date. Such studies are necessary before specific questions regarding vehicle

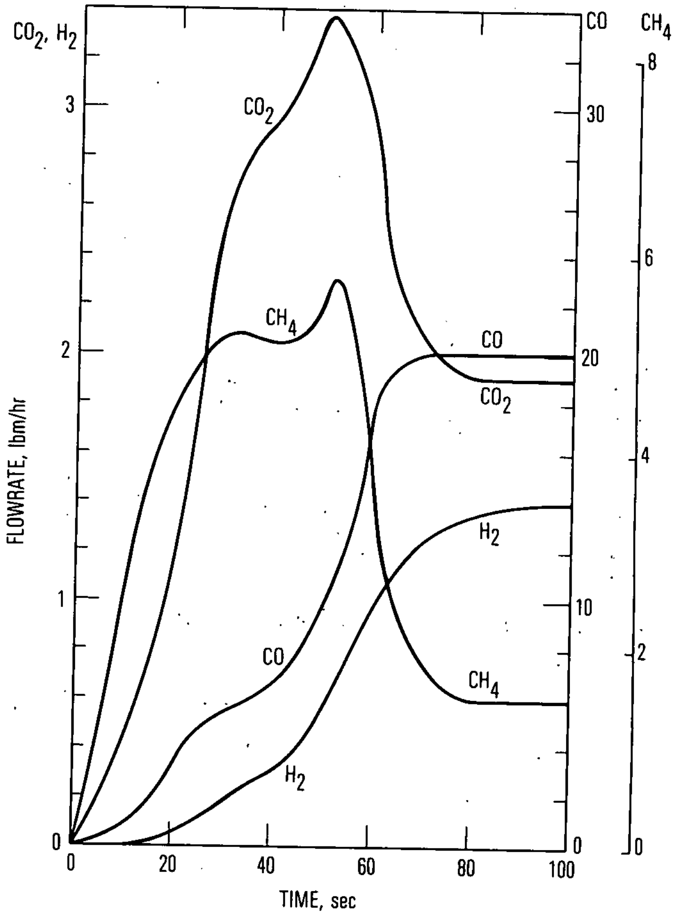


Figure 4-32. Product gas output flow rates during warmup of JPL fuel reformer (Ref. 4-1)

driveability can be answered. However, some general comments regarding the operation of vehicles using a fuel reformer can be made. First, the simplest approach to starting such a vehicle is to use gasoline alone until the reformer has warmed up. Starting the vehicle on gaseous fuels from the reformer is likely advantageous from the standpoint of emissions and fuel economy, but the time involved after initiating the startup may be too long to be practical. Second, control of the gasoline flow to the reformer during

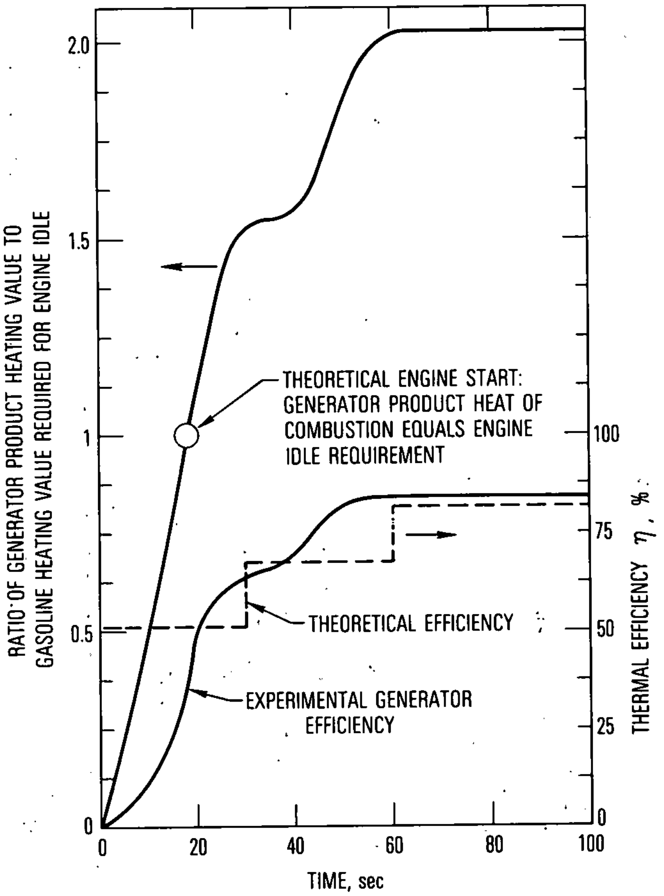


Figure 4-33. Energy content of product gases and thermal efficiency during warmup of JPL fuel reformer (Ref. 4-1)

various vehicle operating modes undoubtedly requires a control system much more complex than that currently in use on automobiles. In this regard, a reasonable approach may be to simply divide the gasoline fed to the engine and reformer in a prescribed ratio depending only on the air-fuel ratio desired in the various vehicle operating modes (e.g., idle, full throttle, part-throttle, or cruise). Whether or not relatively simple reformer starting and control approaches are possible must await the results of future development efforts.

REFERENCES FOR SECTION 4

4-1. "JPL Final Report on EPA-Supported Studies Related to Ultralean Combustion in Spark Ignition Engines with Hydrogen Addition," Rough Draft, Jet Propulsion Laboratory, Pasadena, Calif. (November 1974).

4-2. R. F. Stebar and F. B. Parks, "Emission Control with Lean Operation Using Hydrogen-Supplemented Fuel," SAE Paper No. 740187, Detroit (March 1974).

4-3. R. C. Lee and D. B. Wimmer, "Exhaust Emission Abatement by Fuel Variations to Produce Lean Combustion," SAE Paper No. 680769.

4-4. "JPL Engine Dynamometer Data for the Stock Engine and Autotronics Modified Engine Operated on Gasoline and the Autotronics Modified Engine Operated on Gasoline/H_2 Mixtures," Private Communication.

4-5. F. W. Hoehn, "Combined Dynamometer Test Results of the V-8 Engine and Hydrogen Gas Generator," Internal Memo to H. E. Cotrill, Jet Propulsion Laboratory, Pasadena, Calif. (23 January 1974).

4-6. F. W. Hoehn and M. W. Dowdy, "Feasibility Demonstration of a Road Vehicle Fueled with Hydrogen-Enriched Gasoline," Paper presented at the 1974 Energy Conversion Conference, San Francisco.

4-7. "Exhaust Temperature Data from JPL," Letter to J. Meltzer, The Aerospace Corporation, from J. Stocky, Jet Propulsion Laboratory (26 September 1974).

4-8. M. W. Dowdy and T. G. Vanderbrug, "Lean Mixture Engine Testing and Evaluation Program," Report to Department of Transportation (Draft), Jet Propulsion Laboratory, Pasadena, Calif. (September 1974).

4-9. M. W. Dowdy, "Experimental Data for Maximum Indicated Horsepower" (Curve dated 11 January 1974), Private Communication from Jet Propulsion Laboratory, Pasadena, Calif.

4-10. G. A. Karim and S. R. Klat, "The Knock and Auto Ignition Characteristics of Some Gaseous Fuels and Their Mixtures," Journal of the Institute of Fuel (March 1966), pp. 109-119.

4-11. W. F. Anzilotti and V. I. Tomsic, "Combustion of Hydrogen and Carbon Monoxide as Related to Knock," Fifth International Symposium on Combustion, p. 356 (1955).

4-12. W. Frie et al., "Compact Gas Generator for the Economic Operation of Internal Combustion Engine with Low Pollutant Emissions," Paper presented at Second Symposium on Low Pollution Power System Development, Dusseldorf, Germany (November 1974).

4-13. "Hydrogen Enrichment for Internal Combustion Engines," Briefing presented to Advanced Automotive Power System, Division of the Environmental Protection Agency, by Jet Propulsion Laboratory (3 December 1974).

SECTION 5

EFFECTS OF ENGINE DESIGN CHANGES ON LEAN OPERATION WITH HYDROGEN ADDITION

The spark ignition engines used in the JPL and GM vehicle studies of ultralean combustion using hydrogen addition were standard production V-8 engines which were not designed to be optimum for such operation. Hence, it is of interest to consider how the engines can be altered to improve the emission and fuel economy characteristics of an engine reformer system. Two types of design changes are discussed in this section: (1) an increase in compression ratio above that used for burning gasoline alone and (2) modifications of the fuel induction and ignition systems to reduce the hydrogen requirements for ultralean operation. In addition, the effect of engine CID on fuel economy and vehicle performance when a reformer is used is considered briefly.

5.1 HIGHER COMPRESSION RATIO

As indicated in Section 4.7, the knock resistance of fuel mixtures containing hydrogen, methane, and carbon monoxide is greater than for gasoline alone. Hence, it is possible to increase the compression ratio of engines used in combination with a fuel reformer without encountering engine knock.[10] The most direct effect of increasing the compression ratio is an improvement in the indicated thermal efficiency of the engine. For example, if the compression ratio is increased from 8.5 to 10.0, an improvement in thermal efficiency of 5% (Figure 5-1) can be expected. This improvement in thermal efficiency results in a corresponding increase in fuel economy and maximum horsepower at all rpm. Data for a 1969 V-8 (383 CID) engine are shown in Figure 5-2. Since the 5% improvement in full-throttle horsepower resulting

---

[10] Unless the overall system concept includes the option of all-gasoline operation.

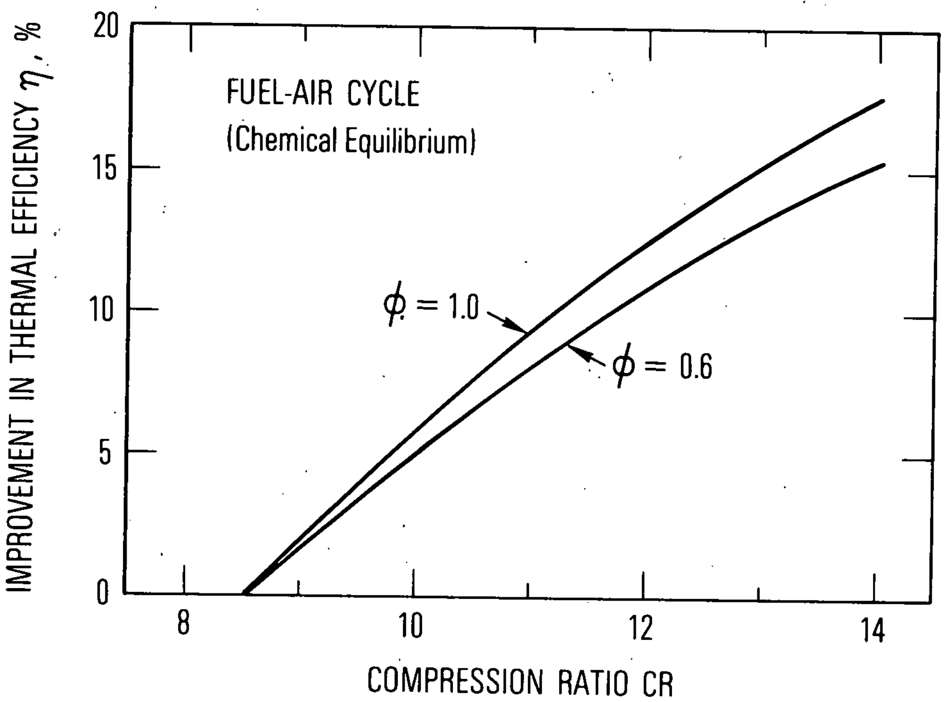


Figure 5-1. Effect of compression ratio on indicated thermal efficiency (Ref. 5-1)

from increasing the compression ratio is only a fraction of the horsepower loss due to lean operation and the use of gaseous fuels, increasing the compression ratio from 8.5 to 10.0 only partially alleviates the power loss problem discussed in Section 4-6.

The question naturally arises as to how increasing the compression ratio affects exhaust emissions. Of particular interest are the HC and NO_x emissions since the CO emissions depend primarily on air-fuel ratio and are not expected to change with compression ratio. Studies (Ref. 5-2) of the effect of compression ratio on emissions have been made for near-stoichiometric combustion (Tables 5-1 through 5-3 and Figure 5-3), but no test data are available for ultralean conditions. JPL has, however, predicted NO_x emissions (Figure 5-4) at lean operation, using a computer program (Ref. 5-3). The available information shown in the figures indicates that

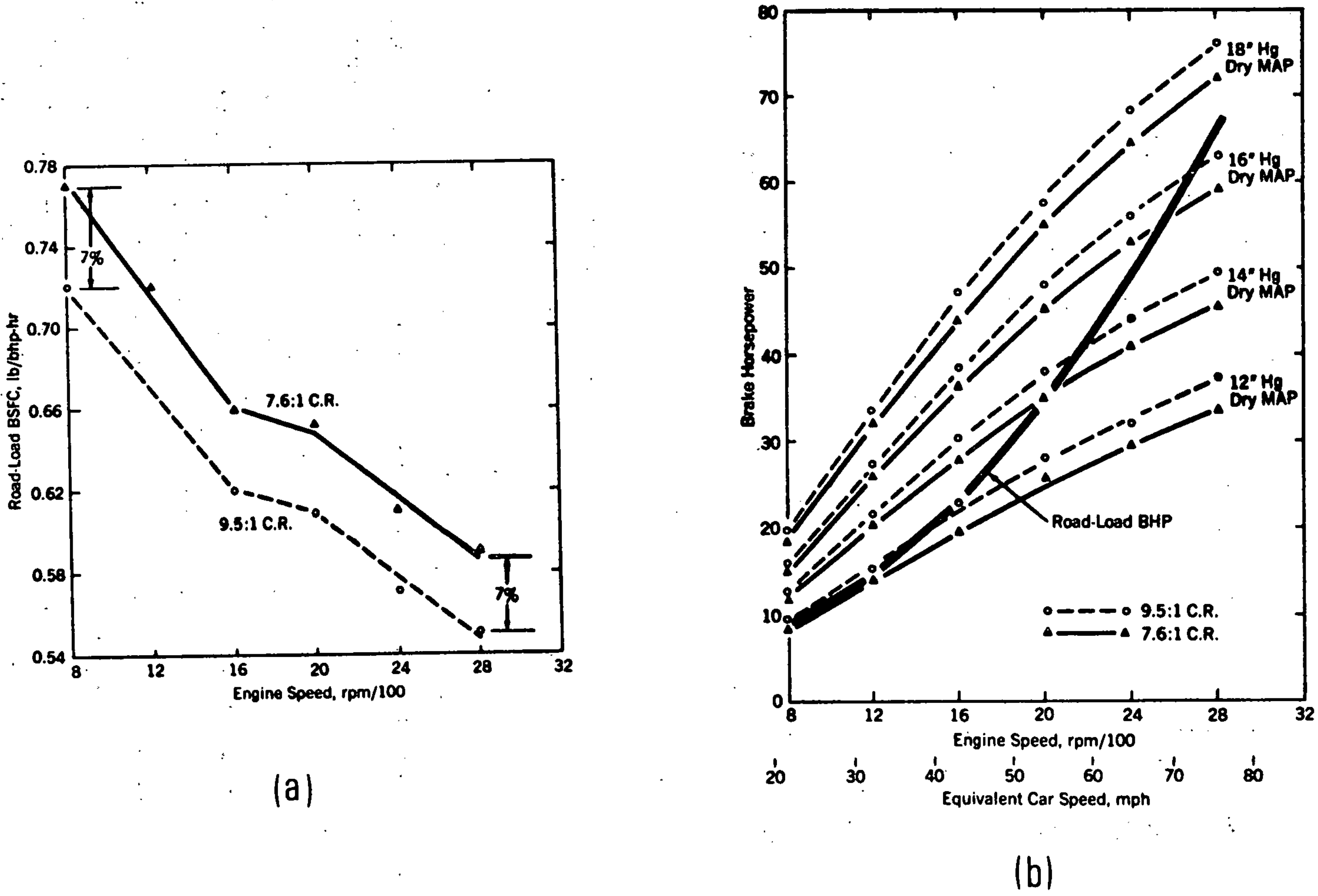


Figure 5-2. Effect of compression ratio on (a) road-load fuel consumption and (b) full-throttle bhp: dynamometer tests, MBT ignition timing, identical carburetion for both CRs (Ref. 5-2)

Table 5-1. Effect of Compression Ratio on NO_x and HC Emissions from a CFR Engine[a, b]

IHP	Equiv. Ratio A/F	NO_x, gm/hr: CR			HC (FID), gm/hr: CR		
		11:1	9:1	7:1	11:1	9:1	7:1
5.0	0.90	48.9	46.0	41.3	1.05	0.69	0.44
5.0	0.95	42.1	43.1	39.8	1.19	0.83	0.47
5.0	1.00	29.5	29.0	30.5	1.63	1.22	0.85
6.0	0.90	65.2	65.4	61.5	1.19	0.85	0.53
6.0	0.95	57.0	58.5	55.4	1.37	0.99	0.59
6.0	1.00	40.1	41.2	41.8	1.91	1.44	1.00
7.0	0.90	82.6	87.2	85.2	1.26	1.03	0.61
7.0	0.95	73.4	75.4	73.7	1.50	1.15	0.74
7.0	1.00	51.8	53.5	54.5	2.12	1.65	1.15
8.0	0.90	100.7	106.7	110.6	1.26	1.22	0.70
8.0	0.95	90.5	90.8	94.6	1.55	1.31	0.89
8.0	1.00	64.1	63.8	69.1	2.27	1.85	1.29
Average	—	62.1	63.4	63.2	1.52	1.19	0.77

[a] Ref. 5-2.

[b] Single-cylinder, variable compression ratio (CR), 41.4-CID RDH engine, 1800 rpm, 212°F jacket, 150° intake air, MBT ignition timing, isooctane fuel.

Table 5-2. Average Mode-By-Mode Analysis of HC and NO_x Data[a]

	9.5:1 CR		7.6:1 CR	
	HC	NO_x	HC	NO_x
Hot cycles, no.	20	12	16	16
Emissions, ppm				
Idle	5	5	7	4
0-30 mph	33	322	36	291
30 mph	15	96	15	90
30-15 mph	7	16	14	15
15 mph	6	11	6	10
15-50 mph	58	810	62	791
50-0 mph	27	5	34	5
Total	151	1265	174	1206

[a] Ref. 5-2.

Table 5-3. Effect of Compression Ratio on Car Emissions[a,b]

Test Data	9.5:1 CR						7.6:1 CR				
Testing laboratory	Ethyl	Ethyl	Ethyl	Eng. Mfr.	Ethyl	Avg.	Ethyl	Ethyl	Ethyl	Ethyl	Avg.
Date (1970)	3-19	4-16	4-17	4-23	5-22	—	2-23	6-17	7-1	7-2	—
Basic ignition timing, °BTC	3	3	3	3	3	—	5	5	5	5	—
Idle CO in drive, %	1.07	1.10	0.70	1.0	1.0	0.99	0.90	1.4	1.25	0.75	1.08
Idle speed in drive, rpm	535	530	515	530	530	—	535	550	540	550	—
7-Mode hot cycles											
HC (NDIR), ppm	165	154	136	148	151	151	162	192	165	178	174
CO, %	0.89	0.47	0.55	0.83	0.61	0.67	0.58	0.91	1.01	0.73	0.81
NO, ppm	1179	1384	1293	1203	873[c]	1265	1308	1193	1004	1317	1206
Bag data											
HC (NDIR), ppm	529	449	418	—	524	480	534	487	487	485	492
CO, %	1.06	0.69	0.75	—	0.80	0.82	0.88	1.07	1.20	0.86	1.00
NO, ppm	1340	1313	1296	—	1069[c]	1319	1451	1215	1089	1344	1275

[a] Ref. 5-2.

[b] 1969 383-CID V-8 engine, 7-Mode FTP, hot cycles, identical carburetion for both CRs, 98% MBT ignition timing.

[c] Not included in averages.

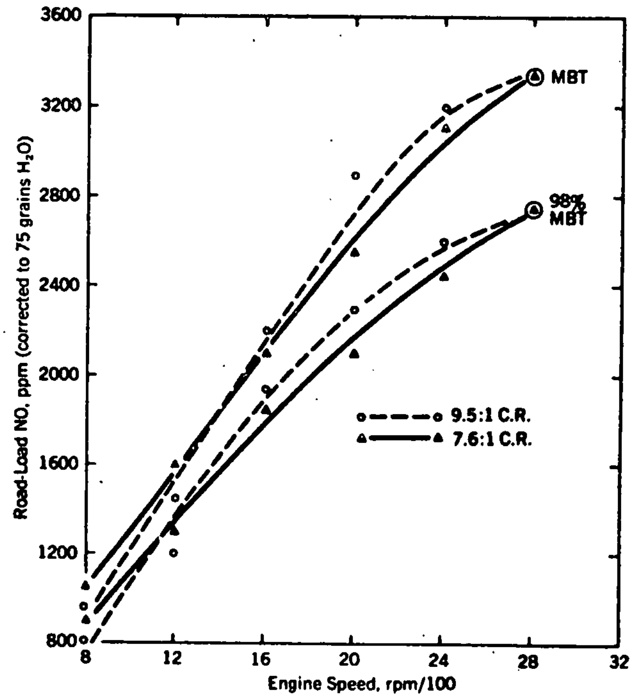


Figure 5-3. Effect of compression ratio on NO_x emissions from a V-8 engine: dynamomemter tests, road-load bhp, identical carburetion for both CRs (Ref. 5-2)

NO_x emissions at a given power level tend to increase only slightly due to an increase in compression ratio, which indicates that the effect of the higher peak cycle temperature is nearly cancelled by the effect of the smaller spark advance used at the higher compression ratios. For a 2-unit increase in compression ratio, NO_x emissions have been found to increase by 10% or less. The situation concerning HC emissions is somewhat more uncertain as emission tests on CFR engines (Table 5-1) consistently show a significant increase in HC emissions at higher compression ratios for steady-state engine operation. At the same time, HC emissions from a V-8 engine averaged over a test cycle, such as the 7-mode Federal Test Cycle (FTC) which includes both transient and steady-state modes, have been found to be slightly lower at higher compression ratios (Table 5-3). This anomaly and

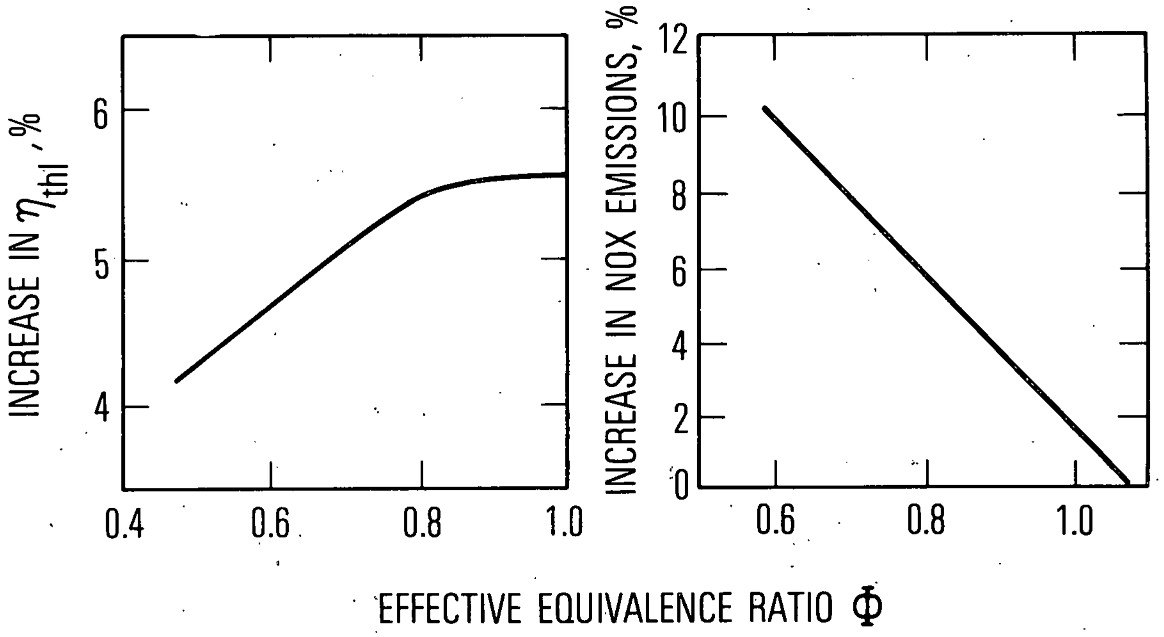


Figure 5-4. Predicted effects on engine indicated thermal efficiency and NO_x emissions for a compression ratio increase of from 8.5 to 10.0 (Ref. 5-3)

the fact that the HC emission depends both on what occurs within the cylinder and just downstream of the cylinder exhaust ports (postexhaust oxidation) makes any prediction of the effect of compression ratio on HC emissions uncertain at this time. If, however, as is likely to be the case, a catalytic reactor is needed to meet the 1977 HC emission standard (0.4 gm/mi) with ultralean combustion, this uncertainty in the effect of compression ratio on HC emissions appears to be of minor significance and should not be a deterrent to using a higher compression ratio engine with the fuel reformer.

Hence, it can be concluded that increasing the engine compression ratio from 8.5 to 10 is a reasonable approach and could result in about a 5% improvement in both vehicle fuel economy and performance without a significant effect on emissions.

5.2 IMPROVEMENTS IN LEAN LIMIT HYDROGEN REQUIREMENTS

Probably the most significant design changes that could be made in the engine are those which would result in reducing the hydrogen flow rates required for ultralean operation. What these design changes should be are not well defined at the present time, but they are currently the subject of research at JPL. The objective of design changes intended to improve the lean limit hydrogen requirements are, however, reasonably well understood. They include (1) better vaporization of the gasoline fed directly to the engine, (2) reduction of cylinder-to-cylinder nonuniformities in air-fuel ratio, (3) greater turbulence in the cylinder during the burning of the ultralean mixture, and (4) an improved ignition system to obtain a hotter spark.

JPL has made computer predictions (Ref. 5-3) of the effect on fuel economy and NO_x emissions resulting from improvement of the lean limit hydrogen requirements. Predictions have been made for what are termed 50% and 75% improvements. The percent of improvement is based on the difference between the current lean limit requirement and a theoretical lean limit requirement which has been calculated from flamability limit data. In essence, the lean limit improvements imply a reduction in the hydrogen weight fraction needed for combustion at a given lean equivalence ratio. As indicated in Table 5-4, when expressed in terms of a reduction in hydrogen weight fraction, a 50% improvement in the lean limit requirements means a 55% to 60% reduction in required hydrogen flow rate, and a 75% improvement means a 75% to 80% reduction in hydrogen flow rate. Hence, for a given maximum hydrogen flow rate capability (lb/hr), improving the lean limit requirement means that an ultralean equivalence ratio can be maintained for higher engine loads and rpm than is the case for the current lean limit requirements.

The predicted effect of lean limit improvements on fuel economy and NO_x emissions is given in Table 5-5 for two maximum hydrogen flow rates, 0.5 and 1.5 lb/hr. The values shown were calculated by JPL, using a computer program. Even though the magnitude of the calculated values

Table 5-4. Effect of Lean Limit Improvements on Hydrogen Weight Fraction Requirements at Ultralean Equivalence Ratios[a]

Equivalence Ratio[b]	Stated Improvements in Lean Limit Requirements, %[c]	Hydrogen Flow Requirements	
		Weight %	Reduction, %
0.5	0	0.100	0
	50	0.043	58
	75	0.027	74
0.6	0	0.050	0
	50	0.019	62
	75	0.009	82

[a] Ref. 5-3.

[b] Effective equivalence ratio as defined by JPL.

[c] Improved lean limit curves (Fig. C-7 in Ref. 5-3) drawn at stated % of horizontal difference between current lean limit and theoretical lean limit curves.

for a particular engine configuration may be in error, it is likely that the predicted relative changes from one configuration to another are reasonably reliable. Several conclusions can be drawn from Table 5-5. First, the effect of lean limit improvements on fuel economy is quite small. Second, the effect of lean limit improvements on NO_x emissions is quite large. For example, if the lean limit hydrogen requirements could be improved by 50%, a hydrogen flow rate of 0.5 lb/hr would be adequate to reduce the NO_x emission level to about 0.3 to 0.4 gm/mi; while using the current lean limit requirements, the same hydrogen flow rate would likely result in NO_x emissions in excess of 1 gm/mi.

5.3 THE EFFECT OF ENGINE CID ON FUEL ECONOMY AND EMISSIONS

As discussed in Section 4.6, lean operation of an engine results in a significant reduction in full-throttle horsepower and a loss in vehicle

Table 5-5. Effect of Lean Limit Improvements on Fuel Economy and NO_x Emissions[a]

Engine Configuration	Hydrogen Flow, lb/hr	Fuel Economy, mpg[b]	NO_x Emissions gm/mi[b]
Stock	—	12.1	1.8[d]
Modified	—	13.8	—
Modified, reformer	0.5	14.4	1.38
	Controlled[c]	14.0	0.50
Modified, reformer	0.5	14.9	0.27
• Lean limit improved 50%	Controlled	14.6	0.27
Modified, reformer	0.5	15.1	0.29
• Lean limit improved 75%	Controlled	14.8	0.29
Modified, reformer	0.5	15.5	0.20
• Lean limit improved 75%, CR = 10.0	Controlled	15.2	0.20

[a] Ref. 5-3.

[b] Predicted by JPL computer program; fuel economy and emissions on the FDC.

[c] Hydrogen flow rate varied from 0.5 to 1.5 lb/hr.

[d] Measured on the 1975 FTP.

performance. One might attempt to regain the lost performance by increasing the size of the engine (CID) used in the vehicle. It is of interest to consider how using a larger engine affects the fuel economy and emissions of the vehicle.

Fuel economy (1975 FTP) data for 1973 vehicles are given in Figure 5-5 as a function of engine displacement. For a given weight vehicle, the fuel economy decreases with increasing engine CID. For example, changing from a 350-CID engine to a 450-CID engine in a 4000-lb vehicle results in a fuel economy decrease from 10.2 to 8.5 mpg or a loss of 17%. This decrease in fuel economy is due primarily to the decrease in engine mechanical efficiency inherent in part-throttle operation and, as shown in the next paragraph, this same trend can be expected when the engine is operated at ultralean conditions.

The mechanical efficiency η_m is defined as

$$\eta_m = 1 - \frac{\text{fhp}}{\text{ihp}} = 1 - \frac{\text{fmep}}{\text{imep}}$$

where

fmep ≡ friction mean effective pressure
imep ≡ indicated mean effective pressure

The indicated mean effective pressure is essentially independent of engine CID, and rpm and can be written (Ref. 5-5) as

$$\text{imep (psi)} = 555\, \eta_i\, \phi \tau\, e_{\text{veff}}$$

where

η_i ≡ indicated thermal efficiency
ϕ ≡ equivalence ratio
τ ≡ throttle fraction
e_{veff} ≡ effective volumetric efficiency

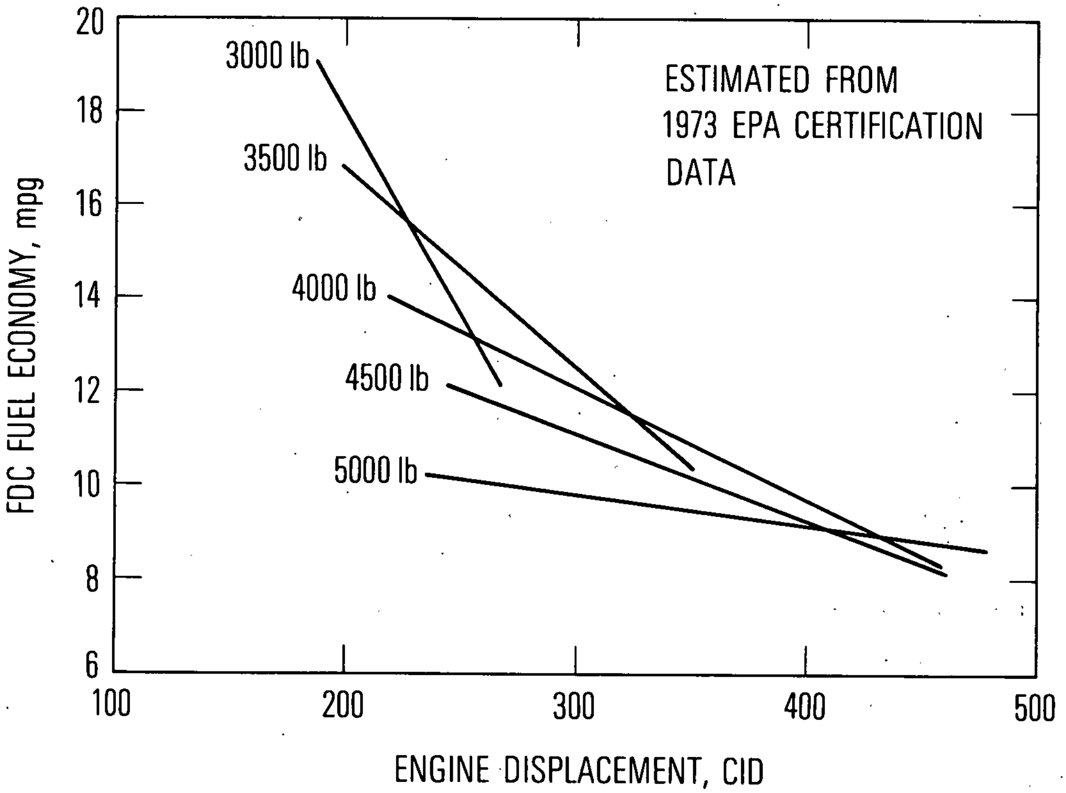


Figure 5-5. Fuel economy as a function of engine displacement (Ref. 5-4).

The friction mean effective pressure is primarily a function of piston speed and thus rpm and depends only weakly on engine size and throttle position. Hence, at a given rpm, fmep is essentially constant:

$$\eta_m = 1 - \frac{C_1(\text{rpm})}{C_2 \eta_i \phi \tau e_{\text{veff}}}$$

with C_1 (rpm) and C_2 being treated as constants. It follows from the expression for η_m that, at full throttle, η_m depends for a fixed rpm only on equivalence ratio independent of engine size, but that for a fixed load (hp), rpm and ϕ, η_m is lower with a larger engine because the larger engine must operate at less throttle opening than the smaller engine to generate the same power. Thus, lean operation of a large engine necessarily results in relatively low mechanical efficiency and is not favorable from a fuel economy point of view.

For operating conditions (equivalence ratio, load, and rpm) which can be reached by both the larger and smaller engines, the emissions of the two engines should be essentially the same; the HC and NO_x emissions of the larger engine might be slightly but not significantly lower. At ultralean operating conditions at which the smaller engine is power limited, the larger engine will permit ultralean operation, and hence lower NO_x emissions, over a wider range of conditions if the fuel reformer can meet the hydrogen flow requirements of the larger CID engine. Thus, to realize a significant improvement in emissions, particularly NO_x, using a larger engine will also likely require a larger fuel reformer than that optimized for a smaller engine.

REFERENCES FOR SECTION 5

5-1. M. H. Edson and C. F. Taylor, "The Limits of Engine Performance - Comparison of Actual and Theoretical Cycles," Digital Calculations of Engine Cycles, SAE TP7, Progress in Technology, Vol. 7 (1964).

5-2. A. E. Felt and S. R. Krause, "Effects of Compression Ratio Changes on Exhaust Emissions," SAE Paper 710831, SAE Meeting, St. Louis, Mo. (1971).

5-3. "JPL Final Report on EPA-Supported Studies Related to Ultralean Combustion in Spark Ignition Engines with Hydrogen Addition," Rough Draft, Jet Propulsion Laboratory, Pasadena, Calif. (November 1974).

5-4. Federal Certification (EPA), Light Duty Test Results for the 1973 Model Year, Private Communication from Environmental Protection Agency.

5-5. C. F. Taylor, "The Internal Combustion Engine in Theory and Practice," Vol. I, The Massachusetts Institute of Technology Press, Cambridge, Mass. (1960).

SECTION 6

COMPARISON OF FUEL REFORMER APPROACHES

Various onboard fuel reformer systems were discussed in detail in Section 3.2. The characteristics of the systems considered are summarized in Table 6-1 so that their advantages and disadvantages can be readily assessed and compared. The fuel reformers developed fall into two broad categories (partial oxidation and steam reforming) with several subgroups (thermal, catalytic, equilibrium, and nonequilibrium) in each category. A qualitative assessment of the relative attractiveness of the various reformer approaches is given in Table 6-2. Tables 6-1 and 6-2 show that the characteristics and relative attractiveness of the reformer approaches vary considerably and that no single approach in its present state of development is best in all respects; that is, low in complexity, high hydrogen yield with high thermal efficiency, no sooting problem, and proven durability. Hence, various tradeoffs must be considered in determining which fuel reformer approach is likely to be best for automotive applications.

Considerable work has been done on both the partial oxidation and steam reforming types of fuel reformers for automotive use since each type offers potential advantages which almost inherently cannot be attained with the other. The primary inherent advantage of the partial oxidation type is its simplicity in that the reforming process is simply a partial combustion of the same substances (gasoline and air) which are used in the engine. The inherent disadvantages of the partial oxidation approach are that (1) the thermal efficiency of the process is relatively low because part of the chemical heat content of the gasoline is released during the reforming reactions and (2) the product stream contains a high concentration of CO in addition to the desired hydrogen and further is highly diluted with nitrogen. The magnitude of these disadvantages depends primarily on the air-fuel ratio used in the reformer; the disadvantages become less significant as the

Table 6-1. Fuel Reformer Characteristics

Developer	Type	Size	Operating Conditions		Sooting	Product Gas			Fraction of Gasoline to Reformer[c]	Fuel Use, Thermal Efficiency, %[d]		Use of Waste Heat
			Temperature, °F	Pressure, psig		Composition, vol %	lb H_2/ lb Gasoline[a]	Gasoline, kcal/mole[b]		Reformer Alone	Reformer and Engine	
JPL	Partial oxidation (thermal)		2000-2400	30	Yes	22% H_2 23% CO 55% inerts	0.10 (experiment)	795	0.65	79.5 (theory)	86.7	Preheat reactants
JPL	Partial oxidation with catalyst A/F = 5	6-in. diam × 20-in. length incl. heat exchanger: 15- to 20-lb wt	1800	2	No	22% H_2 24% CO 1% CH_4 54% inerts	0.12 (experiment)	763	0.60	78.5 (experiment)	87.1	Preheat reactants
International Materials	Steam reforming with CO shift (no catalyst)		2000-2400 for cracking and 700-800 for CO shift	800	No (pulsed operation)	75% H_2 25% CO_2	0.42 (theory)	1214	0.192	96[e] (theory)	99	Preheat reactants; heat of reaction for cracking from fuel
Siemens	Partial oxidation with catalyst, nonequilibrium chemistry, A/F = 1.5, and EGR	4-in. diam × 15-in. length (estimate)	1500	0	No	13% H_2 13% CO 4% CH_4	0.12 (experiment)	1000	0.74	98	98.5	Preheat reactants
Phillips Petroleum	Steam reforming with catalyst		1000	25	Yes (prob.)	30% H_2 4% CO 23% CH_4	0.10 (theory)	1066	0.87	102	102	All heat required from engine waste heat

[a] All reformer performance data based on the use of Indolene fuel.

[b] Lower heating value used for all fuel gases.

[c] To attain 10% H_2 addition by weight; CO and CH_4 weight flow in reformer product gases converted to gasoline equivalent based on energy content.

[d] Thermal efficiency of reformer defined as ratio of the heat content of reformed fuel gases to heat content of gasoline alone.

[e] Part of reformer product fuel gases recycled to supply heat to the reforming process.

Table 6-2. Fuel Reformer Approaches: Advantages and Disadvantages

Approach	Complexity[a]	Hydrogen Yield	Thermal Efficiency	Sooting Problems[b]	Durability	Advantages	Disadvantages
Partial Oxidation							
• Thermal	High	Moderate	Low	Yes	Moderate	None	High operating temp., low thermal efficiency, sooting problems
• Catalyst							
/ Equilibrium	Low	Moderate	Low	No	Low[b]	Simplicity	Relatively low thermal efficiency
/ Nonequilibrium	Low	Moderate	Moderate	No	Low	Simplicity, relatively high thermal efficiency	Possible difficulties in transient operation
Steam Reforming							
• Thermal	High	High	Moderate	No	High	Durability, high hydrogen yield	Complexity, handling water
• Catalyst	Moderate	Moderate	High	Yes	Low	High thermal efficiency, low temperature operation	Sooting problems handling water

[a] The low, moderate, and high ratings are relative to the other reformer approaches listed.

[b] All catalyst systems presently have questionable durability.

air-fuel ratio is reduced. The inherent potential advantages of the steam reforming approach are that (1) with proper control of the process, the reformer products have a high concentration of hydrogen with CO_2 being the only other major constituent present, and (2) with proper use of the waste heat from the process, the thermal efficiency can be high (even greater than one) because the steam reforming reactions are endothermic, resulting in a product gas having a higher heat content than the input gasoline. The inherent disadvantage of the steam reforming approach is its complexity, resulting from the need to supply large quantities of water to the process and the likelihood that the attainment of optimum control requires two steps (thermal cracking and CO shifting). The overall complexity of the steam reforming process and the need to handle large quantities of water in the system have led both JPL and Siemens to pursue the partial oxidation approach. It is not likely that the steam reforming approach, especially using only thermal means to control the process, will be pursued in the future for automotive applications. However, there might be circumstances under which the catalytic steam reforming approach could prove to be attractive if a reforming catalyst could be developed which operated soot-free at 1000 to 1200°F.

Partial oxidation fuel reformers are currently being developed by JPL and Siemens. Both developments presently utilize a catalyst because it was found that using a catalyst permits satisfactory reformer operation at significantly lower temperatures than is possible without one, and in addition the catalyst reduces or eliminates the tendency for sooting at practical air-fuel ratios. The major difference between the approaches taken by JPL and Siemens is that JPL has designed its reformer so that chemical equilibrium is reached within the reactor, while in the Siemens reformer chemical equilibrium is not reached before the products exit the reactor, and nonequilibrium chemistry plays an important role in successful reformer operation. The result of this difference in design philosphy is that the Siemens reformer operates at an air-fuel ratio of ~ 1.5, and the JPL reformer operates at an air-fuel ratio of ~ 5.0. Both reformers are claimed to be soot-free. Siemens utilizes recirculation of a portion (20% to 30%) of the engine exhaust gases as a means of adding a small amount of water to the reformer reactants to aid in eliminating soot formation and to improve slightly the system thermal efficiency. The hydrogen yield (lb H_2/lb gasoline) of the JPL and Siemens reformers are comparable, and both generate about the same amount of CO. The thermal efficiency of the Siemens reformer is significantly higher (95% to 98% compared with 80%) than that of the JPL gas generator. The high thermal efficiency is certainly an attractive feature of the Siemens approach. It should be noted that, since the Siemens design depends on nonequilibrium chemistry to avoid soot formation at low air-fuel ratios, reformer product concentrations could be more sensitive to transient flow effects than the JPL design, which is based on chemical equilibrium at all operating conditions. It is claimed by Siemens that by varying the fraction of the engine exhaust recirculated, satisfactory reformer operation can be maintained over a wide range of engine operating conditions. Data for the Siemens reformer operated on a dynamic engine or vehicle cycle are not as yet available, however.

Another factor in the evaluation of the various reformer approaches is the fraction of the gasoline that must be processed by the reformer in order to generate the weight flow of hydrogen required to permit ultralean engine combustion. If it is assumed that a 10% weight fraction of hydrogen is needed, use of the partial oxidation approach requires that between 60% and 75% of the gasoline must be passed through the reformer (Table 6-1). These large fractions result from the relatively low hydrogen yield of the partial oxidation reformers and the large quantity of CO, and in the case of the Siemens reformer a significant amount of CH_4, also, in the reformer products. Hence, for what are presently thought to be the most practical reformer designs, most of the gasoline used by the vehicle is processed by the reformer, and the engine operates for the most part on gaseous fuels (H_2, CO, CH_4). Thus, the question arises as to whether it would not be advantageous to process all the gasoline through the reformer and operate the vehicle only on gaseous fuels. The advantages and disadvantages of partial and total fuel reforming are listed in Table 6-3.

Table 6-3. Advantages and Disadvantages of Partial and Total Fuel Reforming

Approach	Advantages	Disadvantages
Partial fuel reforming	Slightly smaller reformer Flexibility of dual-fuel operation Easier cold startup Less power loss	Complexity of dual-fuel operation Higher emissions (especially HC and NO_x) Longer engine warmup time Higher fuel octane requirement or lower engine compression ratio Lower fuel economy
Total fuel reforming	Simplicity of only gaseous fuel operation Quicker engine warmup Lower emissions (especially HC and NO_x) Lower fuel octane requirements and higher engine compression ratio Better fuel economy	Greater power loss Larger reformer and need for high thermal efficiency More difficult cold start Less flexible operation with a single-fuel system

SECTION 7

ENGINE-FUEL REFORMER SYSTEM CONSIDERATIONS

In this section, the overall engine-fuel reformer system is discussed, and selected system design requirements and performance characteristics are presented.

7.1 ENGINE-FUEL REFORMER SYSTEM

A schematic drawing of the JPL engine-fuel reformer system, showing the air, fuel (gasoline), and reformer product flows, is given in Figure 7-1. Details of the design and operation of several other types of fuel reformers have been discussed in previous sections, but they all fit into the system in much the same way as the JPL reformer. The system operates

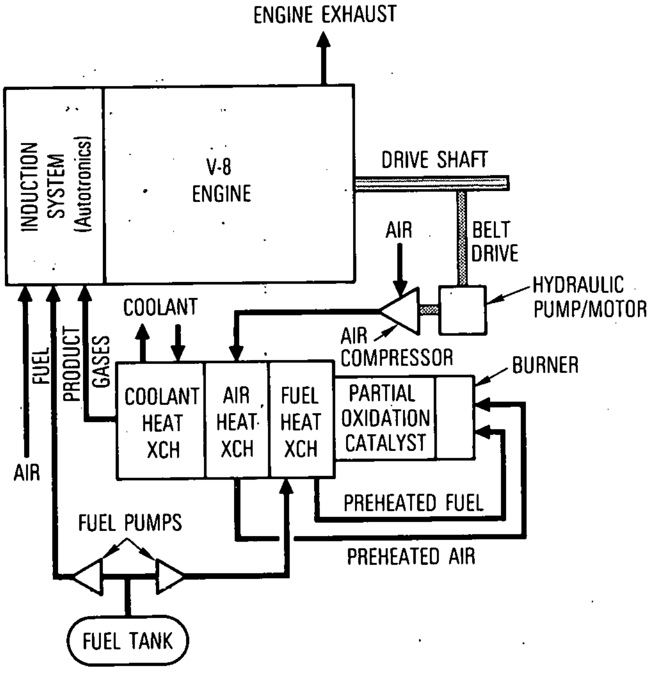


Figure 7-1. Schematic of JPL V-8 engine-fuel reformer system (Ref. 7-1)

essentially as a dual-fuel system with the gaseous fuel being generated by the fuel reformer. The primary additional auxiliary unit which is run off the engine is the air compressor needed to pump air to and through the reformer.

7.2 AIR COMPRESSOR POWER REQUIREMENTS AND THE EFFECT ON FUEL ECONOMY

The power requirements of the air compressor (pump) needed to pump the gases to and through the fuel reformer can be calculated once the air flow (lb/hr) and pressure drop (lb/ft^2) through the system are known. The weight flow of air for a given hydrogen requirement follows directly from the hydrogen yield (lb H_2/lb gasoline) of the reformer and the air-fuel ratio at which the reformer is operated. JPL has measured the pressure drop (Ref. 7-1) through the various parts of the system and expressed the results in terms of the K-factor, where $K = \Delta P/(1/2 \rho V^2)$. The calculated air compressor system requirements are summarized in Table 7-1 for hydrogen flow rates

Table 7-1. Air Compressor System Requirements

Hydrogen Flow, lb/hr[a]	Fuel Flow, lb/hr[a]	Air Flow, lb/hr[a]	Pressure Rise (ΔP), psi[b]	Power to Air, hp	Power Absorbed from Engine hp[c]
0.5	4.2	26	1.5	0.033	0.073
1.0	8.3	43	4.1	0.150	0.33
1.5	12.0	62	8.5	0.45	0.99
2.0	15.0	78	13.4	0.90	1.98

[a] For the JPL fuel reformer.

[b] Based on experimentally determined pressure drop characteristics of the JPL reformer; $K/A^2 \approx 1000$.

[c] $\eta_c = 0.65$, $\eta_{HPM} = 0.7$.

between 0.5- and 2.0-lb H_2/hr. Both the pressure drop through the system and the mechanical power absorbed from the engine increase rapidly with hydrogen flow. For hydrogen flows between 1.0 and 1.5 lb/hr, the reformer compressor requirements are comparable to those of the air pump used to supply air to the catalytic converter on 1975 vehicles.

The effect of fuel reformer auxiliary loads and system losses (excluding the thermal losses in the fuel conversion process itself) has been predicted by JPL, using a computer program (Ref. 7-1). Fuel economy results are given in Table 7-2 for three cases: (1) no penalty due to reformer auxiliary systems, (2) nominal or best estimates of auxiliary loads and losses, and (3) maximum or upper bound limits for auxiliary loads and losses. In all cases, for hydrogen flow rates between 0.5 and 1.5 lb/hr, the reduction in the predicted vehicle fuel economy on the FDC is only 2% to 3%. The power to run the air compressor is by far the largest auxiliary load and accounts for most of the fuel economy penalty.

Table 7-2. Predicted Effect of Reformer Auxiliary Loads and System Losses on Vehicle Fuel Economy

Penalty	Hydrogen Flow, lb/hr	Predicted Fuel Economy, mpg[d]	Percent Reduction
None	0.5	14.69	-
	Controlled[c]	14.43	-
Nominal[a]	0.5	14.41	1.90
	Controlled	14.04	2.70
Maximum[b]	0.5	14.25	3.00
	Controlled	13.99	3.05

[a] Best estimate of auxiliary loads and losses.

[b] 10% to 20% greater losses than best estimate.

[c] Hydrogen flow varied from 0.5 to 1.5 lb/hr, depending on need.

[d] Fuel economy (FDC) predicted by JPL computer program (Ref. 7-1).

The nominal effect of the thermal efficiency (~ 80%) of the JPL fuel reforming process is to reduce vehicle fuel economy by ~ 8%. Hence, including both the effect of the thermal efficiency of the reformer and auxiliary loads and losses, an improvement in average engine thermal efficiency of 10% to 11% due to hydrogen addition (ultralean operation) is needed before a net gain in fuel economy can be realized.

7.3 SELECTED SYSTEM OPERATING CHARACTERISTICS

During the course of the evaluation of the V-8 engine-fuel reformer system, it became necessary to estimate various system operating characteristics in order to better interpret and understand the results of the detailed experimental studies and computer analyses made by JPL at various system operating points and driving cycles. For the convenience of the reader, some of the system operating characteristic curves developed are included with this report as Figures 7-2 through 7-6.

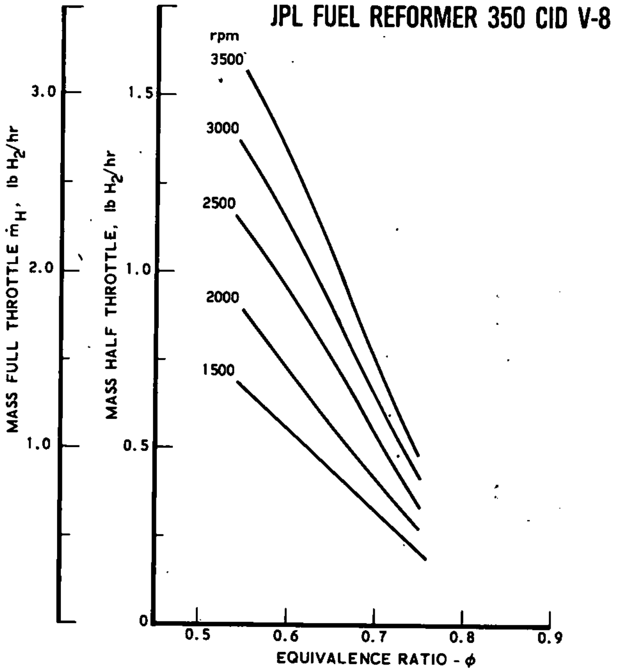


Figure 7-2. JPL system estimated full-throttle H_2 requirements

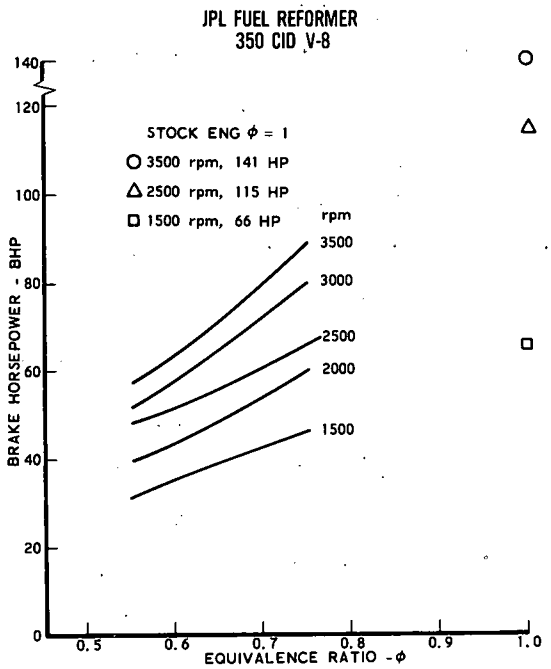


Figure 7-3. JPL system estimated full-throttle bhp

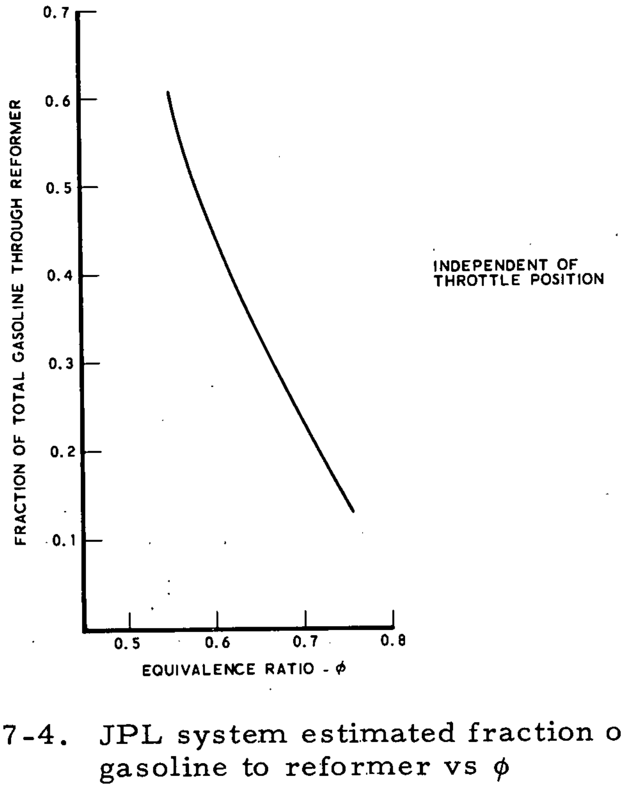


Figure 7-4. JPL system estimated fraction of gasoline to reformer vs ϕ

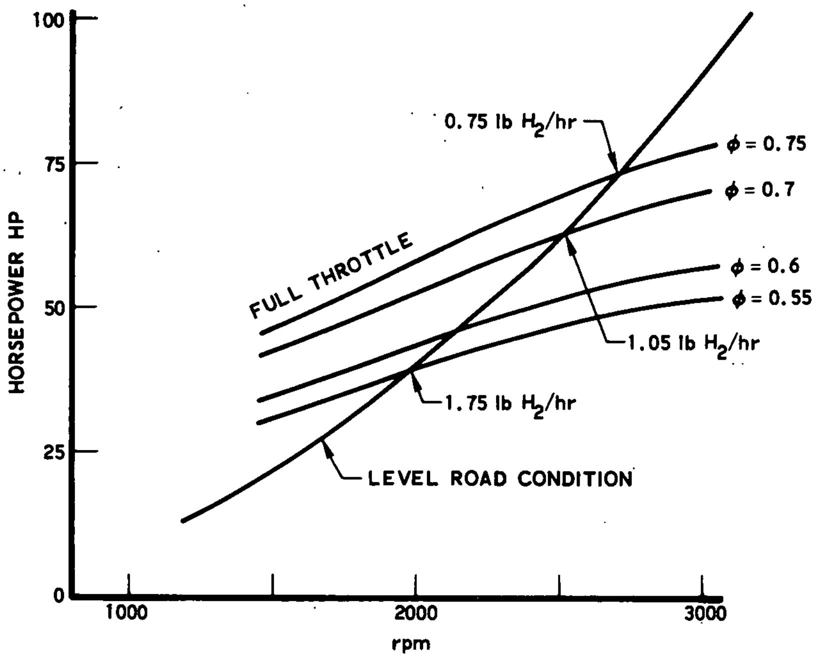


Figure 7-5. JPL system estimated level road load, equivalence ratio, and hydrogen flow matches

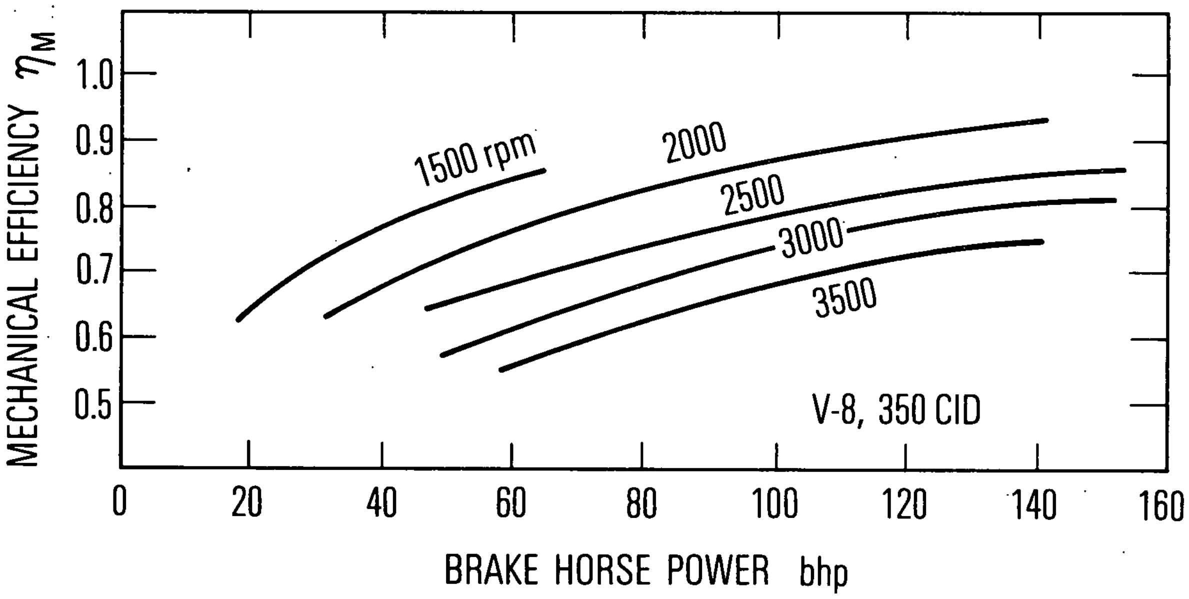


Figure 7-6. Engine mechanical efficiency vs rpm and bhp

REFERENCES FOR SECTION 7

7-1. "JPL Final Report on EPA-Supported Studies Related to Ultralean Combustion in Spark Ignition Engines with Hydrogen Addition," Rough Draft, Jet Propulsion Laboratory, Pasadena, Calif. (November 1974).

SECTION 8

OTHER LEAN COMBUSTION APPROACHES

As noted in Section 1, several other approaches which enable lean overall air-fuel ratio operation are under development. The more promising approaches are discussed in this section under two general categories: (1) modifications to conventional engines and (2) stratified charge engines.

8.1 MODIFICATIONS TO CONVENTIONAL ENGINES

A number of specific design modifications to the homogeneous charge spark ignition engine have been designed, tested, and evaluated. In general, they utilize improved carburetion and/or charge heating or warmup techniques to provide better fuel atomization and vaporization, better cylinder-to-cylinder charge distribution, and the ability to operate leaner than conventional automotive spark ignition engine systems.

To provide a basis of perspective, the section immediately following briefly reviews some of the inherent limitations of conventional carbureted systems. Next, brief discussions summarize the principles of operation and emissions and fuel economy characteristics of the Dresserator systems, the Ethyl carburetor, and the Ethyl turbulent flow manifold systems. These concepts are representative of the methods being pursued to overcome the limitations of conventional carburetion.

8.1.1 Inherent Limitations of Conventional Carbureted Systems

Ideally, the air-fuel mixture reaching the combustion chambers should be uniform from cycle to cycle and from cylinder to cylinder, and the fuel should be largely vaporized or finely atomized and uniformly mixed in the air stream. Real systems fail to achieve this ideal, and the cylinders receive air, vaporized fuel, atomized particles and droplets of liquid fuel, and (at part throttle) a liberal amount of exhaust residual. This departure from ideal conditions affects power, driveability, and fuel economy.

8.1.1.1 Atomization and Vaporization

Conventional venturi-type atomizers are capable of producing desired levels of atomization only at the high venturi air velocities obtained at high engine power levels. Directly at the jet of the carburetor, the ratio of the mass of air to the mass of vaporized fuel is high. However, as the mixture travels through the manifold, vaporization increases (heat is supplied), and the ratio of the mass of air to the mass of varporized fuel falls. Since the work of an engine is directly dependent on the mass of air inducted, complete vaporization of the fuel is not normally desired as the vaporized fuel would displace air and reduce power output. On the other hand, too little vaporization in the manifold may lead to poor distribution of the fuel from cylinder to cylinder. Although the carburetor delivers a fixed air-fuel ratio, the air-fuel ratio may vary greatly from cylinder to cylinder. A figure of 60% vaporization in the manifold at wide-open throttle is generally selected as a reasonable value for acceptable distribution and good power output capability. Normally, heat for fuel vaporization is supplied through a "hot spot" located directly beneath the throttle (Figure 8-1). The heavy droplets of fuel in the air stream impinge on the hot surface and are atomized and vaporized.

However, under cruising power conditions, more complete vaporization of the fuel is conducive to higher thermal efficiency (lower fuel consumption). For example, as shown in Figure 8-2, increasing the fuel vaporization from 60% to 100% reduces the brake specific fuel consumption by 16%.

8.1.1.2 Maldistribution Effects

The carburetor and manifold profoundly influence the distribution of fuel to the cylinders. One cause is the throttle plate, which at part throttle diverts the flow from the nozzle toward the wall of the manifold (Figure 8-1). In addition, flow passing the throttle plate sets up a low-pressure region on the underside of the trailing edge, tending to deflect

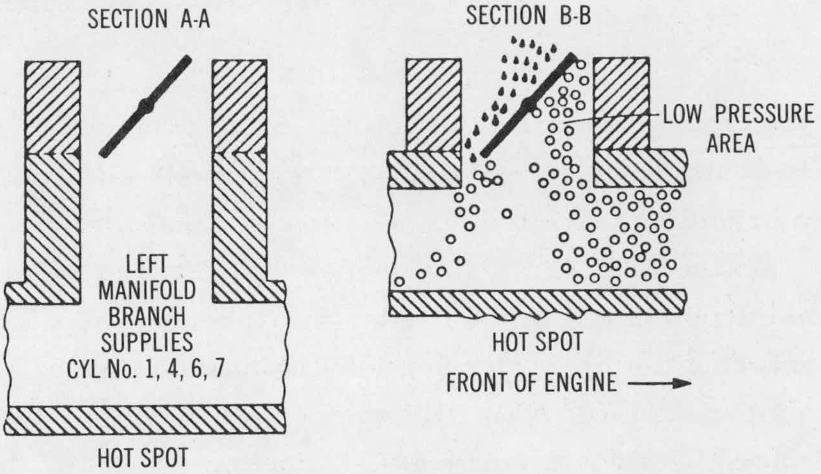


Figure 8-1. Cross sections of intake manifold of V-8 engines showing different heights of risers and throttle in worst position for distribution (Ref. 8-1)

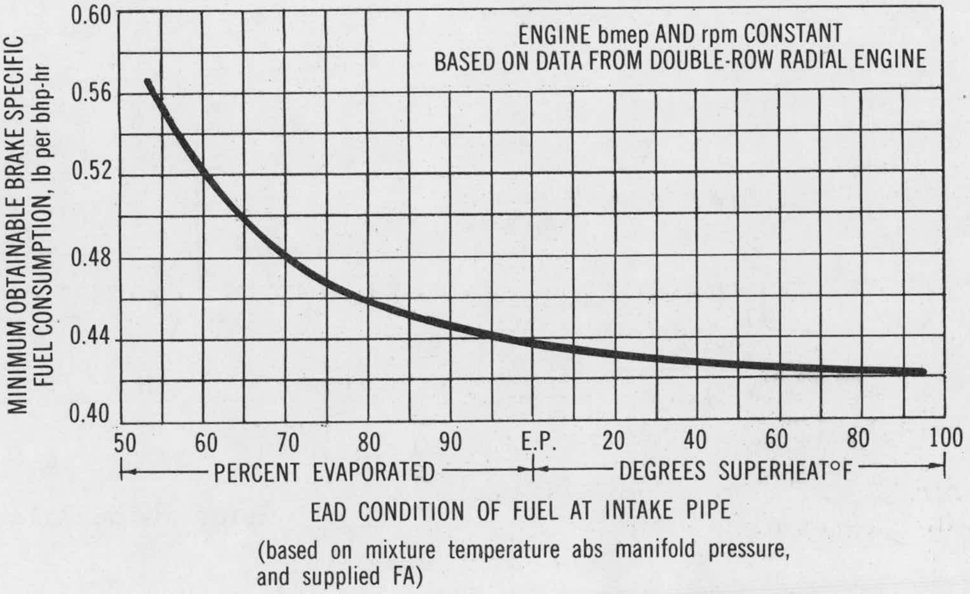
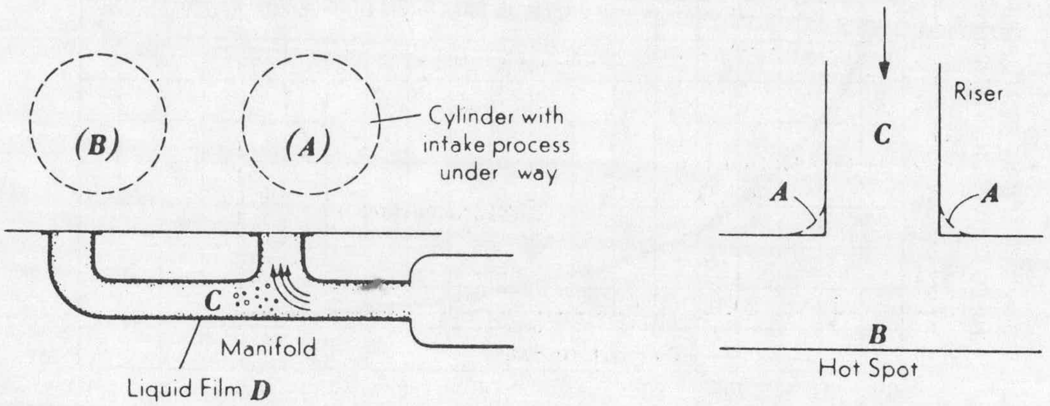


Figure 8-2. Example of typical correlation of minimum obtainable BSFC with equilibrium air distillation of fuel (Curve shown is typical of that found with large radial engines under cruising power conditions but is greatly influenced by engine design and operating conditions) (Ref. 8-1)

fuel toward the front cylinders (as shown). Slight changes in the position of any carburetor component (in particular, the throttle and choke) in the air stream can markedly change the distribution patterns.

In the case of the intake manifold (Figure 8-3a), the inertia of heavy liquid droplets (C) in the header may prevent them from turning the corner and entering the branch or leg leading to the cylinder (A). Also, the fuel flowing on the manifold wall (D) experiences the same difficulty (which can be minimized by smooth surfaces). As a consequence, the end cylinder (B) receives an overrich charge. The inverse problem exists at the riser (Figure 8-3b), where the charge descends into the header. Here the sharp 90-deg bend discourages a thick film and streamline flow and aids the drops and film to impinge on the hot spot.



(a) Division at a port (b) Division at the Riser hot spot

Figure 8-3. Fuel flow in manifold (Ref. 8-1)

Figure 8-4 shows the cylinder-to-cylinder air-fuel variations for both conventional carburetor and vaporizing tank operation for an 8-cylinder engine operating in the 30-mph cruise mode. Maldistribution is almost completely eliminated when the fuel is completely vaporized and completely mixed before entering the inlet manifold. The maximum air-fuel ratio spread with carburetor operation was 2.3. At other vehicle operating modes, the air-fuel ratio spread varied from 1.3 to 2.0 with the carburetor, and from 0.3 to 0.7 with vaporizing tank operation (Ref. 8-2).

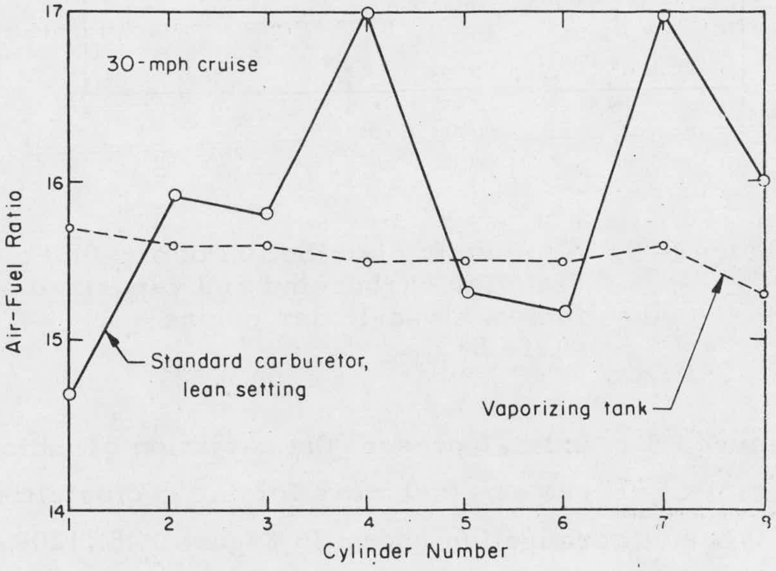


Figure 8-4. Geometric distribution of air-fuel ratio for carbureted and vaporized fuel in an 8-cylinder engine (Ref. 8-2)

Similar data for a 6-cylinder engine is shown in Figure 8-5. Geometric maldistribution is poor at wide open throttle with standard carburetion, but fairly good at part throttle (road load). Distribution with vaporized and premixed fuel is good at all operating conditions.

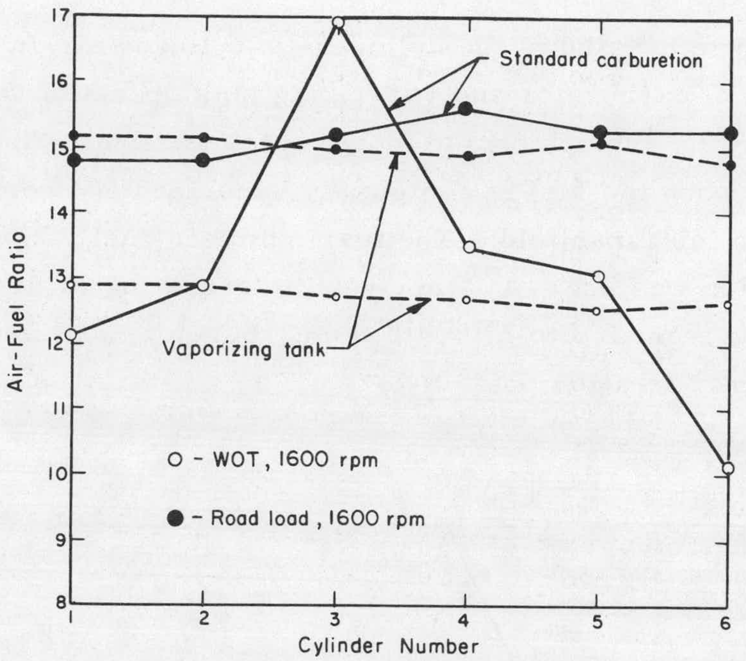


Figure 8-5. Geometric distribution of air-fuel ratio for carbureted and vaporized fuel in a 6-cylinder engine (Ref. 8-3)

Figures 8-6 and 8-7 present the variation of indicated specific fuel consumption (ISFC) versus air-fuel ratio for the geometric maldistribution with the two types of carburetion shown in Figure 8-5 (1200- and 2400-rpm cases and the 1600-rpm case). In this engine test series, there did not appear to be significant differences between standard carburetion and vaporizing tank operation. However, the range of air-fuel ratios is broader where vaporization tank carburetion gives near minimum specific fuel consumption. This broader range could simplify fuel metering for maximum economy (Ref. 8-3). These data (Figures 8-6 and 8-7) are for one engine only. Different engines, of course, could exhibit somewhat different characteristics, depending on the particular carburetor and intake manifold designs employed.

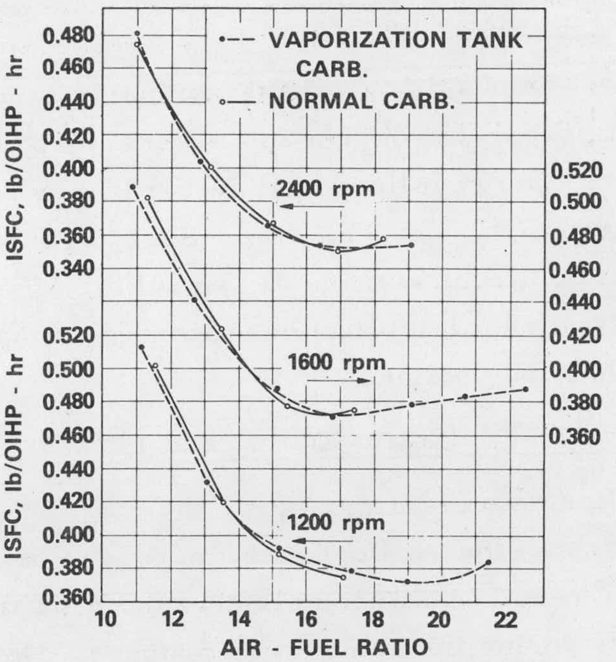


Figure 8-6. Effect of air-fuel ratio on indicated specific fuel consumption at WOT tank vs normal carburetion (Ref. 8-3)

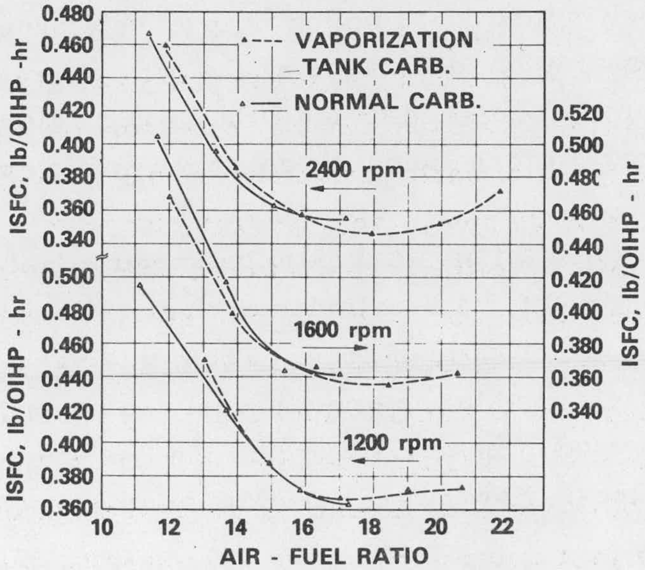


Figure 8-7. Effect of air-fuel ratio on indicated specific fuel consumption at WOT tank vs normal carburetion (Ref. 8-3)

8.1.1.3 Air-Fuel Ratio Accuracy

Recent emphasis on exhaust emission reduction has resulted in improvements in carburetor design and manufacture. Carburetor tolerances have been cut. The rich limit carburetor is now set only 6% richer than lean limit instead of 12% as formerly. Each carburetor is checked on a flow stand, and an adjustment is made to control the off-idle mixture ratio. Finally, the idle adjustment itself has been limited so that excessively rich mixtures at idle cannot be obtained.

8.1.1.4 Lean Air-Fuel Ratio Effects and Limits

As illustrated in Figure 8-8, the engine indicated thermal efficiency[11] increases as the air-fuel ratio increases or as the mixture becomes leaner. This results because an increased quantity of air decreases the temperature rise during the combustion process. However, the temperature and pressure rise (and, thus, work) per unit of fuel energy supplied are increased because the specific heats are lower at lower temperatures. Conversely, if excess fuel is present, it does not increase the work in proportion to the increase in fuel, and the efficiency decreases as the mixture is made richer. Since specific fuel consumption is inversely proportional to thermal efficiency, fuel consumption decreases as the air-fuel ratio increases until the lean mixture limit is reached or until the mixture flame speed is so low that the ignition spark timing cannot be advanced sufficiently to assure complete combustion.

The effect of air-fuel ratio on specific fuel consumption is illustrated in Figure 8-9 for a 6-cylinder engine. The best economy was obtained at a lean air-fuel ratio of 16.4. From this value up to the lean misfire limit (A/F = 21.5), the brake specific fuel consumption increased because of the long period of combustion resulting from slow flame speeds at the very lean mixtures. However, in tests of another 4-cylinder engine

---

[11] Thermal efficiency is the fraction of the heat energy supplied that is converted into work.

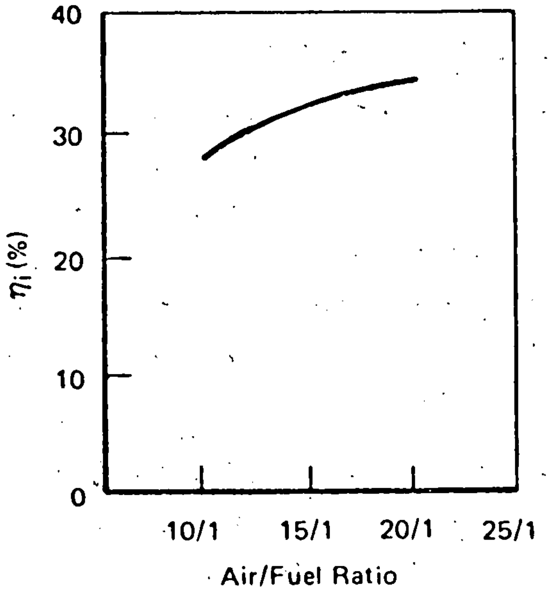


Figure 8-8. Effect of air-fuel ratio on indicated thermal efficiency at fixed compression ratio (example only)

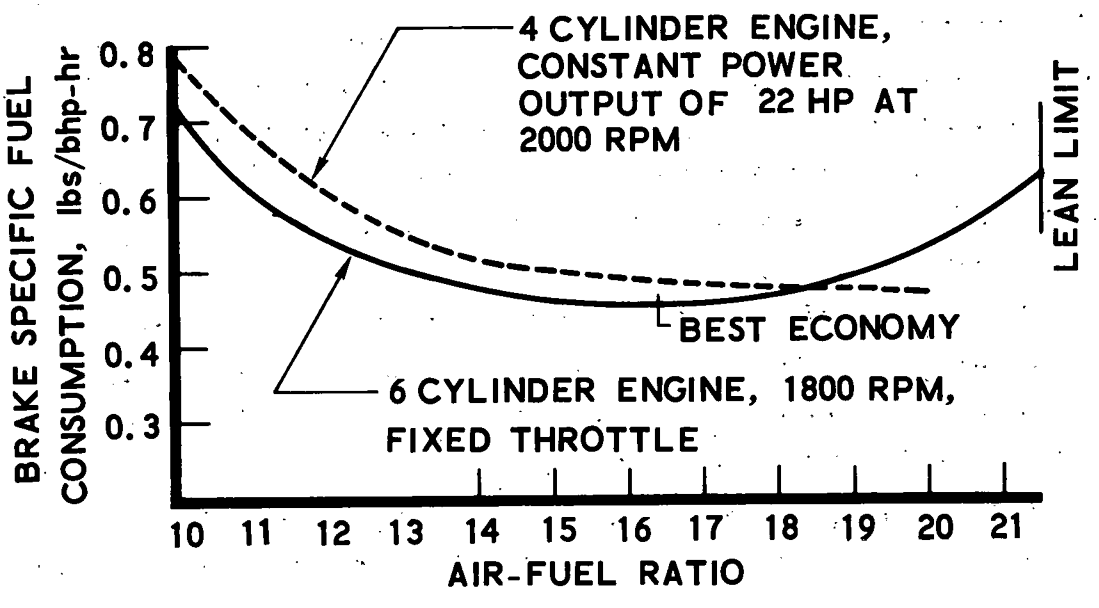


Figure 8-9. Effect of air-fuel ratio on specific fuel consumption

(Figure 8-9), the specific fuel consumption continued to decrease up to air-fuel ratios of 20. Of course, specific fuel consumption values at fixed steady-state conditions (such as shown in Figure 8-9) do not bear a one-to-one correlation with vehicle fuel economy as measured in mpg. They do, however, provide indicative trends and approximations of fuel economy changes at the given test condition.

Because of the potential for reducing exhaust emissions and at the same time increasing automotive fuel economy, a great many methods have been proposed to obtain "lean-mixture" operation (air-fuel mixtures leaner than those normally used in carbureted engines). In the fuel system area, these include heated intake air, fuel vaporizing, and dispersing devices. As noted earlier, lean operation reduces the specific power output at a given throttle setting; however, at part-load conditions a wider throttle opening can be used to obtain the desired power at the leaner mixture with a concomitant reduction in induction system pumping losses. These pumping losses represent the pressure drops experienced by the air in passing through the restrictions of the air cleaner, carburetor body, throttle valve, and inlet manifold. Since the throttle valve pressure drop is reduced by opening the throttle (less restriction), a wider throttle opening with lean operation at a given power output thus can reduce the overall induction system pumping losses.

8.1.1.5 Overall Effects

Unfortunately, direct data are not sufficient to quantify the overall effects of incomplete atomization, vaporization, and maldistribution on fuel economy in conventional carbureted spark ignition engines. Reference 8-3 test data (Figures 8-6 and 8-7) indicate that a relatively large maldistribution of fuel causes only slight losses in engine fuel economy but that improved distribution greatly extends the lean operating limit of the engine. In the case of lean-mixture operation, including the benefits of reduced pumping losses at part throttle operation, the fuel economy results are also uncertain because this type of operation is just now being experimentally evaluated and little data appear in the available literature. The test data shown in Figure 8-9 indicate approximately a 6% improvement in operating at an air-fuel ratio of 20 instead of 15. Schweitzer (Ref. 8-6) indicates a postulated improvement of 7% in fuel economy by operating at an air-fuel ratio of 22 instead of 15. However, as compared with emission-controlled vehicles operating with rich mixtures, estimates of fuel economy improvements (at the same emission levels) for lean-mixture operation range up to 30% (Ref. 8-6).

8.1.2 Dresserator System

The Dresserator system is a development of Environmental Technology, Santa Ana, California, a division of Dresser Industries, Inc., Dallas, Texas. The system, as described in Ref. 8-7, comprises a commercial air filter with a smoothed air flow path; an atomizing Dresserator core which, with a pressure fuel system, replaces the carburetor; a single plane intake manifold; and enlarged and insulated exhaust manifolds. Basically, the system is designed to permit engine operation at lean air-fuel mixtures, thereby providing emission control benefits and improved fuel economy.

The manufacturer states that the major portion of the emission benefits derived from this system comes from the improved combustion provided by the Dresserator carburetor. This device is a form of variable geometry venturi atomizer which is linked with a fuel metering apparatus, permitting control of the air-fuel ratio to some nominal level. The venturi is a mechanically activated variable area device designed to maintain the flow of air and fuel at the speed of sound through the throat over most of the operating range of the engine. Fuel is injected from a spray bar upstream of the venturi. Fuel flow rate and the venturi size are simultaneously controlled by linkage with the vehicle foot accelerator pedal.

Specific details concerning the configuration and operation of the components of the induction system are lacking. Figure 8-10 (Ref. 8-8) shows three successive design generations of the variable area venturi.

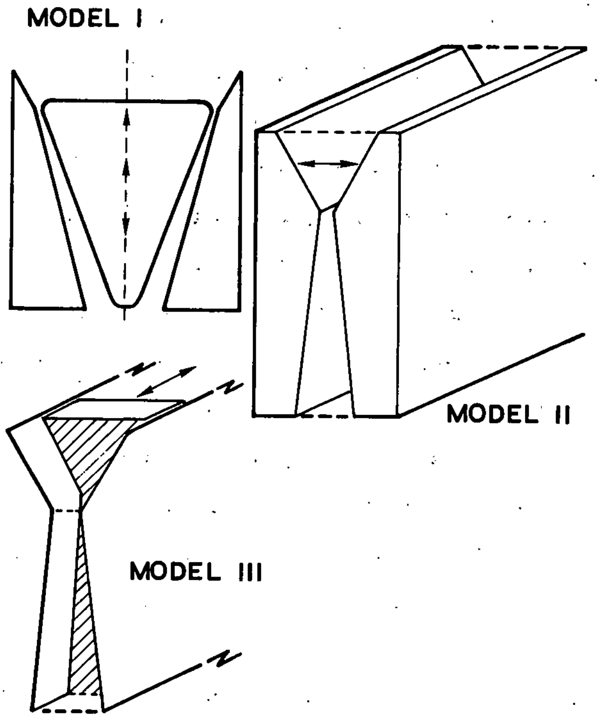


Figure 8-10. Dresserator models (Ref. 8-8)

Configuration III, the most advanced design, incorporates fixed jaws with a transverse sliding element which varies the flow area through the venturi. The manufacturer claims that the sonic feature of this design produces a very homogeneous air-fuel mixture which behaves like a colloidal suspension, producing minimum impaction on the walls of the intake manifold and providing more uniform cylinder-to-cylinder distribution. These factors contribute to the ability of the Dresserator system to operate at lean air-fuel ratios of from 18 to 19:1. The sonic feature is also asserted to permit very close control of air-fuel ratio as a result of the constant speed feature of the flow through an opening of known size acting as a mass flow indicator and control device (Ref. 8-7).

The manufacturer claims and confirming tests demonstrate that the device meets the 1975 California emission standards without the use of a catalytic converter. A number of Dresserator system emission tests have been made with results such as those shown in Table 8-1. All of these test data were reported by the manufacturer, except for entry No. 2, which shows the results of a confirming test conducted by the California Air Resources Board Laboratories (May 25, 1973). It is noted that the Dresserator testing has largely been conducted with disconnected vacuum advance. Although fuel economy has not been an initial object of study, the manufacturer claims a 5% to 10% improvement in fuel economy under these conditions and believes this performance could be improved with the vacuum advance optimized for the Dresserator system. Entry No. 7 shows the results of one of several Dresserator tests with the vacuum advance operative, indicating that some improvement by this technique may be possible with some slight loss in emission control.

In July 1973, the Dresserator System was reported to be in a research prototype state of development (Ref. 8-7). Test fixtures of several versions of the Dresserator core have been made, utilizing both float bowl and compressive fuel feed systems (Ref. 8-10).

In July, two vehicles were delivered by Dresser to the EPA for testing: (1) a 1973 Chevrolet Monte Carlo equipped with a 350-CID engine and an automatic transmission, and (2) a 1973 Ford Capri with a 159-CID engine and an automatic transmission.

The modified Monte Carlo had an Edelbrock single plane manifold, a 1970 distributor with idle retard, no vacuum advance, no EGR, and no air pump. Other items are standard equipment.

The center divider of the Capri manifold was removed in the plenum area only. The stock cam was exchanged for one with slightly less overlap, reducing internal EGR. It presently has no EGR or vacuum advance. Enlarged and lagged (insulated) exhaust manifolds were installed. Other items are standard equipment.

Table 8-1. Initial Dresserator System 1972 Federal CVS Test Results

Vehicle	Emissions, gm/mi			Fuel Economy, mpg	Reference
	HC	CO	NOx		
1971 Ford Galaxie, 351 CID					
● Dresserator	0.3 - 0.5	4.5 - 7.5	1.2 - 1.7	10.5-11.0	8-7
● Dresserator	0.32	4.68	1.58	10.8	8-9
● Dresserator with conventional exhaust	0.8 - 1.0	6 - 7	1 - 1.3	11-13	8-7
● Baseline	1.5 - 2.5	30 - 40	4.0 - 4.2	10.4-10.6	8-7
1973 Chevrolet Monte Carlo, 350 CID					
● Dresserator	0.65-0.95	4.9 - 6.2	1.16-1.60	11.2-12.0	8-10
6. Baseline	1.71	24.0	2.42	11.6	8-10
● Dresserator with vacuum advance	1.07	5.84	2.00	13.0	8-10
1975 California Standards	0.9	9.0	2.0		

Both cars were equipped with the Model 2 Dresserator, which uses a low pressure fuel-feed system and a cold start compensation system with varying fuel pressure which acts for 120 to 190 sec after a cold start. The Dresserator inductors use neither a conventional choke nor accelerator pump.

Emission tests were performed as directed in the 1972 FTP and the 1975 FTP (Federal Register, Vol. 37, No. 221, Part II, November 15, 1972). In addition, the Federal Highway Driving Cycle (FHDC) was run on each vehicle to determine highway fuel economy.

Two 1975 FTP's and one FHDC were run on the Monte Carlo. Two 1975 FTP's and two FHDC's were run on the Capri with the ignition timing set at 8° BTDC. The timing on the Capri was then advanced 3° (to 11° BTDC) and a hot start 1972 FTP and FDHC were run. Advancing the ignition timing on the Capri was expected to increase fuel economy and raise the hydrocarbon emissions slightly.

Both vehicles achieved 1975 interim emissions levels (Tables 8-2 and 8-3). The Capri in fact was well within the California interim levels of 0.9-gm/mi HC, 9.0-gm/mi CO, and 2.0-gm/mi NO_x. With the timing set at +8° (before TDC), the Capri produced less than half the allowable (federal) HC, CO, and NO_x. Advancing the timing 3° resulted in a 10% increase in both LA-4 and highway fuel economy, while HC emissions increased by 50%. However, HC emissions were still less than 1.0-gm/mi. The Monte Carlo ran at less than two-thirds the permissible levels for HC, CO, and NO_x.

NO_x emissions from both vehicles were below the 1977 level of 2.0-gm/mi. Neither vehicle met the 1977 HC and CO levels of 0.41 and 3.4-gm/mi, respectively.

The values for the "Hot 1972 FTP" in Table 8-2, to study the effects of advancing the timing, were calculated from the second and third parts of the 1975 FTP.

Fuel economy for both vehicles was good. The 1972 FTP fuel economy and fuel consumption values listed in Tables 8-2 and 8-3

Table 8-2. Dresserator System Test Results: 1973 Capri, 159 CID[a]

Test	Timing	Mass Emissions, gm/mi (gm/km)				Urban		Highway	
		HC	CO	CO_2	NO_x	Fuel Economy, mpg	Fuel Consumption, liters/100 km	Fuel Economy, mpg	Fuel Consumption, liters/100 km
1975 FTP	+8°	0.57 (0.35)	5.29 (3.29)	471 (293)	1.16 (0.72)	19.2	12.3	24.2	9.7
1975 FTP	+8°	0.60 (0.37)	5.72 (3.55)	500 (311)	1.35 (0.84)	17.5	13.4	24.4	9.6
Average		0.59 (0.36)	5.50 (3.42)	486 (302)	1.25 (0.78)	18.4	12.9	24.3	9.7
1975 Interim Standards		1.5 (0.93)	15.0 (9.32)	- -	3.1 (1.93)				
1972 FTP[b]	+8°	0.77 (0.48)	8.13 (5.05)	481 (299)	1.39 (0.86)	17.9	13.1		
1972 FTP[b]	+8°	0.87 (0.54)	8.79 (5.46)	514 (319)	1.62 (1.01)	16.7	14.1		
Average		0.82 (0.51)	8.46 (5.26)	498 (309)	1.51 (0.94)	17.3	13.6		
Hot 1972 FTP[c]	+8°	0.40 (0.25)	3.40 (2.11)	486 (302)	1.15 (0.71)	18.0	13.1		
Hot 1972 FTP[c]	+8°	0.41 (0.25)	3.28 (2.04)	452 (281)	0.89 (0.62)	19.4	12.1		
Average		0.41 (0.25)	3.34 (2.08)	469 (293)	1.07 (0.67)	18.7	12.6		
Hot 1972 FTP[c]	+11°	0.62 (0.39)	3.43 (2.13)	426 (265)	1.27 (0.79)	20.6	11.4	27.0	8.7
1973 Certification Values (1973 Capri, 159 CID, Automatic)									
1972 FTP		2.7 (1.68)	38 (23.6)		2.8 (1.74)	15.4	15.3		

[a] Ref. 8-11.

[b] 1972 FTP values calculated from Bags 1 and 2 of 1975 FTP.

[c] Hot 1972 FTP values calculated from Bags 2 and 3 of 1975 FTP.

Table 8-3. Dresserator System Test Results: 1973 Monte Carlo, 350 CID[a]

Test	Mass Emissions, gm/mi (gm/km)				Urban		Highway	
	HC	CO	CO_2	NO_x	Fuel Economy, mpg	Fuel Consumption, liters/100 km	Fuel Economy, mpg	Fuel Consumption, liters/100 km
1975 FTP	1.12 (0.70)	5.35 (3.33)	720 (447)	1.77 (1.10)	12.4	19.0	20.3	12.6
1975 FTP	1.13 (0.70)	4.88 (3.03)	660 (410)	1.34 (0.83)	13.4	17.6		
Average	1.13 (0.70)	5.11 (3.18)	690 (429)	1.55 (0.97)	12.9	18.3		
1975 Interim Standards	1.5 (0.93)	15.0 (9.32)		3.1 (1.93)				
1972 FTP	1.26 (0.78)	6.07 (3.77)	738 (459)	2.05 (1.27)	11.8	19.9		
1972 FTP	1.18 (0.73)	5.52 (3.43)	674 (419)	1.47 (0.91)	12.9	18.2		
Average	1.22 (0.76)	5.80 (3.60)	706 (439)	1.76 (1.09)	12.4	19.1		
1973 Certification Values (1973 Chevrolet Monte Carlo, 350-2V, Automatic)								
1972 FTP	2.7 (1.68)	20.0 (12.4		2.4 (1.49)	12.0	19.6		

[a] Ref. 8-11.

(calculated from the data taken from the first and second parts of the 1975 FTP) are seen to be better than the values from the certification tests of the comparable 1973 models. No significant driveability problems were encountered.

Table 8-4 contains similar data for these two vehicles, based on Dresser tests made prior to shipping the cars to EPA in Ann Arbor, Michigan.

8.1.3 Ethyl Carburetor

The Ethyl Corporation has had a long term development program on emission control systems utilizing lean thermal reactors (Ref. 8-13). Its system incorporates lean carburetion, EGR, and thermal reactors. The effectiveness of the lean reactor system depends primarily on improved carburetion. A 3-venturi carburetor (Figure 8-11) was developed to provide a high degree of atomization and mixing, along with close-tolerance metering of the air-fuel mixture. This carburetor utilizes high air velocities for mixing. Other design characteristics also help. These include the geometry of the fuel nozzle and the use of perforations in the primary throttle plate through which the mixture passes under some conditions. Also, a mixing tube extends into the intake manifold beneath the primary throttle.

High air velocities are produced by the use of a small primary venturi for light loads and two variable secondary venturis for higher power conditions. Thus, the high velocities present under all conditions not only provide mixing but also give strong metering signals at any engine condition. These signals, in turn, promote metering accuracy. In addition, the strong venturi signal permits elimination of the separate idle system and allows fuel for idling and light load conditions to be provided through the main nozzle of the primary venturi, which also benefits mixing. Other refinements incorporated in the carburetor include a device for temperature-compensating the idle mixture ratio, an internal control to increase mixture flow during deceleration, and a temperature-modulated choke that closely relates both the degree and

Table 8-4. Dresser Industries Test Data[a]

<table>
<tr><th colspan="6">1973 Chevrolet Monte Carlo, 350 CID, 9:1 CR, 5° BTC Basic
4500-lb Inertia Weight, Single Bag 1973 CVS Tests</th></tr>
<tr><th></th><th>HC, gpm</th><th>CO, gpm</th><th>NO, gpm</th><th>Mileage</th><th>CVS</th></tr>
<tr><td>Baseline</td><td>1.60-2.20</td><td>23.0-43.8</td><td>2.0-2.59</td><td>11.2-11.9</td><td>Cold[b]</td></tr>
<tr><td>Baseline</td><td>1.31</td><td>11.9</td><td>2.76</td><td>13.1</td><td>Hot</td></tr>
<tr><td>Modified</td><td>0.65-0.95</td><td>4.9-6.2</td><td>1.16-1.60</td><td>11.2-12.0</td><td>Cold</td></tr>
<tr><td>Modified</td><td>0.67-0.80</td><td>4.8-5.2</td><td>0.96-1.20</td><td>12.2-12.7</td><td>Hot</td></tr>
<tr><th colspan="6">1973 Capri V-6, 2600 cc, 8.7:1 CR, 8° BTC Basic
3000-lb Inertia Weight, Single Bag 1973 CVS Tests</th></tr>
<tr><td>Baseline</td><td>1.57-1.80</td><td>29.3-32.8</td><td>2.13-2.48</td><td>14.4-15.6</td><td>Cold[c]</td></tr>
<tr><td>Baseline</td><td>1.23-1.28</td><td>17.8-19.7</td><td>2.0-2.34</td><td>16.8-17.5</td><td>Hot[c]</td></tr>
<tr><td>Modified</td><td>0.36-0.54</td><td>4.94-6.42</td><td>1.15-1.55</td><td>15.7-17.1</td><td>Cold</td></tr>
<tr><td>Modified plus 3° basic spark</td><td>0.94</td><td>6.70</td><td>1.45</td><td>18.7</td><td>Cold</td></tr>
<tr><td>Modified</td><td>0.09-0.17</td><td>2.1-3.2</td><td>0.9-1.3</td><td>16.5-17.5</td><td>Hot</td></tr>
<tr><td>Modified plus 3° basic spark</td><td>0.26-0.44</td><td>3.4-3.7</td><td>1.02-1.22</td><td>20.5-21.1</td><td>Hot</td></tr>
</table>

[a] Ref. 8-12.

[b] Very sensitive choke.

[c] 2750-lb inertia.

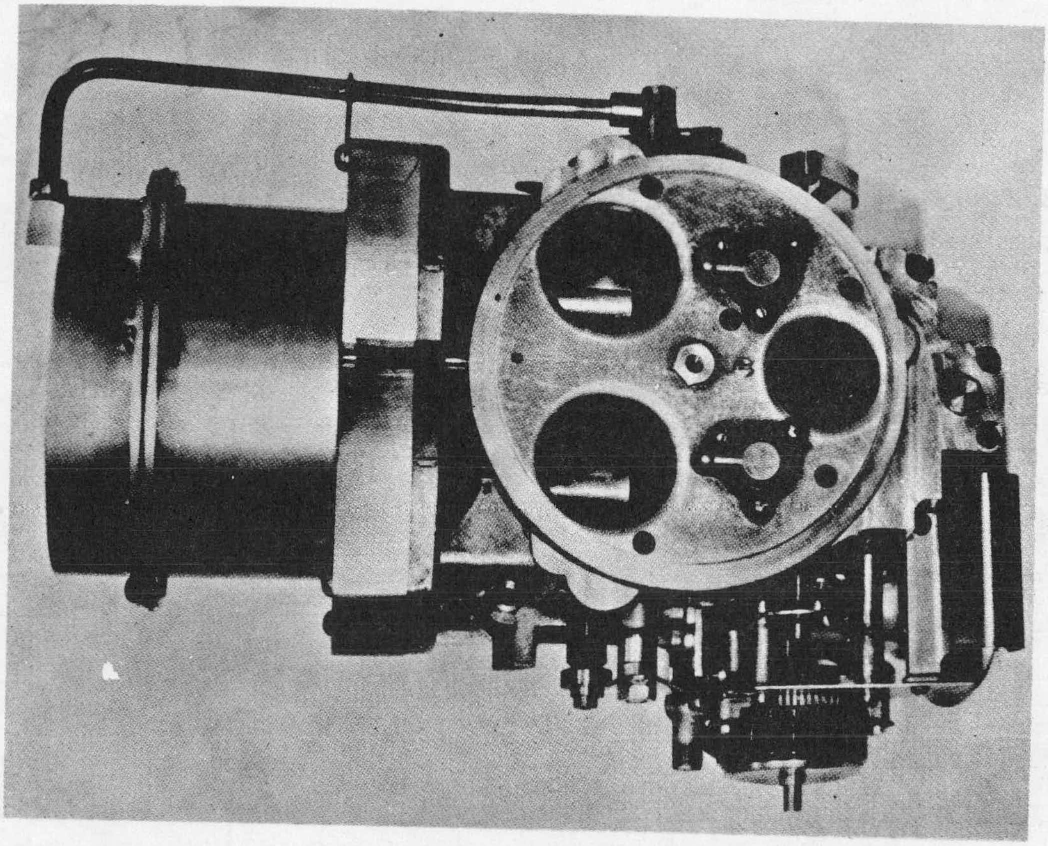


Figure 8-11. Experimental 3-venturi carburetor (Ref. 8-13)

duration of choking to engine and under-hood temperatures. Use of these systems, even in conjunction with EGR, permits an idling air-fuel ratio of about 17.2:1 and operating air-fuel ratios of 17-18:1 across the speed range. Enrichment to a 12:1 air-fuel ratio occurs at full power.

Several systems are used in the lean reactor to improve performance and emissions during the first few minutes after starting. Choking is carefully regulated, and the idle air-fuel mixture is temperature-compensated. In addition, the intake manifold is modified in the hot-spot section beneath the carburetor throats to transfer heat rapidly from the exhaust gas side of the manifold crossover to the intake side. This provides more rapid vaporization of fuel on the cold start. This modification consists of discs of finned stainless steel that replace portions of the normal cast iron structure in the hot-spot area. A crossover heat control valve directs exhaust gas from one side of the engine through the crossover and out the other side during the cold start period. After warmup, this valve opens, and the exhaust gas bypasses the hot-spot area. Carburetor air also is preheated rapidly by the use of a muff-type preheater installed at one reactor outlet. The conventional temperature control valve in the air cleaner opens after warmup to maintain carburetor air at the normal temperature.

A 1972 Fury III with a 360-CID engine and the Ethyl lean thermal reactor system was tested by EPA (Ref. 8-14). Three tests were conducted in accordance with the 1975 FTP as described in the November 15, 1972, Federal Register. All test work was conducted at 4500-lb inertia weight. The results from the tests are shown in Table 8-5. These results demonstrate that emission levels well below 1975 interim standards can be achieved with this system. For comparative purposes, an average 1972 FTP result was calculated using results from the first two bags of the reported 1975 test work. Comparison of this data with 1973 certification of emissions levels and fuel economy is shown. This vehicle demonstrated significantly better emissions and fuel economy than a similar 1973 certification vehicle. The general impression of vehicle driveability was good.

8.1.4 Ethyl Turbulent Flow Manifold

The latest development in the Ethyl work with lean systems is the turbulent flow manifold (TFM). In its early form, it was used with the Ethyl 3-barrel carburetor on two 1971 full-size cars equipped with thermal reactors. The engine in one car had a standard compression ratio (designed to operate on low-octane gasoline), while the engine in the other car had its compression ratio raised one unit for improved fuel economy.

The TFM proved to be so effective in improving mixture quality that good distribution resulted even when the TFM was used with standard carburetors. In fact, a very satisfactory lean-burn emission control system results when a standard carburetor, modified to provide lean mixtures, is combined with the TFM and the ignition timing is optimized for emissions

Table 8-5. Emissions and Fuel Economy of Ethyl Lean Reactor System[a]

1975 FTP					
	Emissions				Fuel Economy, mpg
	HC, gm/mi	CO, gm/mi	NO_x, gm/mi	CO, gm/mi	
Test 1	0.78	5.93	1.42	756.49	11.23
Test 2	0.78	5.51	1.30	769.51	11.05
Test 3	0.87	5.77	1.42	746.80	11.96
Average	0.81	5.74	1.38	757.60	11.41
1975 Interim Standards	1.5	15.0	3.1	-	-
1976 Interim Standards	0.4	3.4	0.40	-	-
1972 FTP					
Average of 3 tests	1.06	7.07	1.42	755.91	11.23
1973 Certification results	2.6	38.0	2.4	-	9.7

[a] Ref. 8-14.

and fuel economy. Ethyl calls this the turbulent flow system (TFS). Since this system does not include thermal reactors, it does not achieve the ultimate emission levels of the lean reactor system. Instead, it represents an interim, less expensive, lead-compatible system. To date, it has achieved levels meeting the 1975 49-state emission standards with good fuel economy.

The TFM used with a modified 4-barrel carburetor is shown in Figure 8-12. Only the primary barrels of the carburetor are shown, since the secondary barrels deliver their fuel-air mixture directly to the intake manifold runners. As Figure 8-12 shows, the major components consist of the entry mixing tube, conditioning chamber, and exit tubes. The conditioning chamber is exposed to exhaust gas heat on the outside, although some other designs use jacket water heat. The TFM (1) helps to increase mixing because of the longer J-shaped path and (2) helps to vaporize fuel droplets because of the heated surface of the conditioning chamber. These droplets, if sufficiently large, impinge on the walls rather than negotiate the J path of the vaporized mixture.

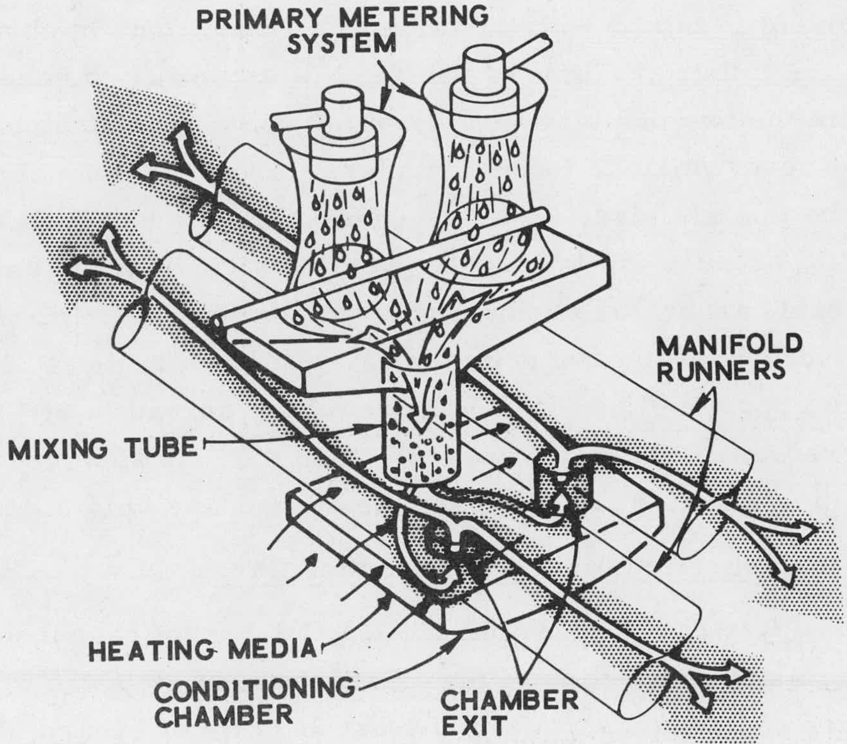


Figure 8-12. Turbulent flow manifold (Ref. 8-15)

At first, it was believed that a large part of the effectiveness of the TFM was due to the heated walls of the conditioning chamber. After studying the mechanism in more detail and reducing the heat available to the chamber walls, Ethyl concluded that the TFM effectiveness is due more to mixing than to temperature. Although several models of the TFM still use exhaust heat, other designs use only water heating without any apparent decrease in effectiveness.

It is also important to remember that, for the 4-barrel carburetor, only the mixture from the primary barrels passes through the conditioning chamber; the secondary barrels open directly into the manifold branches. A similar approach is used with 2-barrel staged carburetors.

A typical comparison of the spread in air-fuel ratio between the TFM and a standard carburetor-manifold system is shown in Figure 8-13. The improvement in mixture distribution at all conditions is obvious.

Another attribute of the TFM is its ability to accept unbalanced mixtures from the two primary barrels of a 4-barrel carburetor and still provide a homogeneous mixture to the cylinders. Even the normal air leakage at idle past the throttle blade in the secondary barrels has little effect on the mixture quality because the leakage is evenly distributed. In one experiment, using a 4-barrel carburetor mounted on a production intake manifold, the idle mixture screws of the two primary barrels were deliberately maladjusted by one turn in opposite directions. The resulting spread in air-fuel ratio among the 8 cylinders was 3.33 air-fuel ratio. When the test was repeated for the same carburetor mounted on the TFM, the spread was only 0.45.

8.1.4.1 Application of TFS to Full-Size Cars

After early tests with the Ethyl 3-barrel carburetor indicated the excellent mixing properties of the TFM, the next step was to experiment with production carburetors. After at least 2000 mi of breakin driving, three 1974 Make-A full-size cars were equipped with the TFS, which consisted of the TFM, a production carburetor modified to run lean, and an ignition

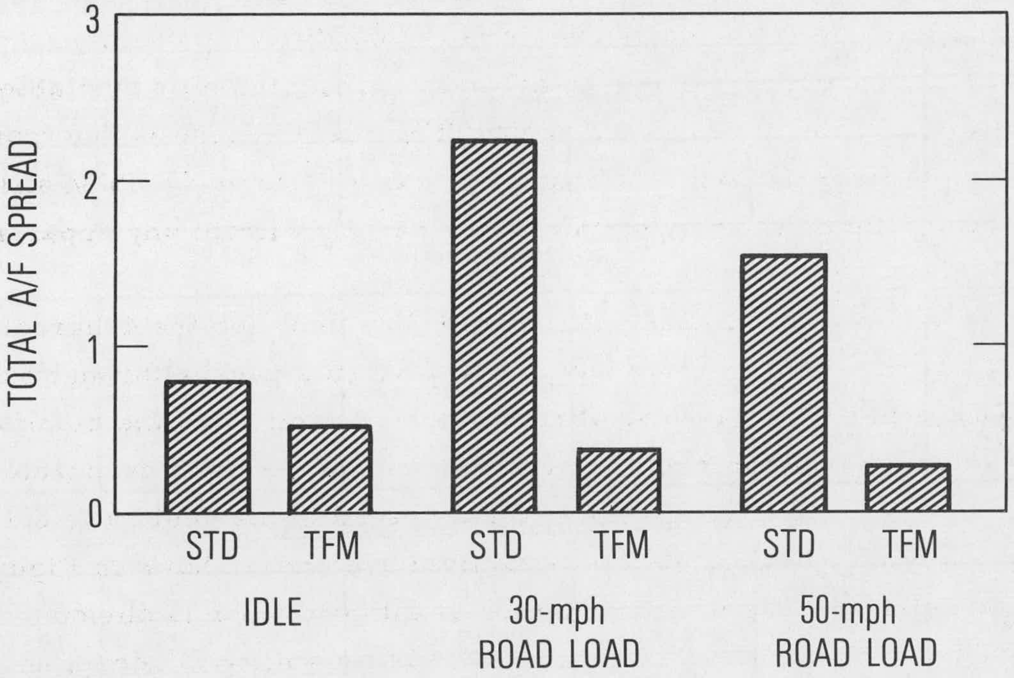


Figure 8-13. Total air-fuel ratio spread among cylinders: standard and turbulent flow manifolds, 1974 Make A, V-8 engine (Ref. 8-15)

timing adjustment. Emissions and fuel economy data for these cars, before and after the TFS conversion, are shown in Table 8-6. Fuel economy was measured using two driving cycles established by the EPA. These are the 1975 CVS method (the driving procedure used to measure emissions) and the newer 10-mi highway driving cycle (FHDC). The former is a cold-start city driving test, while the latter is a nonurban cycle with no stops and an average speed of nearly 50 mph.

The goal in these modifications was to meet the 1975 49-state federal emission standards using nothing but the TFS. This was achieved, along with improvements in fuel economy of 7.6% and 2.5% on the CVS and the highway dynamometer cycles, respectively. The fuel economy improvement results from the combination of better fuel-air distribution, elimination

Table 8-6. Emissions and Fuel Economy for TFS-Equipped Cars[a, b]

Vehicle	1975 CVS-CH Emissions, gm/mi			Dynamometer Cycle Fuel Economy, mpg[c]	
	HC	CO	NO_x	1975 CVS	High-way
As Received					
1	2.55	14.36	1.56	10.5	16.2
2	2.32	14.67	1.62	11.0	16.8
3	2.25	22.43	1.64	9.9	15.9
Average	2.37	17.15	1.61	10.5	16.3
With Turbulent Flow System					
1	1.42	7.55	2.26	12.2	17.2
2	1.23	7.42	2.39	11.2	17.0
3	1.24	7.31	2.03	10.4	15.9
Average	1.30	7.43	2.27	11.3	16.7

[a] Ref. 8-15.

[b] 1974 Make-A full-size cars, ~ 350-CID V-8 engine, single determinations.

[c] Chassis dynamometer cycles used by EPA to measure fuel economy of manufacturers' 1975 emission certification cars; fuel economy measured by gravimetric method rather than EPA's carbon balance method.

of the auxiliary air pump, and a small advance in ignition timing. Since the goal was to achieve the emission levels of the 1975-76 U.S. 49-state standards with good fuel economy, the combination of advanced ignition timing and a change in EGR rate resulted in the higher NO_x level, which is still well below the standard.

A fourth car was equipped with exhaust port liners in addition to TFS. Exhaust port liners are simple stainless-steel tubes placed in the exhaust ports to reduce heat loss to the coolant and underhood air. This combination resulted in emission levels below those of the 1975 California standards, along with improved fuel economy, as shown in Table 8-7. The major effect of the port liners was to reduce hydrocarbon emissions. Because of the hydrocarbon reduction, additional exhaust gas recycle and spark advance could be combined to achieve lower NO_x values along with good fuel economy. In fact, the fuel economy improvement on the CVS cycle of 7% over the nonmodified car is about the same as that shown for the cars using TFS only.

Table 8-7. Emissions and Fuel Economy of a Car Equipped with TFS and Exhaust Port Liners[a, b]

Vehicle	1975 CVS-CH Emissions, gm/mi			Dynamometer Cycle Fuel Economy, mpg	
	HC	CO	NO_x	1975 CVS	Highway
As received	1.84	19.06	1.55	9.7	16.3
With TFS plus exhaust port liners:					
Ethyl	0.52	6.79	1.85	10.4	16.2
Manufacturer A	0.52	6.15	1.73	10.6	15.8

[a] Ref. 8-15.

[b] 1974 Make-A full-size car, ~350-CID V-8 engine, single determinations.

8.1.4.2 Application of TFS to Smaller Cars

The previous discussion was limited to the use of the TFS on full-size cars with V-8 engines. Other designs of the TFM have been applied to smaller cars using 6-cylinder and 4-cylinder engines. The conditioning chamber was heated by exhaust gas in some cases and by water in others. A water-heated chamber is usually more convenient in cross-flow design engines, in which the intake and exhaust ports are on opposite sides of the cylinder head. Some of these designs involve staged carburetors (secondary barrels bypass the conditioning chamber), while others use single-barrel carburetors. Data for the cars tested to date are presented in Table 8-8. These cars do not have exhaust after-treatment such as port liners or thermal reactors. The full-size and compact cars utilize EGR, while the smaller cars do not.

Table 8-8. Typical Turbulent Flow System Results[a]

Vehicle	Weight, lb	Engine Type	1975 CVS-CH Emissions, gm/mi			Average Spread in A/F[b]	
			HC	CO	NO_x	TFS	Non-Modified
Full size[c]	5000	V-8	1.30	7.4	2.2	0.46	1.47
Compact	3500	6	1.30	5.5	2.1	0.38	1.04
Subcompact	2750	4	1.10	9.7	2.1	0.25	N.A.
European	2500	4	1.48	7.7	2.0	0.21	0.60
U.S. 49-state 1975-76 standards			1.5	15.0	3.1		

[a] Ref. 8-15.

[b] Maximum spread in A/F among the cylinders averaged for idle and several cruise speeds to 50 mph.

[c] 3-car average.

As Table 8-8 shows, all TFS conversions yielded emission levels below the 1975-76 U.S. 49-state standards. For those cars in which measurement of air-fuel ratio spread is available, the improvement in the overall spread is readily apparent. An improvement in driveability accompanied each conversion, and when compared with cars meeting 1974 U.S. emission standards fuel economy gains were realized.

8.1.5 NO_x versus Fuel Economy for Homogeneous Lean Mixtures

The relationship between NO_x emission level and vehicle fuel economy (as measured on the FTP) is illustrated in Figure 8-14 for the homogeneous lean mixture modifications described in Sections 8.1.2 (Dresserator), 8.1.3 (Ethyl 3-barrel carburetor plus lean thermal reactor), and 8.1.4 (Ethyl). For NO_x levels below 2 gm/mi (the 1975-76 California standard), significant fuel economy gains are evident for these modifications to the conventional spark-ignition engine. The results for the Dresserator system are the most dramatic: ~25% fuel economy improvement for the Capri, ~13% improvement for the Monte Carlo, and ~ 14% improvement for the 351-CID Ford. The TFS system fuel economy improvement was ~7%, while the Ethyl lean thermal reactor results (with 3-barrel carburetor) are ~15% higher than 1973 model year certification test values for the same model vehicle.

8.2 STRATIFIED CHARGE ENGINES

A comprehensive review of the current status of alternative automotive engines was recently performed by The Aerospace Corporation for the EPA/AAPS (Refs. 8-16). The following discussions of open-chamber and divided-chamber stratified charge engines are based on the material reported in Ref. 8-16.

8.2.1 Introduction

8.2.1.1 General Description

In principle, the stratified charge engine represents a modification of the conventional spark ignition engine. The main objective of the modification is to achieve heterogeneous combustion in which a rich fuel mixture is generated around the spark plug and a lean mixture is generated in remaining combustion chamber zones. The resulting two-stage combustion process permits operation of the engine at very lean overall fuel-air ratios which is conducive to low emissions, good fuel economy, and reduced sensitivity of the engine to fuel octane and cetane numbers.

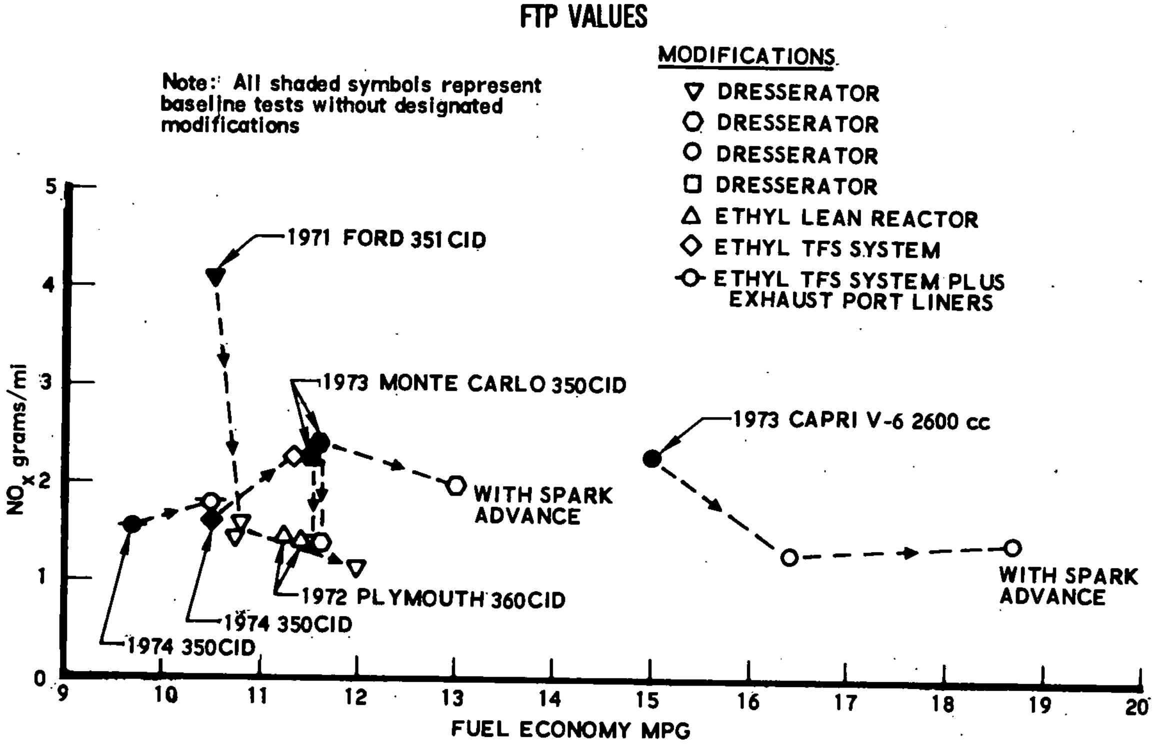


Figure 8-14. Homogeneous lean mixtures: NO_x vs fuel economy

Stratified charge engines can be divided into two distinct classes: open chamber and divided chamber. In open-chamber configurations exemplified by the Texaco TCCS and Ford PROCO engines, a single combustion chamber similar to that of conventional spark ignition engines is used. During engine operation, 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 combustion chamber's outer regions where the combustion process is completed in an oxygen-rich environment. Attempts are currently being made to replace the more expensive fuel injection systems employed in these engines with conventional carburetors.

The divided-chamber stratified charge engines or prechamber engines exemplified by Honda's CVCC engine concept employ two interconnected combustion chambers per cylinder. During the induction 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. Upon ignition in the prechamber, hot gases expand into the main chamber where combustion is then carried to completion. The principal advantage of prechamber engines over conventional engines is their ability to operate with very lean overall fuel-air 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. 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 fuel-air mixture regime.

8.2.1.2 Historical Developments

The concept of stratified charge operation of internal combustion engines and its potential was first evaluated in the early 1920s by Ricardo in England. In the U.S. work on the open-chamber configurations commenced in the 1940s with Texaco's invention of a "knockless" engine. Research and development of a number of alternate designs have since been pursued by several investigators and organizations throughout the world. In 1965, the U.S. Army Tank Automotive Command (TACOM) initiated funding of exploratory research of both open-chamber and divided-chamber stratified charge engine concepts which culminated in the development efforts conducted over a number of years by Ford and Texaco. The objective of the TACOM-sponsored Ford and Texaco programs was the development of a military engine with multifuel capability and improved fuel economy relative to conventional spark ignition engines. Minimization of exhaust emissions was added later as a design requirement under joint EPA and TACOM sponsorship.

The level of funding of the Texaco effort was recently increased prior to completion of the program. The TACOM support of the Ford development program was previously terminated. However, Ford is conducting a number of in-house programs aimed at the development of its stratified charge engine concepts for potential use in passenger cars.

The concept of a divided combustion chamber can be traced back to the first oil engines in the pre-Diesel era. These early prechamber engines operated on naphtha or heavy oils that were injected into a spherical prechamber and vaporized in contact with the red-hot thermally insulated prechamber walls. Later, attempts were made to apply the prechamber concept to the spark ignition gasoline engine, as documented by numerous patents. However, to date, a practical and widespread application of the prechamber concept has been realized only in high-speed Diesel engines, principally due to the pioneering efforts of Ricardo. Currently, a number of organizations and individuals are working on the development of prechamber engines. In particular, the Honda Motor Company of Japan has conducted extensive design, development, and test work on its CVCC concept during the past several years.

8.2.2 Open-Chamber Stratified Charge Engines

8.2.2.1 Powerplant Descriptions

8.2.2.1.1 Texaco Controlled Combustion System

The Texaco controlled combustion system (TCCS), illustrated in Figures 8-15 and 8-16, is a stratified charge process utilizing controlled air swirl and fuel injection combined with positive spark ignition (Refs. 8-17 through 8-19). Injection of the fuel into the swirling air commences near the end of the compression stroke, and the injected fuel is immediately ignited to establish a flame front adjacent to the injection nozzle. As injection continues, additional fuel is supplied to the flame front and burned very rapidly as soon as a combustible mixture has been formed.

3

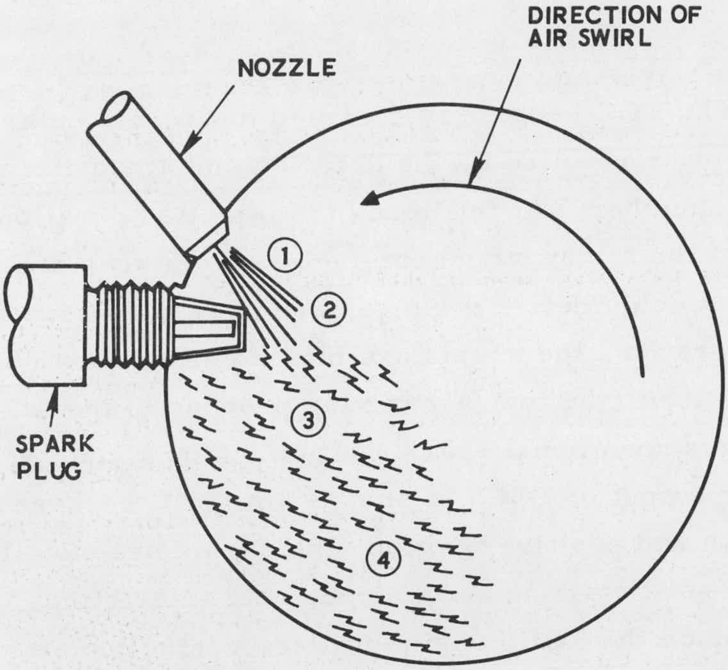


Figure 8-15. Texaco controlled combustion system

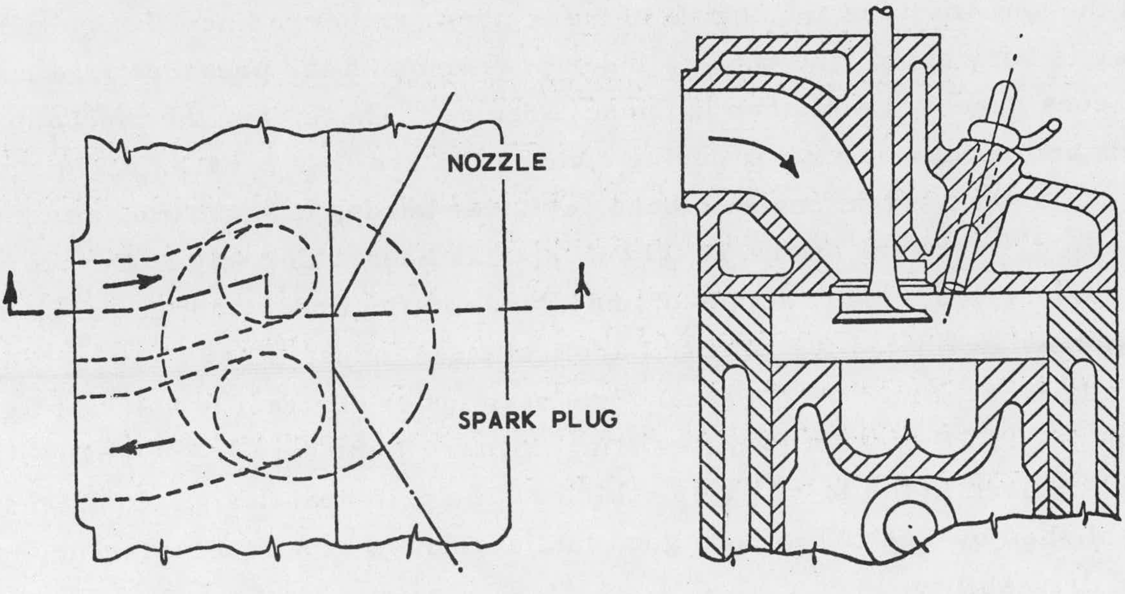


Figure 8-16. Texaco cup combustion chamber; stratified charge engine (Ref. 8-19)

The TCCS engine is designed to operate unthrottled with the power output being controlled by the duration and amount of fuel injected into the combustion chamber. At full load, the injection duration covers about one revolution of the air swirl. Lean fuel-air mixtures are utilized at part load, and near-stoichiometric mixtures are used at full load. Because of the lean mixture operation, the engine exhibits excellent part-load efficiencies. However, at full load, the fuel-air mixing process in the TCCS engine is not as efficient as in conventional spark ignition engines; as a result, the maximum power output of TCCS is smoke limited. Because of the direct cylinder injection and positive ignition, the engine has excellent throttle response and very good starting and warmup characteristics.

Since the TCCS engine utilizes controlled air swirl, fuel injection, and positive ignition, it is not penalized by some of the inherent disadvantages of conventional spark ignition and Diesel engines. Ignition of the first increment of fuel at the time of injection eliminates the possibility of mixture preignition and engine knock frequently encountered in gasoline engines, regardless of the octane number of the fuel used. Relative to Diesels, the positive ignition utilized in the TCCS engine minimizes the ignition delay and the sensitivity of the engine to fuel cetane number. Since the ignition delay is very short, the peak cylinder pressure and the pressure rise rates are considerably lower than in Diesel engines. Therefore, the mechanical loads and shocks occurring during combustion are less severe, permitting the use of lightweight engine blocks, cylinder heads, crankshafts, and pistons.

Work on the TCCS concept has been under way at Texaco since the early 1940s. Many single-cylinder and multicylinder configurations have since been evaluated for potential application in industrial and automotive installations. Since 1965, Texaco has been under contract to TACOM to adapt the TCCS concept to the existing military light-duty L-141 engine, which is used in the M-151 jeep vehicle. The principal design objectives established by TACOM include good fuel economy, broad fuel tolerance, and good driveability.

An emission-control system, consisting of EGR, oxidation catalysts, and intake air throttling, was added later in the program at the expense of some loss in fuel economy. With the exception of these components, the design and construction of the TCCS engine is comparable to other reciprocating engines. The TCCS engine has been operated up to the maximum governed speed of 3800 rpm. Texaco feels that speeds up to 5000 rpm are feasible with TCCS by modifying the intake system and by utilizing a more favorable bore-to-stroke ratio.

Following the successful application of engine intake pressure charging in exploratory tests, Texaco has incorporated a commercially available turbocharger into some of its L-141 TCCS engines. Other changes to the engine are minor, with the most significant being a reduction of the compression ratio from 10:1 to 9.3:1, to reduce the structural loading of the lightweight L-141 engine and to minimize engine knock at full power (Ref. 8-19). On the dynamometer, the engine has demonstrated good operating characteristics and is free from misfire and knock when operated on combat gasoline, CITE fuel, and No. 2 Diesel fuel.

8.2.2.1.2 Ford Programmed Combustion Concept

The PROCO engine concept, as applied to the military L-141 light-duty engine, is illustrated in Figure 8-17. In this engine, air enters through a special intake system which is designed to impart a moderate swirl around the cylinder bore axis in order to assist in attaining proper fuel-air mixing and good combustion. The air charge inducted into the cylinder is then compressed and transferred into the cup-shaped combustion chamber located concentrically in the piston. The top opening of the chamber has been optimized and represents 35% of the bore area, resulting in 65% squish action (Ref. 8-20).

In addition, Ford has incorporated the PROCO concept into several of its 351-CID V-8 engines. With the exception of a symmetrical piston cup and a dished cylinder head surface, the PROCO components used in this engine are equivalent to those employed in the L-141 engine (Ref. 8-20).

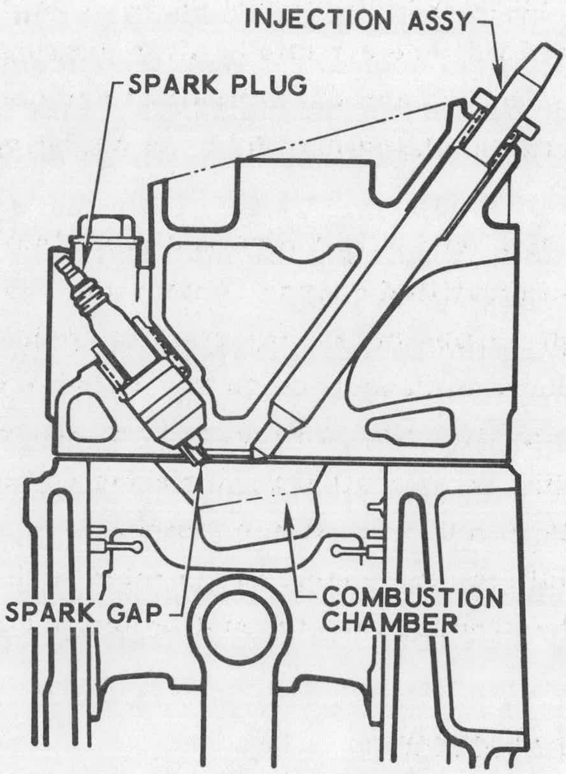


Figure 8-17. Ford L-141 PROCO engine (Ref. 8-20)

A compression ratio of 11.0 has been selected by Ford for all PROCO engines. Fuel is injected into each cylinder through an inclined nozzle configured to provide a low, penetrating, wide-angle, conical fuel spray. The spray mixes with the air and forms a rich mixture in its center surrounded by increasingly leaner mixtures and a layer of air adjacent to the cylinder wall to minimize quenching effects. A specially designed spark plug located just above the spray cone is used to ignite the mixture near the TDC position of the piston. From there, combustion proceeds very rapidly into the rich mixture cone and then into the leaner regions. As the piston descends, the swirling charge expands from the piston cup into the cylinder where the combustion process is then carried to completion.

The NO_x emissions of the engine are further reduced by means of EGR, which is inducted into the air intake after passing through a water-cooled heat exchanger. The HC and CO emissions are controlled by a system consisting of a low-inertia insulated manifold, an oxidation catalyst, and air throttling.

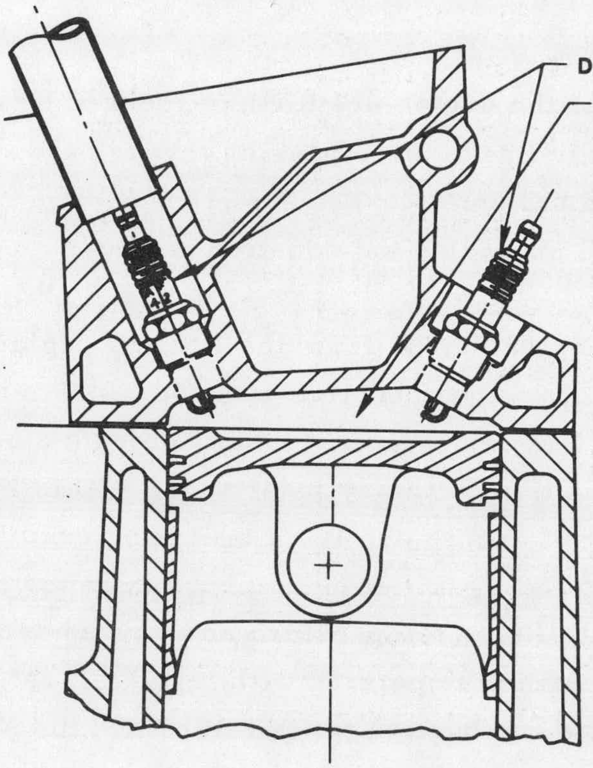
The HC, CO, and NO_x emissions from the PROCO engine are inherently low due to the stratified charge combustion process. HC is minimized internally through the use of late injection in conjunction with a soft fuel spray used to prevent fuel deposition on the cylinder wall. CO is inherently low because of the availability of excess air at all operating conditions except at full power. NO_x is controlled by initiating combustion in the fuel-rich zone, which results in moderate flame temperature and oxygen-free, postflame gases. In addition, the engine is operated with high EGR rates and late combustion to further reduce the combustion temperature and the NO_x formation rates.

8.2.2.1.3 Fast Burn Concept

The Ford Fast Burn engine concept is a modification of the PROCO engine in which the fuel injection system has been replaced by a carburetor. To date, two configurations, Fast Burn Phase I and Fast Burn Phase II, have been tested by Ford (Ref. 8-21).

Fast Burn Phase I consists of a bowl-in-piston combustion chamber similar to the PROCO design. The engine has a compression ratio of 11.0 and utilizes high air swirl rates in conjunction with improved carburetion, EGR, and modified spark advance. This concept was tested in a modified 351-CID engine installed in a 1972 Torino automobile.

The design features of the Fast Burn Phase II concept are illustrated in Figure 8-18. The engine incorporates a conventional combustion chamber and two spark plugs per cylinder and utilizes EGR, low-intake air swirl, and a normal production-compression ratio. Initial concept evaluation tests have been conducted on a 400-CID V-8 engine installed in a 1973 Torino vehicle.



- DESCRIPTION
  - MODIFICATION OF PROCO ENGINE
  - FUEL INJECTION REPLACED BY CARBURETION
  - PHASE I TYPE
    - PROCO-TYPE COMBUSTION CHAMBER
    - HIGH AIR SWIRL
    - HIGH CR (11.0)
    - EGR
  - PHASE II TYPE
    - CONVENTIONAL COMBUSTION CHAMBER
    - TWO SPARK PLUGS
    - LOW INTAKE AIR SWIRL
    - EGR
    - NORMAL CR
- DEVELOPMENT
  - INITIAL VEHICLE TESTS CONDUCTED

Figure 8-18. Ford Fast Burn Phase II stratified charge engine (Ref. 8-21)

8.2.2.2 Power and Efficiency

The power output and torque characteristics of the Texaco naturally aspirated L-141 TCCS engine at full throttle are presented in Figure 8-19 as a function of engine speed (Ref. 8-19); also shown are the performance curves of the conventional L-141 engine. In the midspeed regime, the TCCS engine develops somewhat higher power than the standard engine. Conversely, in the low-speed regime and near maximum speed, the TCCS engine has slightly lower power. The loss in engine power at speeds above 3200 rpm is due primarily to injection limitations imposed by engine smoke. Similar smoke-limited power characteristics were obtained by Texaco for other fuels.

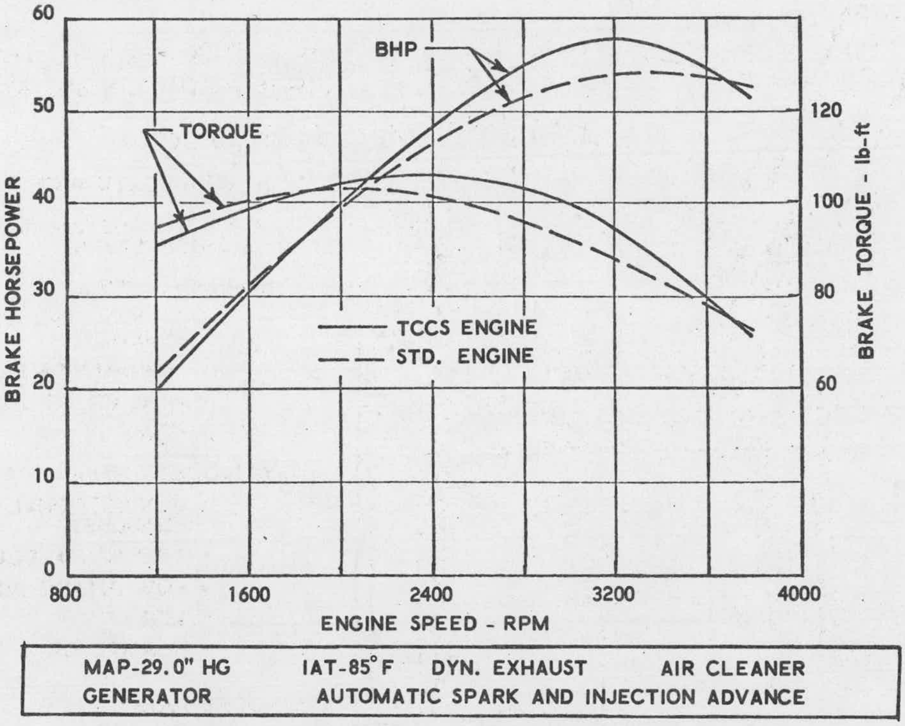
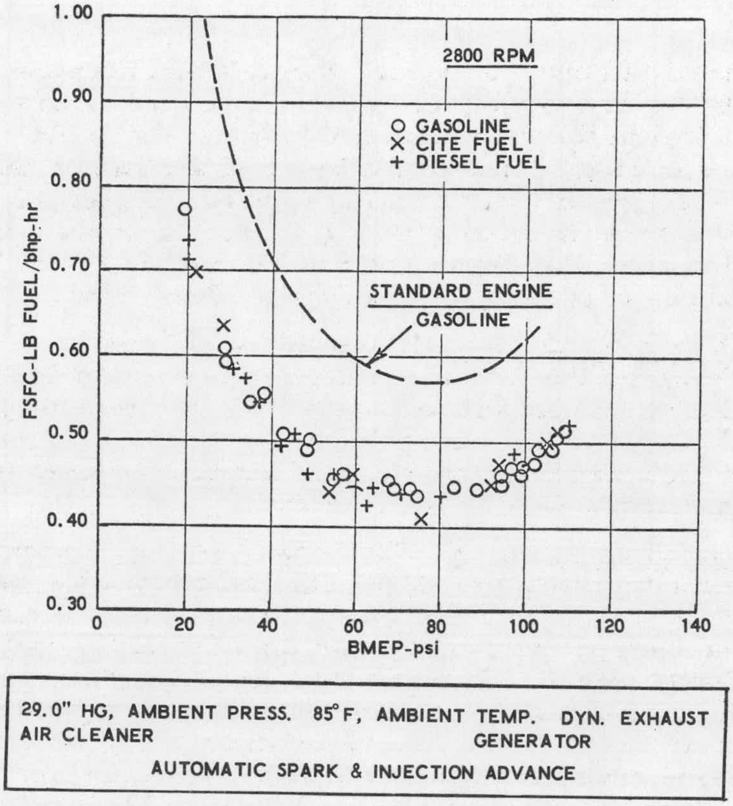


Figure 8-19. Full-load brake performance, Texaco naturally aspirated L-141 TCCS engine (Ref. 8-19)

The superior efficiency capability of the TCCS engine concept with respect to the standard L-141 spark ignition engine is illustrated in Figure 8-20, showing the BSFC of the two engines at 2800 rpm as a function of brake mean effective pressure (BMEP). As indicated, the BSFC of the TCCS engine is essentially independent of the type of fuel used. For a BMEP of 30 psi, the TCCS engine has a 27% lower BSFC; at 100 psi, the difference is ~ 24% in favor of TCCS. Similar trends were obtained by Texaco for other engine speeds (Ref. 8-19). However, when making such comparisons, consideration must be given to the fact that the efficiency of the standard L-141 engine is lower than that of the average Otto cycle engine.

The fuel consumption characteristics of the turbocharged TCCS engine and of Ford's PROCO engine are comparable to the naturally aspirated TCCS engine.



29.0" HG, AMBIENT PRESS. 85°F, AMBIENT TEMP. DYN. EXHAUST
AIR CLEANER GENERATOR
AUTOMATIC SPARK & INJECTION ADVANCE

Figure 8-20. Multifuel brake performance, Texaco naturally aspirated L-141 TCCS engine (Ref. 8-19)

8.2.2.3 Emissions

8.2.2.3.1 General

A stratified charge engine inherently emits less pollutants (HC, CO, NO_x) into its exhaust manifold than a carbureted engine with a comparable power output. The reasons for the lower emissions are as follows:

a. Hydrocarbons. Unburned HC emissions result primarily from the incomplete burning of fuel in contact with the combustion chamber walls. HC emissions also result from capture of fuel in various crevices and from insufficient oxygen in the body of the fuel-air charge. In the stratified charge engine, the wall and crevice effects are largely eliminated by concentration of fuel in the rich combustion zone; the walls are in effect lined with a layer of air rather than a mixture of fuel and air. Unburned hydrocarbons exist in relatively high concentrations in the rich zone, but these are readily burned by the excess oxygen in the lean stage of the burn.

b. Carbon Monoxide. Emissions of CO result whenever the fuel-air charge is uniformly deficient in oxygen. As in the case of HC, the rich stage of the stratified charge engine process does result in high initial CO formation, but this is effectively oxidized to CO_2 at the lean end.

c. Oxides of Nitrogen. Three factors play an important and integrated role in the formation of NO_x: peak cycle temperature, exposure time at high temperature, and oxygen availability. In the case of the stratified charge engine, the initially rich combustion limits the peak temperature and assures an oxygen-free environment behind the flame front. By the time the flame reaches the lean zone, the temperature is reduced so that the residence time with both high temperature and oxygen present is greatly minimized. As a result, the NO_x formed during combustion is rather low. These effects are depicted in Figure 8-21. In the early stages of combustion, the three pollutants are formed in the concentrations shown in the left-hand area of the figure. By the end of the combustion cycle, the concentrations correspond to those in the right-hand area of the figure, with the transition analogous to "tunneling" across the figure. Particularly in the case of NO_x, the conditions corresponding to near-stoichiometric combustion are not permitted to exist long enough for NO_x to build up to equilibrium levels.

NO_x control to the 1976 level has been achieved by both PROCO and TCCS engines without reduction catalyst aftertreatment. EGR alone has proven sufficiently effective to meet this goal. Further, the oxygen-rich characteristics of the exhaust make the stratified charge engine much more tolerant of high EGR rates than carbureted engines. Recirculation rates of 25% are standard practice for stratified charge engines, and rates as high as

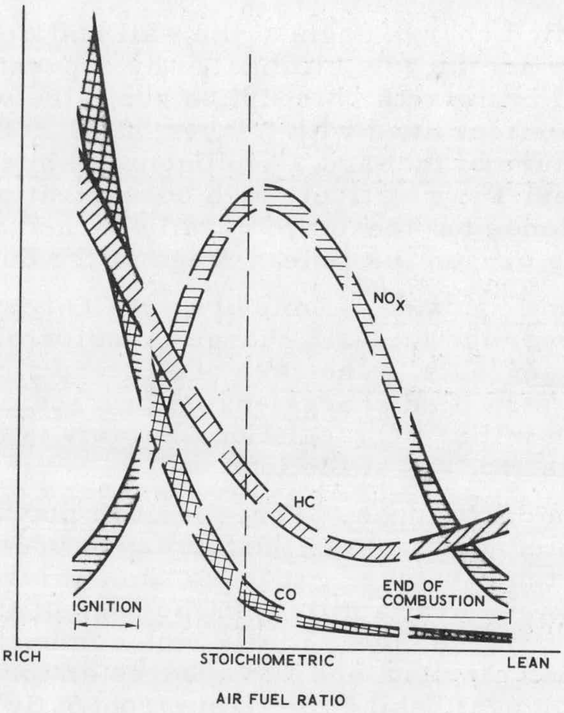


Figure 8-21. Effect of stratified charge combustion on emissions

50% (at idle) have been used without causing misfire. The stratified charge engine has sufficient NO$_x$ control margin to permit EGR cutoff at high loads without exceeding the 1976 level, thereby greatly reducing driveability degradataion under peak load conditions.

8.2.2.3.2 Texaco TCCS Engine

Initial vehicle emission tests on the Texaco TCCS engine were conducted by the National Air Pollution Control Administration (NAPCA) in 1968. In 1970, NAPCA tested an M-151 vehicle, which was equipped with a turbocharged L-141 TCCS engine and was driven on the dynamometer over the then proposed FDC (LA 4-S3). The emissions from this engine, measured in accordance with the CVS technique (3000-lb inertia weight), are listed along with the emissions of the standard M-151 vehicle in Table 8-9. Both vehicles

Table 8-9. Exhaust Emissions of a Turbocharged M-151 TCCS Vehicle[a]

Emissions[b]	M-151 TCCS, gm/mi		Standard M-151, gm/mi
	No Catalyst	With Catalyst	
HC	3.85 - 4.58	1.74 - 1.97	6.4
CO	9.08 - 9.62	2.29 - 2.69	76.2
NO_x	1.74 - 1.92	1.81 - 2.23	3.4

[a] Ref. 8-19.

[b] LA 4-3S Test Cycle, 3000-lb inertia weight, cold start, gasoline fuel.

were tuned for maximum performance with no consideration given to emissions. Although the uncontrolled TCCS vehicle had substantially lower emissions than the standard vehicle, the levels are significantly above the 1975-76 federal emission standards. As shown in Table 8-9, some reduction in the HC and CO emissions was achieved by means of a commercially available oxidation catalyst, which was installed downstream of the turbocharger.

In an attempt to meet the 1976 standards, Texaco added an emission control system to its basic TCCS engine design. This system consists of the following components:

a. A cooled dual-rate EGR system for NO_x control, incorporating a particulate trap, an EGR cooler, and an EGR mixer. At high loads, the EGR flow rate is about 15% of the air charge, and at low loads the rate is somewhat higher.

b. A closely coupled platinum catalyst incorporating a turbulence device.

c. A second platinum catalyst and a copper chromite catalyst installed downstream of the closely coupled catalyst.

d. Intake air throttling at idle and low loads in order to increase the exhaust temperature and improve the catalyst conversion efficiency at the expense of some loss in fuel economy.

The M-151 vehicle incorporating this baseline emission control system was tested by the EPA and by Texaco in 1972; the test results are presented in Table 8-10. As indicated, vehicle emissions are below the original 1976 statutory federal emission standards. However, it should be noted that the vehicle was not able to either negotiate all the acceleration modes in the cycle or achieve the maximum cycle speed of 57 mph.

Upon completion of a 50,000-mi durability run, the vehicle was retested by the EPA, and the emissions were lower than the 1976 statutory emission standards. However, major maintenance work was required on the engine during the durability test, including several ignition system modifications, replacement of four catalysts, cleaning of the EGR system, and major valve train maintenance (Ref. 8-23).

Table 8-10. Exhaust Emissions, Controlled Naturally Aspirated M-151 TCCS Vehicle Equipped with Exhaust Gas Recirculation System and Oxidation Catalysts[a]

Test Lab[b]	Mass Emissions, gm/mi		
	HC	CO	NO_x
Texaco	0.25	1.17	0.33
Texaco	0.28	0.62	0.32
Texaco	0.40	0.26	0.29
EPA	0.40	0.26	0.30
EPA	0.33	0.15	0.31
EPA	0.37	0.30	0.31

[a] Refs. 8-19 and 8-22.

[b] 1975 CVS C/H, 2750-lb inertia, lead-free gasoline.

The effect of emission control system modifications on the emissions and fuel economy of the baseline TCCS system incorporated into an M-151 vehicle is illustrated in Table 8-11. The modifications, which were implemented in 1972, include fuel injection and ignition retardation, various levels of modulated EGR, and utilization of oxidation catalysts in the engine exhaust. As indicated in the table, the CO and NO_x emissions of both the uncontrolled naturally aspirated and the turbocharged engines are low compared with uncontrolled conventional gasoline engines. Conversely, with respect to HC, the TCCS engines are comparable to conventional spark ignition engines. Some reduction in HC and CO can be achieved by retarded spark timing at the expense of lower fuel economy. As in conventional engines, EGR is quite effective in reducing NO_x, and the use of oxidation catalysts results in a substantial reduction of HC and CO. Similar results were obtained with JP-4 and Diesel fuel (Ref. 8-19).

Emission data from a TCCS-powered Plymouth Cricket vehicle tested in 1973 by Texaco, in accordance with the 1975 FTP, are listed in Table 8-12. The emission control system incorporated in the engine consisted of modulated cooled EGR, two oxidation catalysts, and air throttling at low loads. As indicated, the average emissions obtained in these tests are below the statutory 1976 standards (Tests 30, 32, and 35). While CO is attractively low, Chrysler feels that HC and NO_x are not low enough to provide an acceptable allowance for system deterioration due to mileage accumulation and production tolerances (Ref. 8-25). According to Chrysler, additional development is necessary to simplify the system design and to improve the EGR control.

Emission and fuel economy data obtained on this vehicle by Texaco are also listed in Table 8-12. The data demonstrate that injection and ignition retardation are quite effective in reducing NO_x emissions at the expense of some loss in fuel economy. When adjusted for best economy, the fuel consumption of the TCCS-powered vehicle is about 20% lower than the average 1973 Cricket certification vehicle.

Table 8-11. Effect of Emission Control System Modifications on M-151 TCCS Vehicle Emissions and Fuel Economy, 1975 FTP

Test Lab	Case No.	Engine Configuration	Turbo-charged	Emissions, gm/mi[a]			Fuel Economy, mpg by wt.	Max. Rear Wheel hp at 50 mph	References
				HC	CO	NO_x			
Texaco	1	Max. economy setting, no catalyst	Yes	3.13	7.00	1.46	24.3	40.0	8-24
Texaco	2	Max. economy setting, no catalyst	No	4.24	7.28	1.43	-	-	8-24
Texaco	3	8-deg combustion retard only, no catalyst	Yes	3.24	6.43	1.29	22.4	39.3	8-24
Texaco	4	8-deg combustion retard, low EGR rate, no catalyst	Yes	3.60	6.69	0.84	20.5	38.0	8-24
Texaco	5	8-deg combustion retard, medium EGR rate, two platinum catalysts	Yes	0.33	1.05	0.61	19.7	36.0	8-24
Texaco	6	13-deg combustion retard, high EGR rate, two platinum catalysts	Yes	0.35	1.41	0.35	16.2	28.7	8-24
Texaco	7	5-deg combustion retard, no EGR, two platinum catalysts, alternate vehicle	Yes	0.30	1.07	1.40	20.9	40.0	8-24
Texaco	8	Baseline configuration[b]	No	0.36	0.61	0.31	16.2	-	8-19
Texaco	9	Baseline configuration with reduced EGR	No	0.48	0.57	0.45	17.6	-	8-19
EPA	10	Baseline configuration[b]	No	0.37	0.24	0.31	15.8	-	8-19
EPA	11	Baseline configuration with reduced EGR	No	0.50	0.14	0.70	21.9	-	8-19

[a] Fuel - Gasoline 91 RON, lead-free, +2% oil.

[b] EGR plus two platinum catalysts.

Table 8-12. Exhaust Emissions from a Chrysler Cricket TCCS Vehicle, 2,500-pound Inertia Weight[a]

Test No.	Timing - Injection and Ignition	Catalysts	EGR	EPO[b]	Air Throttled	1975 EPA gm/mi			Fuel Economy, mpg
						HC	CO	NO_x	
Avg: 30, 32, and 35	Retarded for low emissions	Yes	Yes	Yes	Yes	0.36	1.15	0.38	20.1
44	Retarded for low emissions	Yes	Yes	No	No	0.59	0.64	0.54	22.6
47	Retarded for low emissions	Yes	No	Yes	No	0.61	0.85	0.99	22.5
50	Retarded for low emissions	Yes	No	No	No	0.73	1.18	1.20	23.8
53	For best economy	Yes	No	No	No	1.07	0.84	1.89	25.3

[a] Ref. 8-25.

[b] Exhaust backpressure orifice (EPO).

8.2.2.3.3 Ford PROCO and Fast Burn Engines

Results from M-151 PROCO vehicle tests conducted in 1971 by Ford (Ref. 8-20) and by the EPA (Ref. 8-26) are listed in Table 8-13. In these tests, the EGR cutoff and fuel enrichment at high power levels were deactivated to minimize NO_x. As a result, the vehicle had insufficient power to negotiate the two highest acceleration modes of the FDC. Ford estimates that a 20% to 25% larger engine can develop sufficient power to follow the cycle with EGR.

Table 8-13. Emissions and Fuel Economy, M-151 PROCO Vehicles[a]

Vehicle	Emissions, gm/mi[b] HC	CO	NO_x	Fuel Economy, mpg[c, d]	Number of Tests (averaged)	Test Facility
1975 CVS Test Procedure						
PROCO, no catalyst	2.60	13.45	0.32	21.7	1	Ford
PROCO, with catalyst	0.35	1.01	0.35	21.3[e]	4	Ford
PROCO, with catalyst	0.37	0.93	0.33	Not measured	14	EPA
1972 CVS Test Procedure						
PROCO, best fuel economy, no catalyst	4.96	7.75	3.85	23.8	2	Ford
PROCO, no catalyst	3.10	13.75	0.33	21.2	1	Ford
PROCO, with catalyst	0.54	1.18	0.37	19.6	4	Ford

[a] Refs. 8-20 and 8-26.

[b] Baseline (carbureted) vehicle emissions are 4.55 gm/mi HC, 41.6 gm/mi CO, and 4.4 gm/mi NO_x.

[c] Computed from the mass emission data (mpg).

[d] Baseline (carbureted) vehicle fuel economy 17.2 mi/gal.

[e] 20.4 mpg, measured with a burette.

As indicated in Table 8-13, the low-mileage emissions achieved with the derated M-151 PROCO vehicle with catalyst are below the 1976 statutory federal emission standards. Ford emphasizes, however, that this was accomplished with carefully controlled engine adjustments derived from a long series of optimization tests (Ref. 8-20). As a result of the high EGR rates employed, the engine operated close to its misfire limit. Without catalyst but with EGR, the HC emissions of the PROCO engine are about 40% lower than for the carbureted configuration, and CO is about 70% lower. At the best economy setting, PROCO gives about 10% lower HC and NO_x and about 85% less CO.

In 1973, Ford conducted Automobile Manufacturers' Association (AMA) durability tests on a 1972 Montego automobile equipped with a 351-CID PROCO engine and two Englehard PTX oxidation catalysts. As shown in Table 8-14, the low-mileage emissions are below the statutory 1976 standards. However, after accumulation of 26,000 mi, the HC and CO emissions had increased significantly, indicating excessive catalyst performance degradation. Subsequent installation of a 240° F thermostat, combined with higher backpressure operation of the engine at part load, resulted in a substantial reduction in HC (Ref. 8-21).

The effect of EGR on the emissions and fuel economy of this vehicle, tested without catalysts, is illustrated in Table 8-15. A reduction of the EGR flow rate from 25% to 15% resulted in substantially lower HC emissions at the expense of higher NO_x. Apparently, the variations in EGR performed by Ford had little effect on fuel economy.

Table 8-16 presents emission test data obtained by Ford from a 351-CID V-8 Fast Burn Phase I engine installed in a 1972 Torino vehicle (Ref. 8-21). With an oxidation catalyst, the vehicle meets the 1976 statutory emission standards for CO and NO_x at low mileage and exceeds the HC standard slightly. However, after 25,000 mi, the HC and CO emissions are substantially higher than the 1976 standards. The rapid deterioration of the catalyst is attributed to the high HC and CO levels of the raw engine exhaust, resulting in high thermal loading of the catalyst.

Table 8-14. 351-CID PROCO Durability: 1972 Montego, 110T722, 1975 Test Procedure[a]

Test	Catalyst Mileage	Emissions, gm/mi			Fuel Economy, mpg[b]
		HC	CO	NO_x	
At the start	10	0.16	0.25	0.32	13.9
At 26,000 mi, after repair and readjustment	26,195	0.79	1.16	0.37	12.75[c]
At 26,000 mi, after application of part-load backpressure system and 240°F thermostat	26,400	0.47	1.11	0.35	13.2

[a] Ref. 8-21.

[b] Based on carbon balance.

[c] Tank mileage for 26,000 mi: 12.7 mpg.

Table 8-15. 351-CID PROCO Emissions and Fuel Economy: 1972 Montego, 110T722, 1975 Test Procedure[a]

NO_x Design Level, gm/mi	EGR, %	Emissions, gm/mi			Fuel Economy, mpg[b]	Remarks
		HC	CO	NO_x		
0.4	25	3.54	28.2	0.34	13.8	No catalyst
1.5	15	1.34	25.0	0.96	13.2	No catalyst

[a] Ref. 8-21.

[b] Based on carbon balance.

Table 8-16. 351-CID Fast Burn Phase I Engine Vehicle Emissions and Fuel Economy: 1972 Torino, 1975 Test Procedure, Average of Two Tests[a]

Mileage	Emissions, gm/mi			Fuel Economy, mpg	Remarks
	HC	CO	NO_x		
Low	6.9	78.3	0.37	11.9	No catalyst
Low	0.44	1.31	0.34	11.9	HC-CO catalyst
25,154	2.52	5.78	0.37	11.4	HC-CO catalyst

[a] Ref. 8-21.

Fast Burn Phase II vehicle emission data are presented in Table 8-17 for operation without catalyst and secondary air, but with EGR. Under these conditions NO_x is rather low but HC and CO are excessive. The data show again the previously noted interrelationship between NO_x, HC, CO, and fuel economy.

Table 8-17. 400-CID Fast Burn Phase II Engine: Vehicle Emissions and Fuel Economy, 1973 Torino, 1975 Test Procedure[a]

NO_x Design Level, gm/mi	Emissions, gm/mi			Fuel Economy, mpg	Remarks
	HC	CO	NO_x		
0.4	3.1	30	0.66	8.6	No catalyst or secondary air
1.5	2.0	7.1	1.5	11.2	No catalyst or secondary air

[a] Ref. 8-21.

Ford has conducted a considerable amount of single-cylinder PROCO engine test work to determine the effects of various operating parameters on emissions and fuel consumption (Ref. 8-20). The parameters investigated include injection and ignition timing, overall air-fuel ratio, and EGR. As expected, retarded ignition and/or injection timing and EGR reduce NO_x at the expense of higher HC, CO, and fuel consumption.

8.2.2.4 Fuel Economy

8.2.2.4.1 Texaco TCCS Engine

As discussed previously, an emission controlled M-151 TCCS vehicle was tested by the EPA in accordance with the 1975 FTP and after the vehicle had been subjected to a 50,000+mi durability run (Ref. 8-23). The emission control system utilized on this vehicle consisted of EGR, three catalysts, and intake throttling. It also required considerable maintenance work during the test period. The fuel economy, determined by the EPA by means of the carbon balance method from six consecutive tests, varied between 14.4 and 17.1 mpg, with the average at 15.5 mpg.

Somewhat better fuel economy data have been reported by Texaco for the standard emission control system and for modified configurations (Ref. 8-19). These data, listed in Table 8-11, show that substantial fuel economy and power losses are associated with the achievement of extremely low NO_x emission levels. For example, a fuel economy loss of 33% and a power loss of 28% are incurred in achieving the 1976 federal emission standards. Catalytic treatment of the exhaust gases combined with moderately retarded combustion results in a 5% to 10% fuel economy degradation (Table 8-11, Case 11). As illustrated in Table 8-12, similar results were obtained by Texaco for a TCCS-powered Plymouth Cricket automobile equipped with EGR and oxidation catalysts and tested with a 2500-lb inertia weight. The best fuel economy obtained with this vehicle without EGR was 25.3 mpg. This compares with 20.6 mpg for the average 1973 model year Cricket certification vehicle tested at 2250-lb inertia weight (Ref. 8-27).

Level-road fuel economy data for the naturally aspirated L-141 TCCS engine, installed in an M-151 light-duty vehicle, are presented in

Figure 8-22 (Ref. 8-19). The fuel economy of the TCCS-powered vehicle is considerably better than the fuel economy of the vehicle with the standard engine, especially at low vehicle speeds. For example, at 30 mph TCCS has a 30% better fuel economy, and at 55 mph the difference is still 20%. The variations shown in Figure 8-22 for the different fuels are directly related to the fuel heating value per gallon and confirm the previously noted trends of specific fuel consumption (Figure 8-20).

Except for vehicle speeds below about 30 mph, the level-road fuel economy of the turbocharged M-151 TCCS vehicle is about 5% to 25% better than that of the naturally aspirated TCCS engine, with the higher improvement realized at vehicle speeds of 50 mph and above (Ref. 8-19).

Fuel economy tests over a closed-road course conducted by TACOM on TCCS and standard engine-powered M-151 vehicles support the Texaco results. On the basis of the TACOM tests, the fuel economy of the

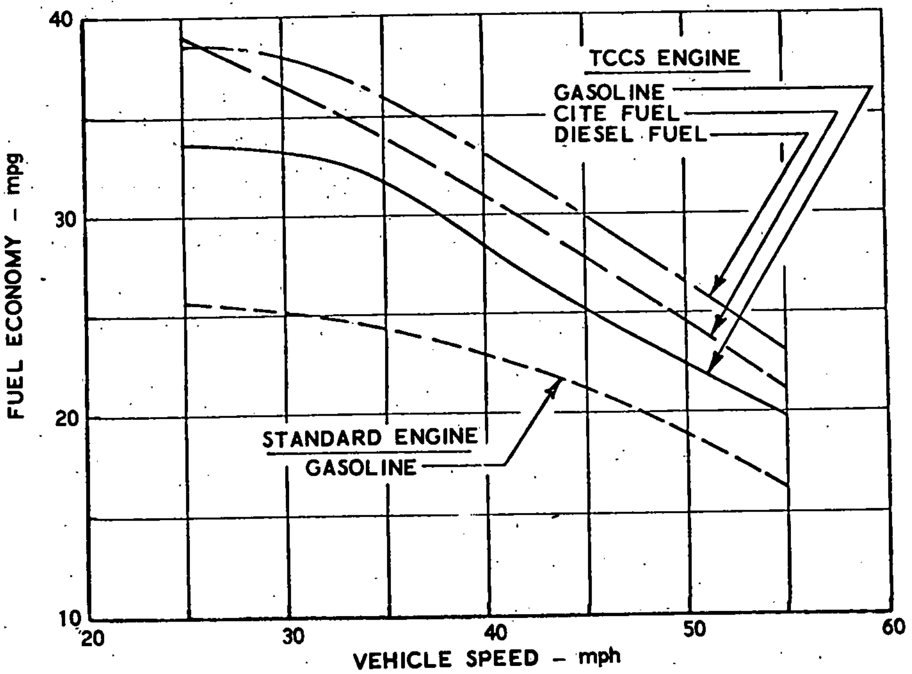


Figure 8-22. Level-road fuel economy, M-151 light-duty vehicle, Texaco naturally aspirated, L-141 TCCS engine (Ref. 8-19)

TCCS vehicle is between 39% and 73% better, depending upon the average speed over the course (Ref. 8-19). Again this illustrates the excellent part-load fuel economy characteristics of the TCCS engine.

8.2.2.4.2 Ford PROCO and Fast Burn Engines

The improved fuel economy characteristics of open-chamber, stratified charge engines with respect to conventional spark ignition engines were demonstrated by Ford early in its development program (Ref. 8-28). At that time, 430-CID FCP engines were installed in three vehicles and tested at constant speeds between 30 and 70 mph. The data show a 28% to 34% improvement in the fuel economy of the FCP vehicles compared with similar standard vehicles.

Fuel economy data for the M-151 vehicle equipped with conventional and PROCO engines and tested over the FDC are presented in Table 8-13. The values were computed from the measured exhaust-gas composition by means of the carbon-balance method (Ref. 8-20).

The fuel economy of the PROCO vehicle adjusted to meet the 1975 statutory emission standards at low mileage is about 24% better than that of the conventional-engine-powered M-151 vehicle and 12% better than the average 1973 model year vehicle tested at 2750-lb inertia weight (Ref. 8-29). When adjusted for best efficiency, the fuel economy of PROCO is approximately 36% better than the average 1973 certification car tested at 2750-lb inertia weight. However, this improvement results in a substantial increase of the HC, CO, and NO_x emissions.

Fuel economy data obtained by Ford for its PROCO engine installed in a 1972 Montego vehicle are listed in Tables 8-14 and 8-15. The computed fuel economy of this vehicle over the Federal Driving Cycle is about 24% better than that of the average 1972 certification vehicle in the 4500-lb weight class.

The fuel economy of the Fast Burn Phase I, 351-CID engine, installed in a 1972 Torino automobile and tested according to the 1975 FTD, is listed in Table 8-16. The vehicle meets 1976 emission standards at low mileage with a definite improvement in fuel economy over a standard Torino.

After 25,000 mi, the fuel economy of the vehicle is essentially unchanged although the HC and CO emissions have increased considerably (Ref. 8-21).

As shown in Table 8-17, the fuel economy of the Ford Fast Burn Phase II vehicle adjusted to low NO_x levels is not very impressive and is equivalent to conventional 1973 Torino cars. However, the fuel economy improves substantially by adjusting the NO_x emissions to the 1.5-gm/mi level (Ref. 8-21).

8.2.3 Divided Chamber Stratified Charge Engines

8.2.3.1 Honda CVCC Engine

8.2.3.1.1 Description

The original Honda CVCC (compound vortex controlled combustion) engine is an in-line, 4-cylinder, 4-stroke, 1948 cm^3 displacement (119 $in.^3$) engine with a rating of 70 bhp at 5000 rpm (Ref. 8-30). Prechamber volume is in the range of 10% to 20% of the total combustion chamber volume and both parts of the chamber communicate through a small orifice.

Figure 8-23 shows the principle of the design. The smallest venturi of the 3-barrel carburetor supplies a rich mixture to each prechamber. The other two venturis supply the main engine chambers with a very lean mixture. The main chamber utilizes a conventional inlet valve located in the chamber, while the mixture to the prechamber enters through a small inlet valve that opens into the prechamber. The prechamber inlet valve is actuated by means of an independent small rocker arm driven by the engine camshaft. The charge is stratified into three air-fuel ratio regions: (1) a rich region in the prechamber, (2) a lean region in the main chamber, and (3) a region of moderate mixture strength in the vicinity of the orifice connecting both chambers. Combustion is initiated by the spark plug located in the prechamber.

According to Honda, combustion in the CVCC engine is controlled and proceeds at very slow rates to reduce the peak combustion temperature and the formation of NO. However, the temperatures are maintained high enough for a sufficiently long period of time during the expansion

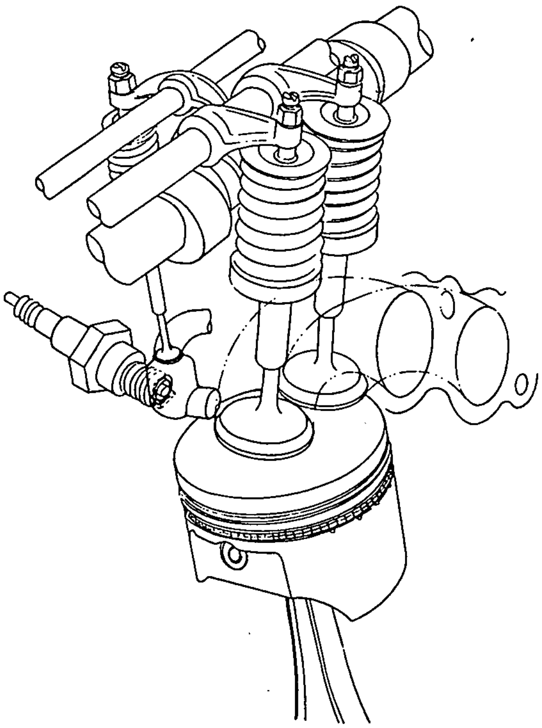


Figure 8-23. Honda CVCC divided-chamber stratified charge engine (Ref. 8-30)

stroke that the oxidation of CO and HC is completed. Conventional unmodified spark plugs are used on this engine. Apparently, optimum values exist for this engine with respect to configuration and size of the prechamber and the relative positions of the auxiliary intake valve, spark plug, torch nozzle, and main combustion chamber. The air-fuel ratios supplied to the prechamber and the main chamber vary with the operating conditions and are controlled with respect to prechamber scavenging, desired emission levels, and other performance parameters.

8.2.3.1.2 Emissions

Evaluation of the emission characteristics of the Honda CVCC engine was conducted by the EPA (Refs. 8-31 and 8-32) and by the Honda Motor Company (Ref. 8-30). In the EPA test program, three Honda Civic cars equipped with CVCC engines were tested according to the 1975 FTP, using an inertia weight of 2000 lb. Two of the vehicles had odometer readings of about 1500 mi, and the third car had just completed a 50,000-mi durability run. A summary of these emission tests is presented in Table 8-18 showing that the emissions from the vehicles were consistently below the 1975 federal emission standards.

One of the three cars was also tested with an inertia weight of 3000 lb. The vehicle had adequate power to keep up with all the accelerations of the FDC. Because of time constraints, this was a hot-start test; i.e., the start of the test did not follow the standard 12-hr vehicle soak

Table 8-18. Emissions and Fuel Economy, Honda CVCC Vehicles[a]

Vehicle	Emissions, gm/mi, (1975 FTP)			Fuel Economy, mpg	
	HC	CO	NO_x	1975 FTP	1972 FTP
Low mileage car (No. 3652: five tests.)					
Average	0.18	2.12	0.89	22.1	21.0
Maximum[b]	0.21	2.28	1.05	22.4	21.5
Minimum[b]	0.15	1.96	0.75	21.9	20.6
Low mileage car (No. 3606: one test.)	0.23	2.00	1.03	20.7	19.5
50,000-mile car (No. 2034: four tests.)					
Average	0.24	1.75	0.65	21.3	19.8
Maximum[b]	0.26	1.85	0.73	22.2	20.0
Minimum[b]	0.19	1.70	0.57	20.8	19.5

[a] Ref. 8-33.

[b] The values of the different columns do not necessarily correspond to the same test.

period but began shortly after the car had been tested at 2000-lb inertia weight. Although the initial period of the cold start was omitted, the test results appear to show that the emissions of a Honda vehicle of 3000 lb, equipped with the same CVCC engine, would be lower than the statutory 1975 emission standards (Ref. 8-31).

Particulate emission tests conducted by Dow Chemical Company under EPA contract indicate that the particulate emissions from the CVCC vehicles are comparable to those from conventional engines using equivalent fuels.

To date, Honda has conducted emission durability tests on seven 2.0-liter CVCC vehicles accumulating a total of 437,000 mi. On the basis of these tests, the deterioration factors are approximately 1.10 to 1.15 for HC, 1.05 to 1.10 for CO, and 1.00 to 1.05 for NO_x, showing good emission control durability for the CVCC engines.

Honda has adapted the CVCC process to large engines to demonstrate the feasibility of the concept in full-size American cars. Table 8-19 presents the Honda emission test results for a Chevrolet Vega car equipped with a 140-CID CVCC engine. The emissions are below the statutory 1975 standards, and the fuel economy is slightly better than for the nonmodified Vega.

Emission data obtained by the EPA for a CVCC-modified 350-CID Chevrolet Impala automobile are listed in Table 8-20 (Ref. 8-32). Tests 1 and 4 demonstrate that CO and HC emissions below the statutory 1975-76 levels can be achieved with this vehicle and that NO_x levels are consistently below the 1976 interim standard of 2.0 gm/mi. During Tests 2 and 3, problems associated with the hot soak period resulted in high levels of CO and HC during the Bag 3 portion of the tests. The high CO level in Bag 3 of Test 2 appeared to be associated with flooding of the prechamber carburetor. This was corrected by Honda technicians prior to Test 3. The high HC level in Bag 3 of Test 3 was apparently caused by a false hot start which resulted in excessive cranking by the EPA driver. Another false start occurred on Test 4, but the more experienced Honda driver was able to

Table 8-19. Low-Mileage Emissions and Fuel Economy of Vega Vehicles Modified for CVCC[a]

Vehicle	Test Date	Emissions, gm/mi[b]			Fuel Economy, mpg[c]	No. of Tests
		HC	CO	NO_x		
CVCC - 140 CID	9-72	0.26	2.9	1.2	17.9	2
CVCC - 140 CID, improved	11-72	0.26	2.6	1.2	18.6	2
CVCC - 140 CID improved	4-73	0.22	2.4	1.2	18.9	3
Original 140 CID	-	2.1	10.6	3.8	17.2	2

[a] Ref. 8-3.

[b] 1975 FTP.

[c] 1972 FTP.

Table 8-20. Low-Mileage Emissions and Fuel Economy of Impala Vehicles Modified for CVCC[a]

Test No.[b]	Emissions, gm/mi[c]			Fuel Consumption, mpg
	HC	CO	NO_x	
1	0.27	2.88	1.72	10.5
2	0.23	5.01	1.95	11.2
3	0.80	2.64	1.51	10.8
4	0.32	2.79	1.68	10.2

[a] Ref. 8-32.

[b] Tests run at 5000-lb inertia and 14.7 rear wheel hp at 50 mph (hp includes 10% for air conditioning).

[c] 1975 FTP.

recover with less HC penalty. It is believed that the hot-start problems experienced on the prototype vehicle can be corrected with further engineering effort.

The consistently low cold-start (Bag 1) HC and CO levels during all of the tests should be emphasized. An average value for Bag 1 of 0.38 gm/mi HC and 3.51 gm/mi CO was observed. Good control of NO_x was also demonstrated with this vehicle. An average value equal to 1.72 gm/mi was obtained for the 1975 FTP testing.

8.2.3.1.3 Fuel Economy

Average, maximum, and minimum fuel economy data reported by Honda for its CVCC-powered test vehicles are listed in Table 8-18. The average 1975 FTP fuel economy of the three Honda vehicles tested (21.3 mpg) was 16% lower than the average of the 2000-lb 1973 certification vehicles tested by EPA (25.5 mpg). Honda data comparing the CVCC-powered Civic to the standard Civic with a conventional engine also show 10% fuel economy penalty for the CVCC version. However, the fuel economy of this car at 3000-lb inertia weight (18.7 mpg) compared quite favorably with both of the average 3000-lb 1973 certification vehicles (16.2 mpg). Comparison of the data in Tables 8-19 and 8-20 indicates slightly better fuel economy for the CVCC-modified Vega and Impala vehicles.

8.2.4 NO_x Versus Fuel Economy for Stratified Charge Engines

The relationship between NO_x emission level and vehicle fuel economy (as measured on the FTP) are shown in Figures 8-24 through 8-26 for the Texaco TCCS, Ford PROCO and Fast Burn, and Honda CVCC, respectively. As can be noted, stratified charge engines have achieved NO_x levels in the 0.4 gm/mi range with EGR.

Open-chamber engines (Texaco TCCS and Ford PROCO) exhibit improved fuel economy (~ 10%) at these low NO_x levels when compared with the original conventional spark ignition engine in the same vehicle. The original engine also produced appreciably higher NO_x levels (3-4 gm/mi). Increasing the NO_x level in the TCCS and PROCO engines by reducing EGR

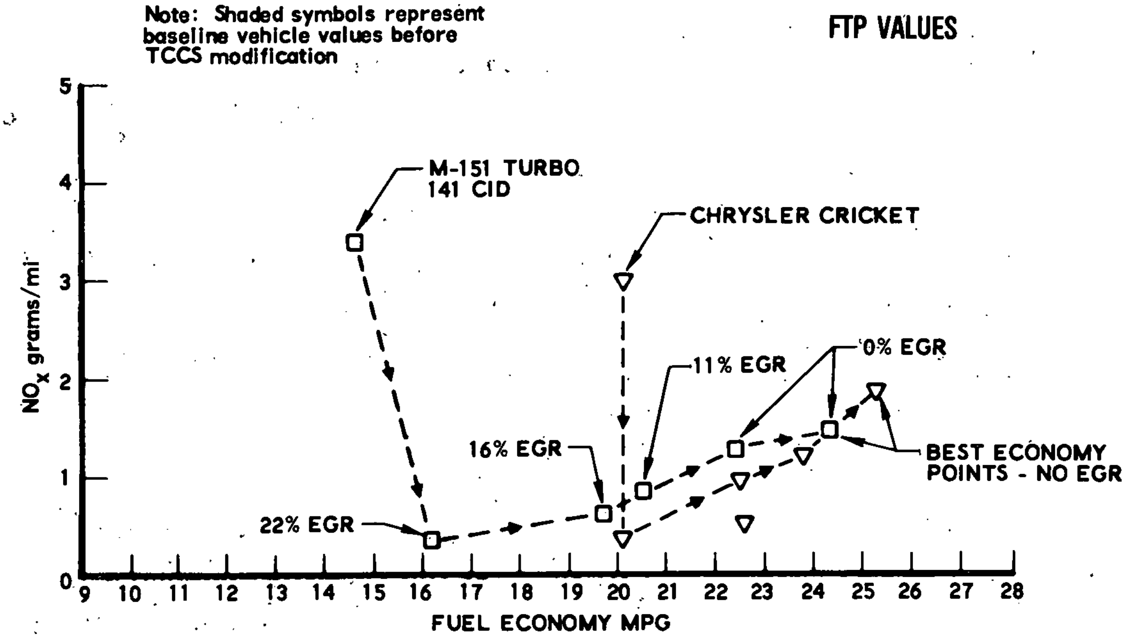


Figure 8-24. Texaco TCCS: NO_x vs fuel economy

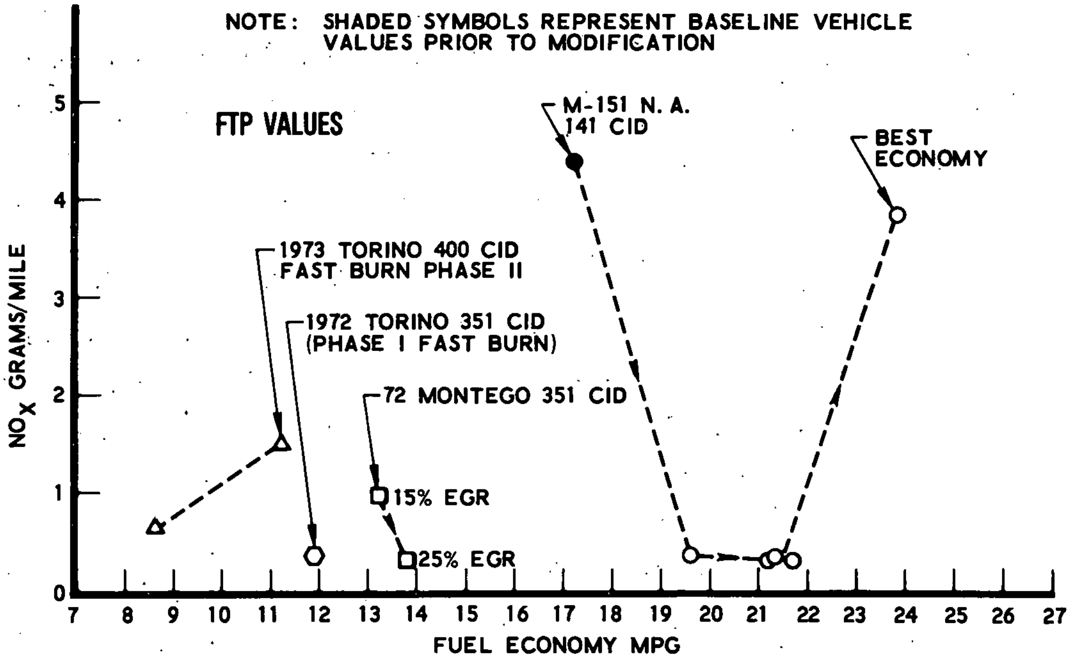


Figure 8-25. Ford PROCO and Fast Burn: NO_x vs fuel economy

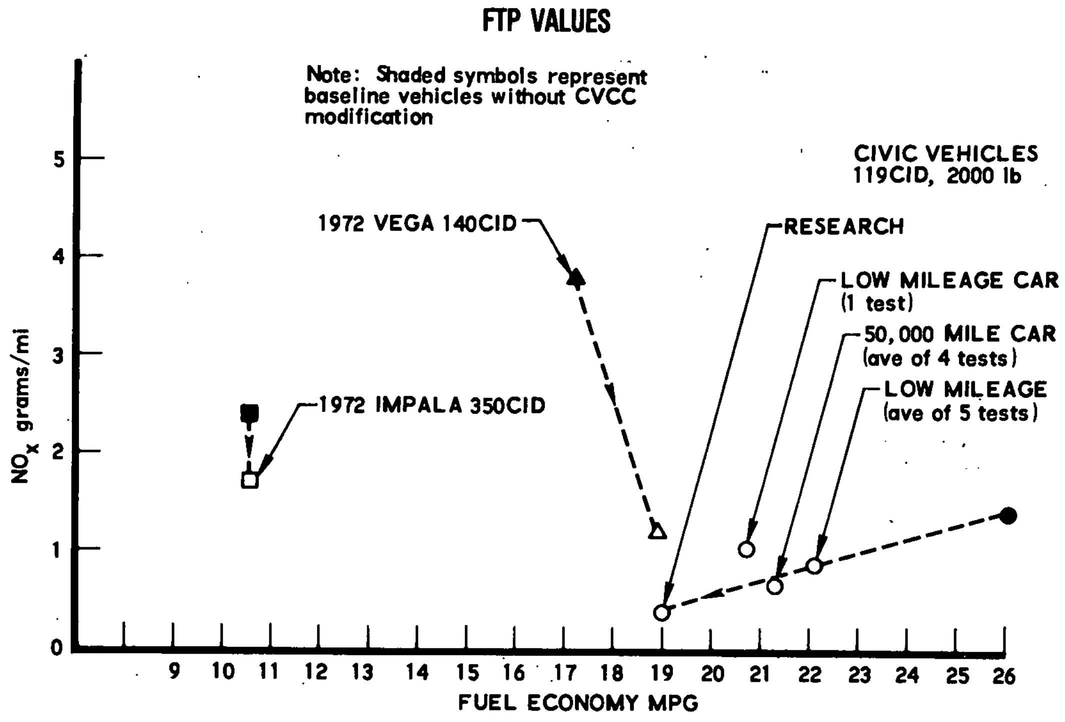


Figure 8-26. Honda CVCC : NO_x vs fuel economy

flow rates results in still further improvements in fuel economy over the conventional engine at the same NO_x level.

The Honda prechamber engine data exhibit a mixed picture. The Honda CIVIC vehicle with the prechamber engine had a fuel economy loss (~ 25%) while at the same time decreasing the NO_x emissions to ~ 0.4 gm/mi from the value of ~ 1.4 gm/mi obtained with the standard or conventional engine. A 1972 Chevrolet Vega, modified by Honda with the CVCC prechamber engine, had a 10% fuel economy improvement over the standard Vega engine and at the same time decreased NO_x from 3.8 to 1.2 gm/mi.

REFERENCES FOR SECTION 8

8-1. E. F. Obert, Internal Combustion Engines, Third Ed., International Textbook Company, Scranton, Pa.

8-2. A Study of the Influence of Fuel Atomization, Vaporization and Mixing Processes on Pollutant Emissions from Motor Vehicle Powerplants, Phase I Report, Battelle Memorial Institute (30 April 1969).

8-3. J. A. Robison and W. M. Brehob, "The Influence of Improved Mixture Quality on Engine Exhaust Emissions and Performances," Paper No. WSCI 65-17, Western States Combustion Institute Meeting (October 1965).

8-4. K. Matsumoto, T. Toda, and H. Nohira, "Oxides of Nitrogen from Smaller Gasoline Engine," SAE Paper No. 700145 (January 1970).

8-5. D. J. Patterson and N. A. Henein, Emissions from Combustion Engines and Their Control, Ann Arbor Science Publishers, Inc., Ann Arbor, Mich. (1972).

8-6. P. H. Schweitzer, "Control of Exhaust Pollution through a Mixture Optimizer," SAE Paper No. 720254 (January 1972).

8-7. V. R. Grundman, Jr., "Testimony Before the Environmental Protection Agency Concerning the Dresserator System (2 July 1973).

8-8. "Visit to Dresser Industries on November 7, 1973," Trip Report Memorandum, Environmental Protection Agency (12 November 1973).

8-9. "Confirmatory Test of 'Dresserator' Emission Control System," Vehicle Emissions Laboratory Report, California Air Resources Board, Sacramento, Calif. (May 1973).

8-10. R. D. Englert, Environmental Technology Division of Dresser Industries, Letter and Data Attachment, Santa Ana, Calif. (12 June 1974).

8-11. "Evaluation of the Dresserator Emission Control System," Technology Assessment and Evaluation Branch, Emission Control Technology Division, Environmental Protection Agency (August 1974).

8-12. R. D. Englert, Environmental Technology Division of Dresser Industries, Letter and Data Attachment, Santa Ana, Calif. (19 July 1974).

8-13. "Consequences of Removing Lead Antiknocks from Gasoline," Ethyl Corporation Report (August 1970).

8-14. "Evaluation of the Ethyl Lean Thermal Reactor System," Emission Control Technology Division, Environmental Protection Agency (June 1973).

8-15. W. E. Adams, F. J. Marsee, and D. L. Lanane, "Lead-Compatible Emission Controls - A Route to Improved Fuel Economy," NPRA Paper No. F&L-74-60, Ethyl Corporation, National Petroleum Refiners Association (7-8 November 1974).

8-16. Current Status of Alternative Automotive Power Systems and Fuels. Volume II: Alternative Automotive Engines, ATR-74(7325)-1, Vol. II, The Aerospace Corporation, El Segundo, Calif. (July 1974).

8-17. E. Mitchell, J. M. Cobb, and R. A. Frost, "Design and Evaluation of a Stratified Charge Multifuel Military Engine," SAE Paper 680042 (January 1968).

8-18. E. Mitchell et al., "A Stratified Charge Multifuel Military Engine - A Progress Report," SAE Paper 720051 (January 1972).

8-19. W. J. Coppoc et al., "Texaco Controlled Combustion System Provides an Engine with Clean Exhaust and Good Fuel Economy," Swedish Engineering Society, Stockholm, Sweden (27 March 1973).

8-20. A. Simko, M. A. Choma, and L. L. Repko, "Exhaust Emission Control by the Ford Programmed Combustion Process - PROCO," SAE Paper 720052 (January 1972).

8-21. "Request for Suspension of 1976 Motor Vehicle Exhaust Emission Standards," Vol. I, Sect. 1-4 - Document Notes, Ford Motor Company, Detroit (18 June 1973).

8-22. H. L. Gompf, "Evaluation of the Texaco Stratified Charge (TCP) M-151 Army Vehicle," Report 73-3, Environmental Protection Agency, Ann Arbor, Mich. (August 1972).

8-23. "Evaluation of the Texaco Stratified Charge (TCCS) M-151 Army Vehicle," Report 73-27 DWP, Emission Control Technology Division, Environmental Protection Agency, Ann Arbor, Mich. (June 1973).

8-24. M. Alperstein, Informal Submittal of Test Data by Texaco (2 August 1973).

8-25. "Application for Suspension of 1976 Motor Vehicle Emission Standards, Part I." Chrysler Corporation, Detroit (1973).

8-26. Environmental News, Environmental Protection Agency, Washington, D.C. (24 September 1971).

8-27. Federal Register, Part III, Vol. 18, No. 84, Environmental Protection Agency, Washington, D.C. (2 May 1973).

8-28. I. N. Bishop and A. Simko, "A New Concept of Stratified Charge Combustion - The Ford Combustion Process (FCP)," SAE Paper 680041 (January 1968).

8-29. T. A. Austin and K. H. Hellman, "Passenger Car Fuel Economy - Trends and Influencing Factors," SAE Paper 730790 (September 1973).

8-30. Hearings Before the Subcommittee on Air and Water Pollution of the Committee on Public Works, U. S. Senate, Serial No. 93-H9, Washington, D.C. (May 1973).

8-31. T. C. Austin, "An Evaluation of Three Honda Compound Vortex Controlled Combustion (CVCC) Powered Vehicles," Report 73-11, Environmental Protection Agency (December 1972).

8-32. "An Evaluation of a 350 CID Compound Vortex Controlled Combustion (CVCC) Powered Chevrolet Impala," Report 74-13 DWP, Environmental Protection Agency (October 1973).

8-33. I. A. El-Messiri and H. K. Newhall, "Recent Results with the Divided Combustion Chamber Concept," Midwestern States Section, The Combustion Institute, Ann Arbor, Mich. (Spring 1971).

SECTION 9

COMPARISON OF LEAN COMBUSTION APPROACHES

The following sections briefly compare the various lean combustion approaches examined in the study in the areas of current status, emission control, fuel economy, and potential areas of applicability.

9.1 CURRENT STATUS

9.1.1 Hydrogen Addition

Lean combustion with hydrogen addition can be implemented with supplementary hydrogen stored onboard the vehicle or hydrogen generated from the primary fuel (gasoline) by means of an onboard reformer. Experimental vehicles have been tested with the hydrogen stored onboard in compressed gas and cryogenic liquid form. This approach, however, is not considered feasible in the near future for general automotive application because of (1) excessive volume and weight requirements and (2) the lack of an extensive distribution system to provide refueling for the motoring public.

Hydrogen can also be stored onboard the vehicle using a metal hydride bed. However, the currently envisioned hydride beds are heavy and require heat exchangers and heaters to remove the stored hydrogen from the bed. Much additional research on metal hydrides and development of the associated systems are needed before metal hydride storage could be adequately and definitively assessed as to its ultimate capability. Thus, it is not considered a viable option in the reasonably near future for general automotive application.

A potentially convenient means of obtaining the hydrogen is the use of an onboard fuel reformer to convert a portion (or all) of the gasoline fuel to gaseous products which contain hydrogen (plus CO, CH_4, etc.). Although fuel reformation is a conventional and routine process in the oil industry, its application to automobiles is a recent area of endeaver. The partial oxidation fuel reformer concepts of JPL and Siemens have been developed in prototype form and evaluated in engine dynamometer tests. No vehicle-mounted reformer tests have been made to date. Several currently unresolved potential problem areas for automotive fuel reformer systems include:

a. Reformer startup characteristics.
b. Reformer durability and lifetime.
c. Control system development, including the ability to schedule reformer fuel flow rates and accomodate transient characteristics.
d. Associated HC control measures, e.g., low temperature lightoff and close proximity oxidation catalysts.

9.1.2 Advanced Carburetors

Carbuetor systems enabling precision air-fuel ratio control and good charge mixing have been in development for several years. The three-venturi Ethyl carburetor was tested in prototype form in 1970. More recently, the variable venturi sonic flow Dresserator system has been extensively evaluated by Dresser Industries and the EPA with a prototype carburetor in vehicle tests. Dresser Industries is currently conducting manufacturing evaluations and is scheduled soon to test preproduction-type models. The Ford Motor Company and the Holley Carburetor Company are licensees of the Dresser concept and therefore should also be examining this carburetor for mass-manufacture characteristics.

9.1.3 Open-Chamber Stratified Charge Engines

For the past several years, Texaco and Ford, under contract to the USA TACOM, have been involved in the development of stratified charge engines for potential application in the military L-141 light-duty engine.

The stratified charge engines manufactured to date by Texaco and Ford are hand-built experimental models which are not suitable for mass production. Specifically, the current cylinder-head design is not compatible with mass-production techniques, and new production equipment and assembly procedures are required for the manufacture of the fuel injection and air-fuel control systems.

A design study for a family of 4-, 6-, and 8-cylinder PROCO engines has been recently completed by Ford, with parts commonality being a prime consideration. On the basis of this study, Ford has concluded that the PROCO engine concept can be incorporated in these engine designs.

9.1.4 Divided-Chamber Stratified Charge Engines

The Honda CVCC prechamber engine has been subjected to extensive performance tests and evaluations and has achieved limited production status in that 1975 model year CIVIC vehicles are being sold in California.

9.2 EMISSION CONTROL

9.2.1 Fuel Reformer Systems

As discussed in Section 4.2, lean operation with fuel reformation can result in considerably reduced NO_x levels over conventional engine operation, according to the available engine plus fuel reformer dynamometer data. The exact level of NO_x emissions in gm/mi over the FDC would, of course, be a function of the particular air-fuel ratio employed, i.e., the leaner the operation, the lower the NO_x. It appears, however, that operation in the 0.5 to 0.6 equivalence ratio range (as enabled by the presence of hydrogen in the mixture) could result in NO_x levels in the range of 0.4 gm/mi (the 1978 emission NO_x standard) or less. CO should also be below the 3.4 gm/mi standard at these operating conditions. However, the data indicate that HC levels are quite high and would require aftertreatment (e.g. catalytic or thermal reactors).

9.2.2 Advanced Carburetors

Although capable of sustaining lean operation up to air-fuel ratios of ~ 22 ($\Phi \sim 0.68$) or more, most advanced carburetor tests are made in the A/F = 17-19 range ($\Phi = 0.79$ to 0.88) in order to simultaneously reduce HC, CO, and NO_x and at the same time avoid any thermal efficiency losses caused by slower flame speeds at the ultralean conditions. In this range of operation (A/F = 17-19), the California 1975 standards (HC = 0.9, CO = 9.0, NO_x = 2.0) have been demonstrated with the Dresser system without oxidation catalysts. A form of lean thermal reactor was also incorporated in the system by enlarging and lagging the exhaust system. Indeed, NO_x levels below 1.5 gm/mi have been accomplished with this approach. This concept would be suitable for the addition of oxidation catalysts for further HC and CO control.

9.2.3 Open-Chamber Stratified Charge Engines

Without incorporation of emission control equipment such as EGR, oxidation catalysts, and intake throttling, the HC and NO_x emissions from the L-141 stratified charge engines are comparable to conventional spark ignition engines, while CO is substantially lower.

With emission control, the stratified charge engine powered vehicles tested to date over the FDC meet the 1978 statutory emission standards at low mileage. However, rapid deterioration of the emission control system performance has been a major problem area.

9.2.4 Divided-Chamber Stratified Charge Engines

Based on Honda test data for the CIVIC vehicle and CVCC-modified versions of the GM Vega and Impala vehicles, it appears that pre-chamber engine-powered vehicles in the 2000-lb weight class would be capable of about 0.4 gm/mi HC, 2 to 4 gm/mi CO, and 0.6 to 1.0 gm/mi NO_x. Vehicles in the 5000-lb weight class would have NO_x emissions of about 1.4 to 1.7 gm/mi, while the HC and CO emissions would be similar to the levels of the 2000-lb vehicle.

9.3 FUEL ECONOMY

9.3.1 Fuel Reformer Systems

Definitive values of fuel economy for fuel-reformer equipped vehicles are not available. As discussed in Sections 4.4 and 4.5, the increase in engine thermal efficiency occurring at ultralean operating conditions is offset to some degree by the inefficiency of the reformer in producing the gaseous products necessary to sustain ultralean operation. It appears, however, that the net benefit in fuel economy would not be any greater than that given by advanced carburetors operating in the A/F = 17 to 19 range.

9.3.2 Advanced Carburetors

Tests of the Ethyl and Dresser systems in the A/F = 17 to 19 range have indicated fuel economy improvements of 10% to 15% over baseline vehicles, while at the same time reducing NO_x levels over that of the baseline vehicle. Dresser vehicle tests incorporating some additional spark advance have shown fuel economy improvements up to 25% (at $NO_x < 1.5$ gm/mi) with a 1973 Capri.

9.3.3 Open-Chamber Stratified Charge Engines

Without incorporation of emission control equipment, the fuel economy of the L-141 stratified charge engine was in the order of 30% higher than for the equivalent conventional engine.

With emission control (EGR, oxidation catalysts, intake throttling) sufficient to meet the 1978 emission standards at low mileage, the fuel economy was equal to or slightly better than average 1973 certification vehicles tested at the same inertia weight.

9.3.4 Divided-Chamber Stratified Charge Engine

According to the Honda test data, the prechamber engine gives fuel economy similar to equivalent conventional 1974 light duty vehicles while achieving lower emission levels (e.g., HC $\sim$ 0.4 gm/mi, CO $\sim$ 2 to 4 gm/mi, $NO_x \sim$ 0.6 to 1.0 gm/mi).

9.4 POTENTIAL AREAS OF APPLICABILITY

The need for or the potential applicability of a given concept is a complex function of a number of key requirements including:

a. Emission control standards
b. Fuel economy characteristics
c. Relative state of development

The requirement to simultaneously reduce HC, CO, and NO_x to a given set of statutory standards and achieve the best possible fuel economy has been of concern since the passage of the Clean Air Act. The more recent emphasis on fuel economy due to the oil embargo and concern for natural resource conservation has only served to accentuate this well-known dilemma.

It is generally agreed that NO_x is the most difficult to control to prescribed levels and that the constraints on engine design, operation, and parametric adjustments which most adversely affect fuel economy are NO_x - related. For example, oxidation catalysts offer a convenient method for control of HC and CO without requiring adjustments to impair fuel economy. Conversely, EGR and spark retard, which have been used to date for NO_x control in conventional spark ignition engines, both adversely affect fuel economy. This is the principal reason that significant efforts have gone into alternative approaches which offer solutions to the NO_x versus fuel economy problem, with considerable emphasis in the general area of lean combustion and its inherent characteristics of lower NO_x concentrations plus potential thermal efficiency advantages.

The promulgation of interim 1975 model year standards and proposals to reset the original NO_x statutory level of 0.4 gm/mi to a higher level have served to confuse the relative advantages of the numerous concepts under evaluation. Since methods are available for HC and CO control which appear generally applicable across the spectrum of lean combustion concepts (e.g., catalytic or thermal reactors), it seems meaningful to assess the potential applicability of the various concepts as a function of the NO_x FTP standard for compatibility, particularly since all available evidence indicates that, to a first approximation, fuel economy is directly affected by the NO_x level achieved for a given concept.

Table 9-1 is one such expression of potential applicability of the concepts examined herein when compared to current methods of NO_x control and to catalytic methods of NO_x control which essentially limit engine operation to near-stoichiometric mixtures.

At a NO_x standard of 3.1 gm/mi, existing vehicles employ EGR and spark retard for NO_x control. On a sales weighted basis, 1973 model year vehicles meeting this NO_x standard had fuel economy reduced ~10% over precontrolled vehicles. Similar systems were employed to meet the 1975-76 California NO_x standard of 2 gm/mi, with even further fuel economy losses.

Table 9-1. Potential Areas of Applicability

Engine-System Approach	NO_x FTP Standard				
	3.1	2.0	1.5	1.0	0.5 and Below
Homogeneous mixtures					
• EGR plus spark retard	X	X			
• Advanced lean carburetion	X	X	X		
• H_2-addition fuel reformers				X	X
Stratified charge					
• Honda CVCC			X	X	X
• Texaco TCCS			X	X	X
• Ford PROCO			X	X	X
• Ford Fast Burn			X	X	X
Reduction catalyst				X	X
Tricomponent catalyst				X	X

In the interest of fuel economy, it would be preferable to use an alternate approach for levels below this value and also for levels of 2 gm/mi and above if it would also improve fuel economy.

The advanced lean carburetion approach, as exemplified by the Dresserator System, offers this capability. The minimum NO_x level for which advanced lean carburetion is applicable is not established. Dresser data acquired to date have been close to 1 gm/mi in some cases, but the preponderance of the data indicates that the cutoff of applicability should be 1.5 gm/mi for current models of the device, at a minimum.

Fuel reformers which offer the potential for extremely low NO_x levels with ultralean combustion fill out the applicability matrix for homogeneous mixture systems. Although these devices would meet NO_x levels above 1 gm/mi, they are not shown as being applicable because they are not developed and do not appear to offer any advantage over lean carburetors, which are in a more advanced state of development, are simpler, and are cheaper.

The various stratified charge engine concepts shown all offer the potential of very low NO_x levels by incorporating large amounts of EGR. At successively higher NO_x levels, the EGR would be reduced or eliminated. At their present state of development, they would not appear to be viable candidates for production unless the NO_x standard were below 2.0 gm/mi. The Honda CVCC engine is a notable exception to this prognosis in that it is currently in production and sold in California to meet the 2.0 gm/mi standard. However, its development was undertaken to avoid the use of oxidation catalysts, which have been generally used by all other manufacturers except Toyo Kogyo, which uses a thermal reactor with its rotary engine. Even at 1.5 gm/mi, it is doubtful that significant numbers of stratified charge engines would be produced unless they could also offer significant fuel economy advantages over an advanced lean carburetor.

Reduction and tricomponent catalysts have the potential to meet NO_x levels below 0.5 gm/mi. Their development history has been extremely poor with regard to durability of operation. In addition, they limit engine operation to the slightly rich side of operation in order to reduce NO_x and thus do not offer the potential for increased engine thermal efficiency offered by lean combustion approaches. It appears extremely doubtful that such catalysts would be employed to meet NO_x levels > 1.0.

GLOSSARY

AAPS	Alternative Automotive Power Systems (formerly a Division of Environmental Protection Agency)
A/F	air-fuel ratio
AMA	Automobile Manufacturer's Association
bhp	brake horsepower
BMEP	brake mean effective pressure
BSFC	brake specific fuel consumption
BTDC	before top dead center
CFR	Cooperative Fuel Research Council
CID	cubic inch displacement
CITE	compression ignition and turbine engine
CO	carbon monoxide
CR	compression ratio
CVS	constant volume sampling
EGR	exhaust gas recirculation
EPA	Environmental Protection Agency
ERDA	Energy Research and Development Administration
FDC	federal driving cycle
Fe	iron
FHDC	federal highway driving cycle
fhp	friction horsepower
FTP	federal test procedures
GM	General Motors

H_2	hydrogen
HC	hydrocarbon
ICE	internal combustion engine
ihp	indicated horsepower
ISFC	indicated specific fuel consumption
JPL	Jet Propulsion Laboratory
MBT	minimum spark advance for best torque
Mg	magnesium
mpg	miles per gallon
NAPCA	National Air Pollution Control Administration
NASA	National Aeronautics and Space Aministration
Ni	nickel
NO_x	oxides of nitrogen
Pb	lead
RON	fuel research octane number
rpm	revolutions per minute
SFC	specific fuel consumption
TACOM	U.S. Army Tank Automotive Command
TCCS	Texaco Controlled Combustion System
TDC	top dead center
TFM	turbulent flow manifold
TFS	turbulent flow system
Ti	titanium
WOT	wide open throttle

*U.S. GOVERNMENT PRINTING OFFICE: 1977-740 - 062/20