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LEAN MIXTURE ENGINES TESTING AND EVALUATION PROGRAM

Volume I: Executive Summary

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16. Abstract

This report is aimed at defining analytically and demonstrating experimentally the potential of the "lean-burn concept". Fuel consumption and emissions data are obtained on the engine dynamometer for the baseline engine, and two lean-burn configurations of the same engine and data comparisons are made. Individual cylinder equivalence ratios are measured to evaluate the cylinder-to-cylinder distribution. Pressure-time traces from individual cylinders are used to get information about ignition delay, combustion duration and cycle-to-cycle pressure variations. Fuel consumption and emissions data for one lean-burn configuration are obtained over the Federal Driving Cycle using a chassis dynamometer and the results are compared with the stock baseline results. Using experimental results and information from the existing literature, the potential of the "lean-burn concept" is assessed using the Blumberg-Kummer cycle analysis program.

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PREFACE

This report, prepared by the Jet Propulsion Laboratory for the U.S. Department of Transportation, presents a study of the potential of the lean burn concept in the areas of fuel economy and exhaust emissions. The report consists of three volumes.

Volume I, the Executive Summary, presents a summary of the experimental equipment and procedure, together with the experimental results and conclusions.

Volume II, the main body of the report, provides a Comprehensive Discussion of each engine improvement and supporting data, the potential of the lean burn approach, and conclusions and recommendations.

Volume III, the Appendices, presents supplemental analysis methods, fuel economy and emissions test data, and a description of the instrumentation used in the testing and evaluation program.

The status of the technology reported is that available in the time period of 28 May 1974 through 31 December 1974.

SUMMARY

Experimental results for fuel consumption and emissions are presented for a 350 CID (5.7 liter) Chevrolet V-8 engine modified for lean operation with gasoline. The lean burn engine achieved peak thermal efficiency at an equivalence ratio of 0.75 and a spark advance of 60° BTDC. At this condition the lean burn engine demonstrated a 10% reduction in brake specific fuel consumption compared with the stock engine; however, NO_x and hydrocarbon emissions were higher.

With the use of spark retard and/or slightly lower equivalence ratios, the NO_x emissions performance of the stock engine was matched while showing a 6% reduction in brake specific fuel consumption. Hydrocarbon emissions exceeded the stock values in all cases. Diagnostic data indicate that lean performance in the engine configuration tested is limited by ignition delay, cycle-to-cycle pressure variations, and cylinder-to-cylinder distribution.

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FIGURES

1. INTRODUCTION

In recent years, the need to reduce automobile exhaust emissions to meet more stringent U.S. Government emissions standards has been joined by the equally important challenge to reduce automobile fuel consumption as a means of easing the energy crisis. This has led to renewed interest in lean burn engine technology as a near-term means for making progress toward achieving these goals. This effort was directed toward an evaluation of the performance potential of the lean burn concept in the areas of fuel economy and exhaust emissions. Only concepts which are practical for near-term application to conventional IC engines using gasoline as a fuel have been considered.

Lean burn operation offers the potential for improved fuel economy by increasing engine thermal efficiency as shown in Fig. 1, which includes curves for three effective combustion intervals. Effective combustion interval is defined as the period from the first significant combustion pressure rise to the time when peak cylinder pressure is reached. It is observed that higher thermal efficiency is achieved with the shorter combustion interval. If it were possible to maintain a constant combustion interval while lowering the equivalence ratio, then significant improvements in engine thermal efficiency could be achieved until the lean flammability limit of the fuel is reached. In most conventional engines, thermal efficiency begins to decrease rapidly well before reaching the lean flammability limit of the fuel because of reduced flame speed. The fuel economy benefits of lean operation in a particular engine configuration are limited by the equipment lean limit for the fuel being used. To achieve the potential of lean burn, it is necessary that the engine maintain a fast-burning charge for lean equivalence ratios.

The lean burn concept also offers potential for controlling NO_x emissions as shown in Fig. 2. This analytical curve indicates that NO_x emissions reach a peak for an equivalence ratio of 0.85; however, significant reductions in emissions can be achieved for leaner equivalence ratios. The goal of simultaneously achieving increased fuel economy and reduced exhaust emissions from lean burn operation depends on the success of making engine modifications

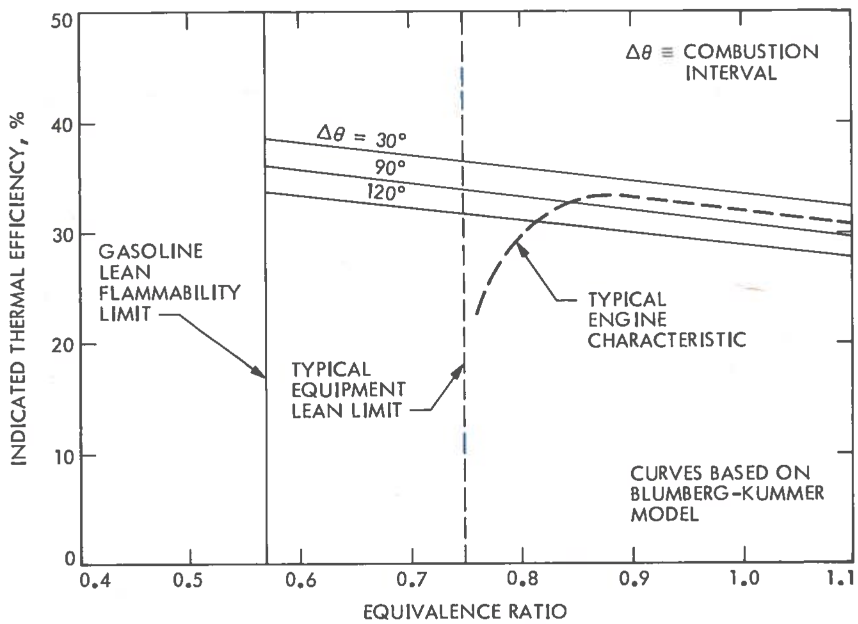


FIGURE 1. THERMAL EFFICIENCY VERSUS EQUIVALENCE RATIO

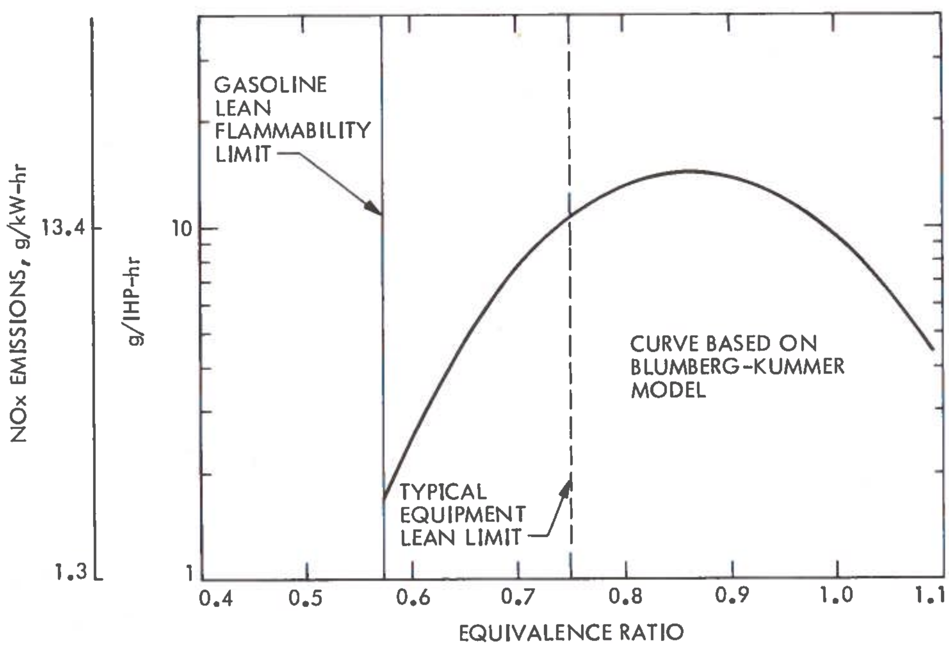


FIGURE 2. NO_x EMISSIONS VERSUS EQUIVALENCE RATIO

which permit the efficient use of gasoline at equivalence ratios approaching the lean flammability limit of gasoline.

For a given fuel it has generally been established that mixture, turbulence, and spark contribute toward extension of the equipment lean limit.[1,2] The relative importance of these factors, however, may vary from engine to engine. Because of the complex interactions of flame initiation, turbulence, and mixture homogeneity, it is seldom possible to predict the performance improvement which will result from a single change to a particular engine; however, certain general characteristics have been found to be important for good lean burn operation.

Since a lean burn engine must operate efficiently near the lean flammability limit of the fuel to achieve the fuel economy and emissions benefits of lean burn, it is desirable that it have a uniform, homogeneous fuel/air mixture equally distributed to each cylinder with a minimum of cycle-to-cycle variations. Sonic carburetors, fuel atomizers, heated manifolds, large mixing volumes, and various turbulence devices have demonstrated the ability to improve mixture distribution; however, many of these devices also have an associated volumetric efficiency penalty. Several induction systems for improving mixture distribution and homogeneity which are currently in various stages of development are the Shell Vapipe,[3] Ethyl Corporation lean reactor,[4] Dresserator,[5] and Autotronics[4] systems.

Ignition system characteristics become more critical with lean operation. Lean mixture ignition is improved with a long-duration, high-energy spark discharging through an extended-reach spark plug having a wide spark gap.[5] For successful ignition to occur it is necessary that the spark kernel, which is established by the spark discharge, interact with the local flow field and develop into a fully turbulent flame front. This flame front development can be inhibited or completely quenched by excessively high or irregular turbulence. For this reason, small-scale, uniform turbulence is desirable at the spark plug.

Once the flame front is established, uniform turbulence of relatively high intensity is needed in the combustion chamber to produce a fast flame speed through the lean mixture. Combustion chamber turbulence is difficult to control since it consists of the superimposed flow fields produced by intake charge flow, piston movement, and combustion chamber shape. This is further complicated by the different kinds of turbulence needed for the development and propagation of a flame front in lean mixtures. Turbulator intake valves[6,7] and the Heron or bowl combustion chamber shapes[7] have been reasonably successful in the lean mixture application.

The evaluation of the lean burn concept reported in this work is based on engine dynamometer tests of stock and lean burn versions of a 1973 Chevrolet V-8. The lean burn version incorporated modifications to improve its ability to burn lean mixtures. Comparisons are made of the fuel consumption and emissions characteristics of the two engines. Results of diagnostic measurements made on the lean burn engine are used to evaluate cylinder-to-cylinder mixture distribution, ignition delay, combustion duration, and cycle-to-cycle pressure variations.

Fuel consumption and emissions data for the stock vehicle over the Federal Driving Cycle (FDC) are discussed. Estimates of FDC performance of a vehicle having the lean burn engine are made using a computer simulation model.

2. EXPERIMENTAL EQUIPMENT AND PROCEDURE

2.1 Engine Descriptions

Tests were made on stock and lean burn versions of a 1973 350 CID (5.7 liter) Chevrolet V-8 using a water brake engine dynamometer. This engine was selected as being representative of a high production engine in general public use for passenger vehicles.

The stock engine has a compression ratio of 8.5 and a 4-bbl carburetor. The ignition system is the standard breaker point type consisting of a coil, condenser, distributor, wiring, and spark plugs. The spark advance characteristic is determined by the 12° BTDC basic timing and the contribution of the centrifugal and vacuum advance mechanisms. Factory-installed devices for emission control include an air injection reactor (AIR) pump, an exhaust gas recirculation (EGR) system, and a positive crankcase ventilation (PCV) system. EGR flow rate is controlled by balancing intake manifold vacuum against the spring tension of the EGR valve. The maximum design EGR flow rate is 8-12% for the Chevrolet V-8.

Several engine modifications were made for the engine dynamometer test configuration to improve the experimental setup and to permit the taking of additional diagnostic data. A pressurized fuel tank was used to improve the accuracy of the gasoline flow rate measurement. The stock dual exhaust manifolds were replaced with special exhaust headers for exhaust gas sampling of individual cylinders. The AIR pump was disconnected for the baseline tests since its use was not compatible with the special exhaust headers.

The lean burn engine configuration was established by dynamometer tests to evaluate the effect of each engine modification on lean performance. The stock carburetor was replaced with an Autotronics induction system which has an air-driven atomizer to promote better gasoline atomization. A positive displacement electric fuel pump supplied by a pressurized fuel tank was used to improve gasoline control.

A single-plane manifold was used to distribute the mixture to the cylinders. The intake manifold was modified by blocking the exhaust hot spot crossover to prevent mixing of individual cylinder exhaust gases and to permit exhaust gas sampling of individual cylinders. Heat for the hot spot was supplied by flowing engine coolant through the crossover passage.

Turbulator intake valves similar to those used successfully by Gabele[6] were incorporated into high compression (9.5 compression ratio) slant plug heads to increase the turbulence in the combustion chamber. The intake valves, shown in Fig. 3, were designed to minimize losses in volumetric efficiency. The slant plug heads have a modified combustion chamber shape to promote turbulence and have an improved spark plug location.

The GM High Energy Ignition (HEI) system was used to provide a higher energy discharge for the ignition of lean mixtures. In preliminary tests the HEI system was shown to provide better lean performance than a multiple spark, capacitive discharge ignition system. Both the centrifugal and vacuum advance features of the distributor were removed. Extended reach spark plugs of a cold heat range (AC plug type R43LTS) with an 0.080 in. (2.03 mm) gap setting were found to give the best performance with the slant plug heads.

Since the purpose of this effort was to determine the benefits of lean burn operation, the lean burn engine did not incorporate an air injection reactor pump or exhaust gas recirculation system for emission control. It did retain the positive crankcase ventilation system.

2.2 Instrumentation

The instrumentation capability is based on a digital data acquisition system which is sufficiently large to permit real-time output of data in engineering units and of calculated parameters such as equivalence ratio and thermal efficiency. Data acquisition rate is 500 measurements per second and 20 seconds are required to print data for 100 channels.

Pressure, temperature, and flow rate measurements use routine technology with gasoline flow rate being measured with turbine flowmeters. Air

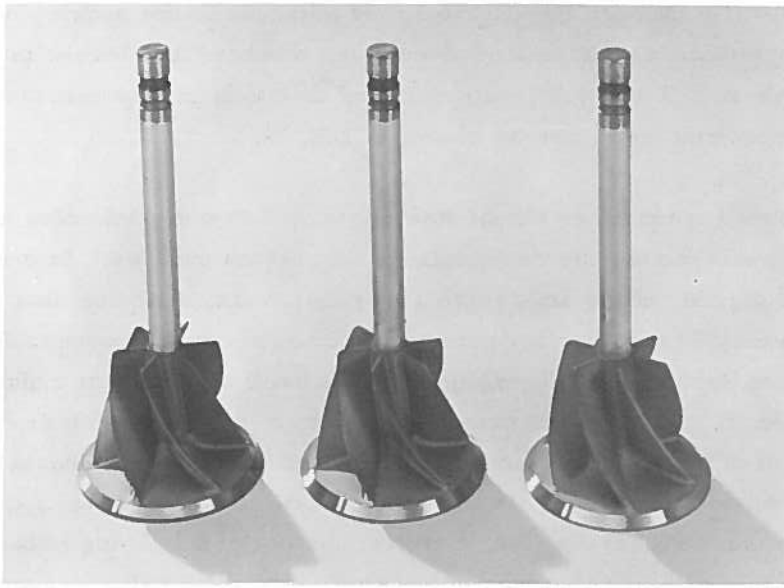


FIGURE 3. TURBULATOR INTAKE VALVES

flow rate is measured using a laminar flow transducer. Temperatures and pressures are sensed to permit real-time computation of all mass flow rates.

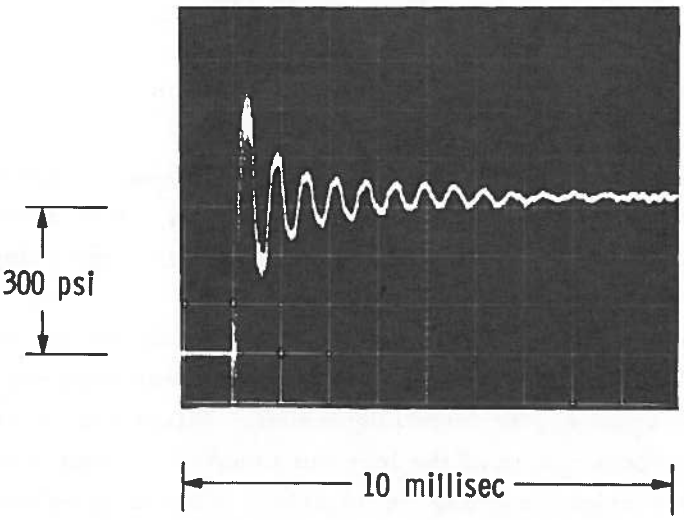
An engine exhaust gas sample was continuously filtered, passed through moisture condensing coils, and then supplied to a flow bench analyzer where emissions were measured using conventional instruments. The carbon dioxide and carbon monoxide analyzers operate on the principle of infrared absorption. Oxides of nitrogen are measured with a chemiluminescent instrument and are reported as NO_2. Unburned hydrocarbons are measured with a flame ionization detection system. The flow bench also includes an oxygen sensor, which, when combined with the other exhaust composition measurements, permits a calculation of equivalence ratio based on exhaust gas analysis. The exhaust gas sample was taken from the common exhaust line and from the individual cylinder exhaust lines of the special exhaust headers. This permits a determination of cylinder-to-cylinder distribution as well as an overall measurement of equivalence ratio and exhaust emissions.

A high-response pressure measurement system was implemented on the V-8 engine to obtain individual cylinder pressure-time data. Pressure taps were drilled directly through the engine heads adjacent to the spark plug locations. A piezoelectric pressure transducer was mounted as closely as possible to the cylinder head. Shock tube tests verified that this pressure measurement had adequate frequency response as shown in Fig. 4.

The pressure transducer signal was converted to a digital value at 1-deg crankshaft intervals during the compression and power strokes. In order to attain accurate digital values at precise 1-deg intervals, a timing disc with holes located every 10 deg was attached to the crankshaft as shown in Fig. 5. Electrical pulses derived from these holes were used as inputs to a phase-locked loop circuit. This circuit provided an output pulse at 1-deg intervals with a maximum of ±0.1 deg deviation from the nominal value. These 1-deg pulses were used as control signals to a circuit that held the value of the analog combustion pressure signal constant from the instant of the 1-deg pulse until the conversion to a digital value was completed. A digital value proportional to the combustion pressure with 0.1% resolution was thus obtained at precise 1-deg crankshaft intervals.

The 360 pressure measurements for one cycle were temporarily stored, then transferred to a permanent location through an averaging program. The time required for the averaging computation dictated that alternate cycles were measured. The acquisition-averaging sequence was continued until averaged data for a specified number of cycles was stored in the permanent location. The averaged motoring and firing data were based on 10 and 100 cycles respectively. Motoring pressure data were obtained by electrically shorting the spark to the cylinder whose pressure was being monitored while maintaining engine RPM and load with the remaining 7 cylinders. The average motoring data, the average firing data, and the difference between these two conditions were then plotted versus crankshaft angle. A mark defining the spark time within ±0.2 deg was also included.

To obtain an indication of cycle-to-cycle pressure variations, photographs were made of multiple pressure traces using a storage oscilloscope. Approximately 10 successive cycles were recorded for both the motoring and firing modes.



RISETIME: 300 psi IN 100 μsec

NATURAL FREQUENCY: 1.7 kHz

FIGURE 4. TRANSDUCER RESPONSE FROM SHOCK TUBE TESTS

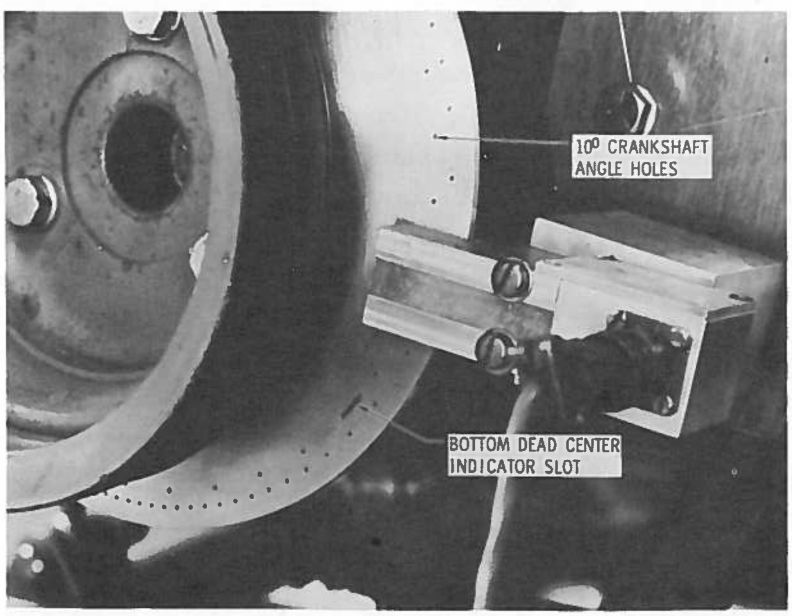


FIGURE 5. PHOTOGRAPH OF TIMING DISC MOUNTED ON ENGINE

3. EXPERIMENTAL RESULTS

3.1 Economy-Emissions Tradeoffs

Sensitivity tests were conducted on an engine dynamometer to determine the effects of equivalence ratio and exhaust emissions. Fuel consumption data from these tests are shown in Fig. 6 for a typical engine operating condition.

For comparison purposes, the corresponding data for the stock engine, tuned to factory specifications, is also shown. The lean burn engine required less fuel than the stock engine for all equivalence ratios and spark advances tested. The fuel consumption of the lean burn engine decreases as the equivalence ratio is decreased, reaching its minimum value at an equivalence ratio of 0.75 and a spark advance of 60° BTDC. Further decreases in equivalence ratio result in a loss of fuel economy. As the equivalence ratio decreases, more spark advance is required to achieve minimum fuel consumption which indicates that the combustion interval and/or ignition delay are increasing as the equivalence ratio is decreased.

The sensitivity of NO_x emissions to variations in equivalence ratio and spark advance is illustrated in Fig. 7. For a given spark advance, decreasing the equivalence ratio is effective in reducing NO_x emissions since it results in lower peak temperatures and less time at high temperatures because of the longer combustion duration and/or ignition delay. Similarly, for a given equivalence ratio, retarding the spark timing is also shown to be an effective means for reducing NO_x emissions since it also leads to reduced peak temperatures and less time at high temperatures. The lean burn engine produces NO_x emissions equal to or less than those for the stock engine for several combinations of equivalence ratio and spark advance; however, for the minimum fuel consumption setting, the NO_x emissions are higher than the stock values.

A representation of the tradeoff between fuel consumption and NO_x emissions is illustrated in Fig. 8. For the range of equivalence ratios and spark advances of the test data, a single curve represents the relationship between

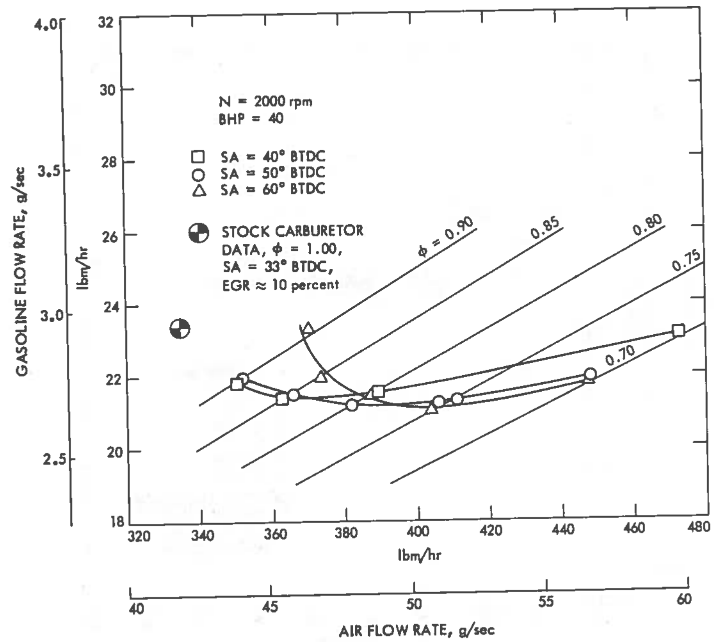


FIGURE 6. FUEL CONSUMPTION CHARACTERISTIC FOR LEAN BURN ENGINE

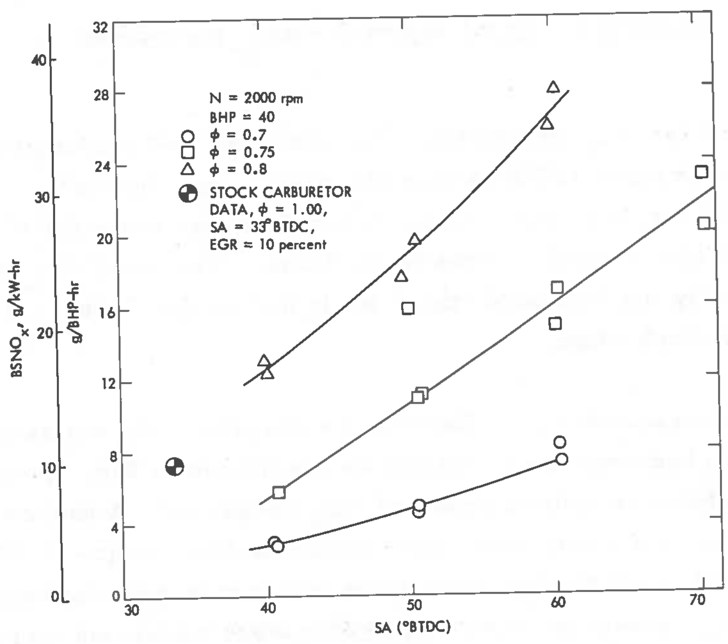


FIGURE 7. $BSNO_x$ EMISSIONS VERSUS SPARK ADVANCE

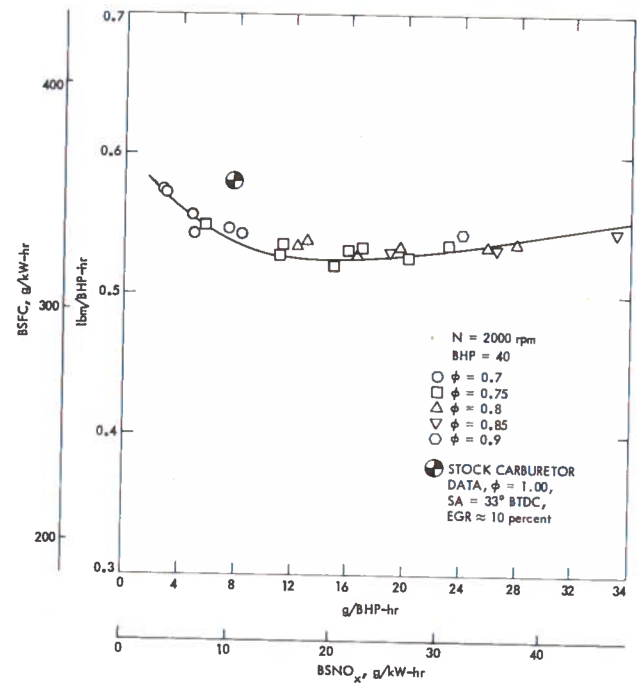


FIGURE 8. BSFC VERSUS $BSNO_x$ EMISSIONS

fuel consumption and NO_x emissions. The minimum fuel consumption for the lean burn engine is about 10% less than the stock value; however, the NO_x emissions at this condition are a factor of 2 higher than those for the stock engine with emission control devices in operation. The stock NO_x emissions level can be met by the lean burn engine while maintaining a fuel consumption 6% less than the stock value.

Although lean operation is effective in controlling NO_x emissions, it can lead to increased hydrocarbon emissions as illustrated in Fig. 9, which shows the relationship between hydrocarbon and NO_x emissions. A dashed line is drawn through the data points which have minimum best torque (MBT) spark timing. For MBT spark timing, reductions in equivalence ratio below 0.9 produce less NO_x emissions; however, hydrocarbon emissions increase rapidly for equivalence ratios less than about 0.8. When the lean burn engine is tuned

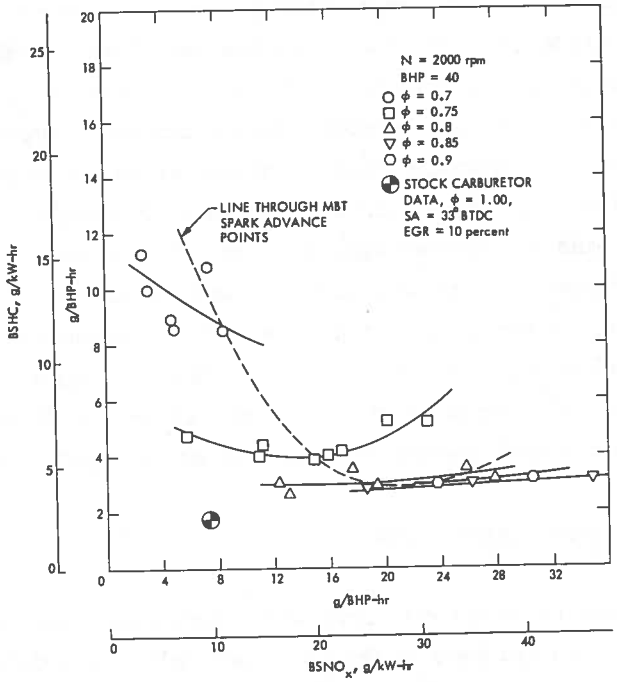


FIGURE 9. BSHC EMISSIONS VERSUS $BSNO_x$ EMISSIONS

to produce NO_x emissions equal to the stock values, the hydrocarbon emissions are much higher than those of the stock engine.

Although retarding the spark timing from its MBT value reduces NO_x emissions, it does not reduce hydrocarbon emissions and, in fact, causes a slight increase in hydrocarbon emissions for the leaner equivalence ratios. This effect is different from that in engines running rich, where retarding the spark timing decreases hydrocarbon emissions by increasing exhaust temperatures which promote further hydrocarbon reaction in the exhaust system. At leaner equivalence ratios, retarding the spark timing leads to an increase in unburned hydrocarbons in the combustion chamber because of lowered peak temperatures. In addition, exhaust temperatures do not appear to be increased sufficiently to promote further hydrocarbon reactions in the exhaust system.

Exhaust temperatures provide a good indicator for hydrocarbon emissions. Individual cylinder exhaust temperatures were measured near the exhaust valves for all tests of the lean burn engine. The average exhaust temperature was computed as the average of the 8 individual exhaust temperatures. Hydrocarbon emissions are shown plotted as a function of average exhaust temperature in Fig. 10. For average exhaust temperatures less than about 1200°F (920°K), hydrocarbon emissions increase sharply. A similar trend exists when hydrocarbon emissions are plotted against minimum individual cylinder exhaust temperature with the sharp increase in emissions occurring at about 1140°F (890°K) in this case. Although the lean burn engine does not reduce hydrocarbon emissions, tests of similar lean burn engines using hydrogen-enriched gasoline[8] have shown that exhaust temperatures are still high enough to permit control of hydrocarbon emissions with the use of an oxidizing catalytic exhaust converter.

3.2 Equivalence Ratio Distribution

The equivalence ratio distributions of the stock and lean burn engines have been measured using an analysis of the composition of the individual cylinder exhausts. The fuel atomizer, which was located directly above the engine intake manifold, was adjusted to give the best possible distribution with the lean burn engine.

The cylinder-to-cylinder equivalence ratio variations of the stock and lean burn engines are given in Fig. 11. A convenient measure of the distribution is the standard deviation of the equivalence ratios for all eight cylinders. The lean burn engine has a smaller standard deviation, which indicates a better cylinder-to-cylinder mixture distribution; however, the maximum spread in equivalence ratio between the richest and leanest cylinders is slightly larger for the lean burn engine.

The stock distribution has an interesting pattern, with 4 cylinders being about 4% richer than average and 4 cylinders being about 4% leaner. This imbalance correlates with the right and left front barrels of the stock carburetor. The relatively good distribution measured with the carburetor may be associated with the choice of test condition, since other investigators[2] have found that carburetors give their best distribution near 2000 rpm and level-road-load power.

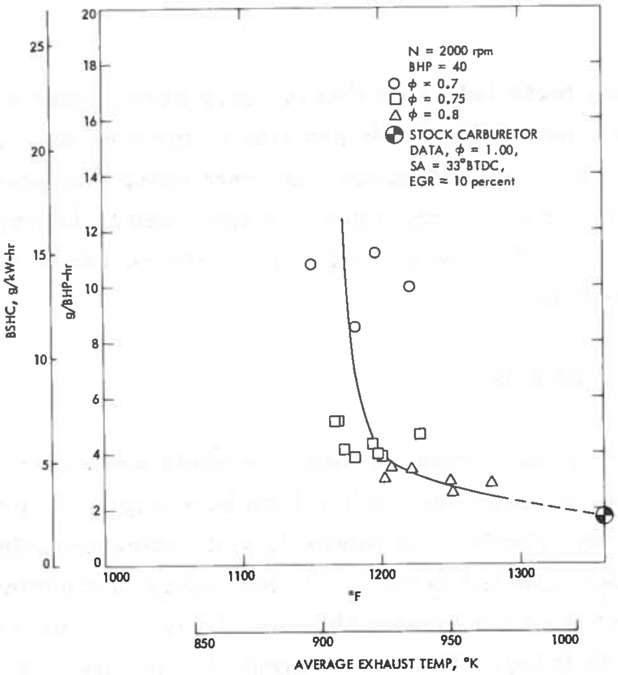


FIGURE 10. BSHC EMISSIONS VERSUS AVERAGE EXHAUST TEMPERATURE

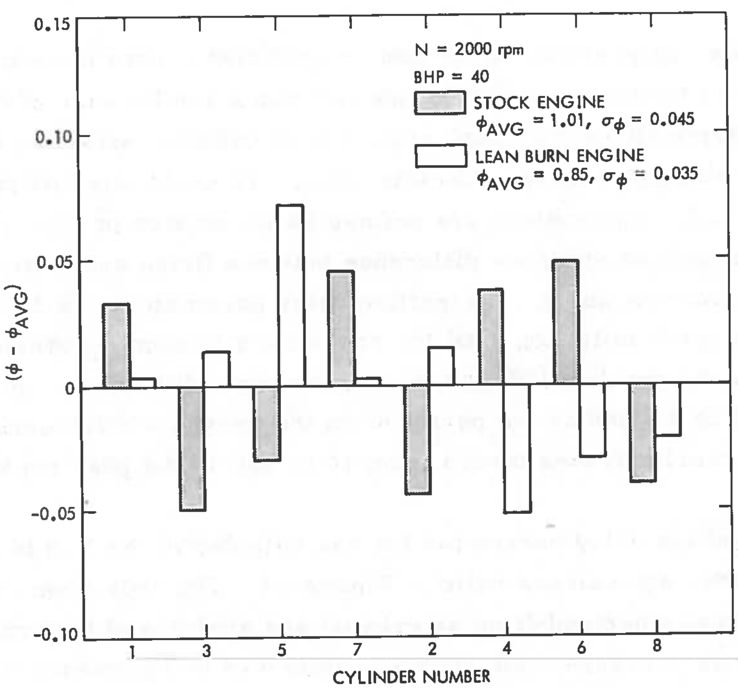


FIGURE 11. CYLINDER-TO-CYLINDER EQUIVALENCE RATIO DISTRIBUTION

Preliminary tests indicated that the lean burn engine equivalence ratio distribution was sensitive to the orientation of the fuel atomizer relative to the intake mainfold. During subsequent flow tests using compressed air and water, the spray pattern of the fuel atomizer was observed to be asymmetric. This asymmetry is reflected in the spread in equivalence ratios observed in the lean burn engine distribution.

3.3 Combustion Effects

The high-response pressure measurements were used to gain understanding of the combustion processes in the lean burn engine by providing information about ignition delay, combustion interval, and cycle-to-cycle pressure variations. Combustion interval is normally defined as that period from spark initiation to the time of the first measurable rise in cylinder pressure above the motoring pressure trace. This corresponds to the time required for transition from a spark kernel to a developed flame front. Thus the effective combustion interval is the combustion interval minus the ignition delay period.

In analyzing pressure-time data, significant errors in estimating ignition delay and the effective combustion interval can arise because of the difficulty of accurately determining the crank angle where cylinder pressure first rises above the motoring pressure characteristic. To avoid this difficulty, two additional combustion parameters are defined as illustrated in Fig. 12, which shows a plot of normalized pressure difference between firing and motoring pressure traces versus crank angle. An ignition delay parameter α is defined as the period from spark initiation until the pressure difference between firing and motoring traces reaches 10% of the peak pressure difference. A flame speed parameter β is defined as the period when the pressure difference between firing and motoring traces moves from 10 to 95% of the peak pressure difference.

The ignition delay parameter for one cylinder of the V-8 is shown plotted versus cylinder equivalence ratio in Figure 13. The individual cylinder equivalence ratio was derived from an exhaust gas analysis of the exhaust from the cylinder whose pressure-time characteristic was being measured. Note that data for the 4 run conditions are at MBT spark timing. The ignition delay parameter is seen to increase significantly as the equivalence ratio is decreased.

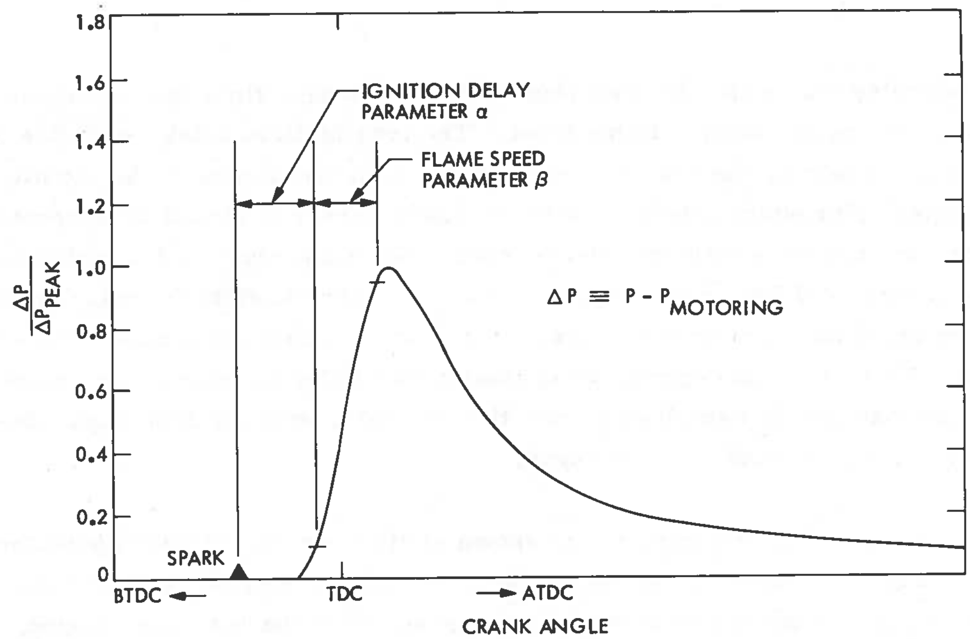


FIGURE 12. DEFINITION OF IGNITION DELAY AND FLAME SPEED PARAMETERS

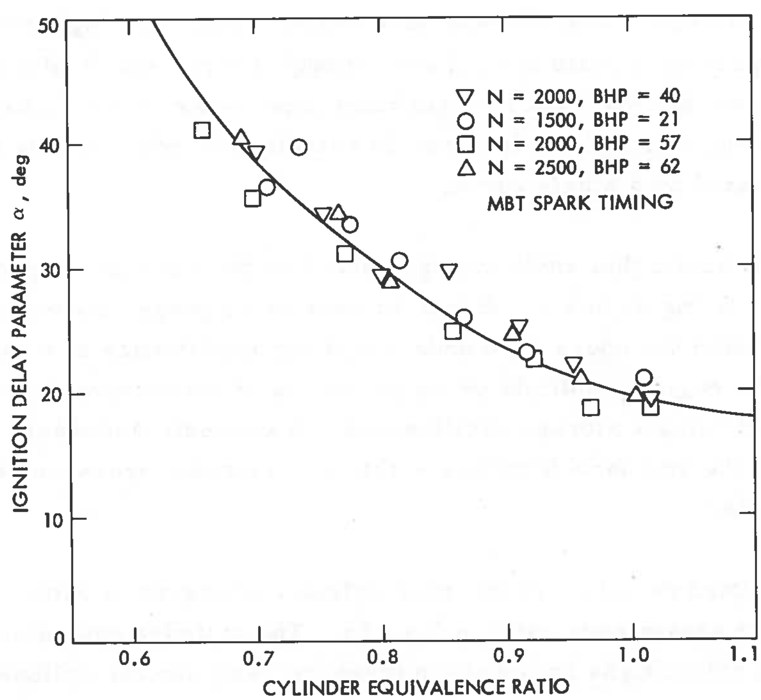


FIGURE 13. IGNITION DELAY PARAMETER VERSUS EQUIVALENCE RATIO FOR SINGLE CYLINDER

This indicates that lean mixtures require a much longer time than stoichiometric mixtures for establishing a flame front. The long ignition delays with this lean burn engine could be the result of excessively high turbulence in the vicinity of the spark plug which interacts with the spark kernel to inhibit development, quench, or produce a distorted flame front. The slant plug heads, turbulator intake valves, and long-reach spark plugs which were used to increase burning velocity could have resulted in the spark gap being located in a high turbulence region. No significant dependence of the ignition delay parameter on engine RPM and load can be established from the test data, with all data being adequately correlated with a single curve.

The flame speed parameter is shown plotted versus cylinder equivalence ratio in Fig. 14. Note that an increase in the flame speed parameter reflects a slower burning charge or slower flame speed. For the lean burn engine, the flame speed parameter increases only slightly as the equivalence ratio is decreased. This indicates that once a flame front is established, the lean mixture is being burned almost as fast as a stoichiometric mixture. This flat characteristic could explain why the lean burn engine maintained high thermal efficiency at an equivalence ratio of 0.75 even though a large spark advance was required. From this test data no significant dependence of the flame speed parameter on engine RPM and load can be established, with all data being adequately correlated by a single curve.

It is well known that small changes occur in the combustion process during successive firing cycles which lead to fluctuating pressure-time characteristics of individual cylinders. To understand the significance of these variations in the lean burn engine, multiple pressure traces of successive firing cycles were compared using a storage oscilloscope. A convenient measure of these fluctuations is the standard deviation of the peak cylinder pressures of successive firing cycles.

The standard deviation of the peak cylinder pressure is shown plotted versus cylinder equivalence ratio in Fig. 15. The cylinder equivalence ratios were based on exhaust gas composition measurements for the cylinder whose pressure was being measured. For the range of conditions tested, cycle-to-cycle peak pressure variations remained fairly constant for equivalence ratios

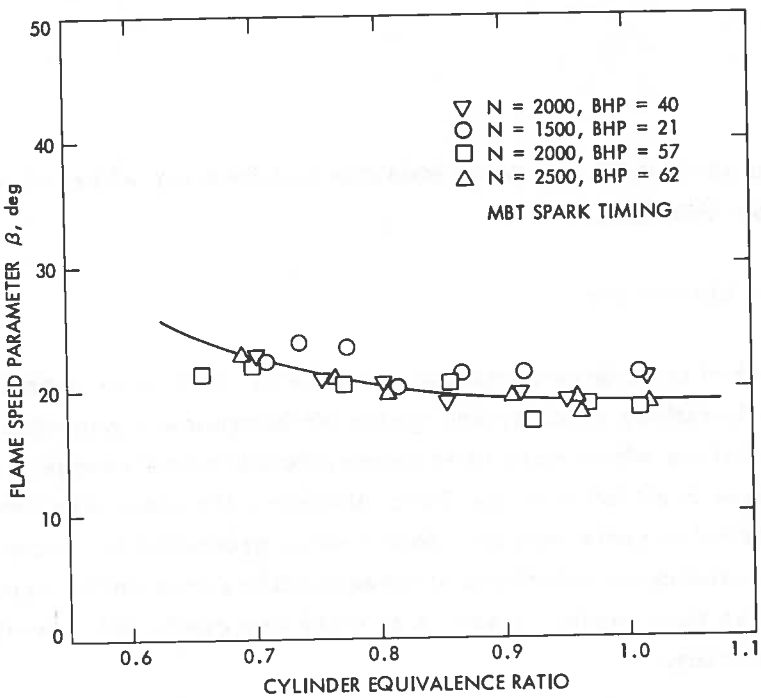


FIGURE 14. FLAME SPEED PARAMETER VERSUS EQUIVALENCE RATIO FOR SINGLE CYLINDER

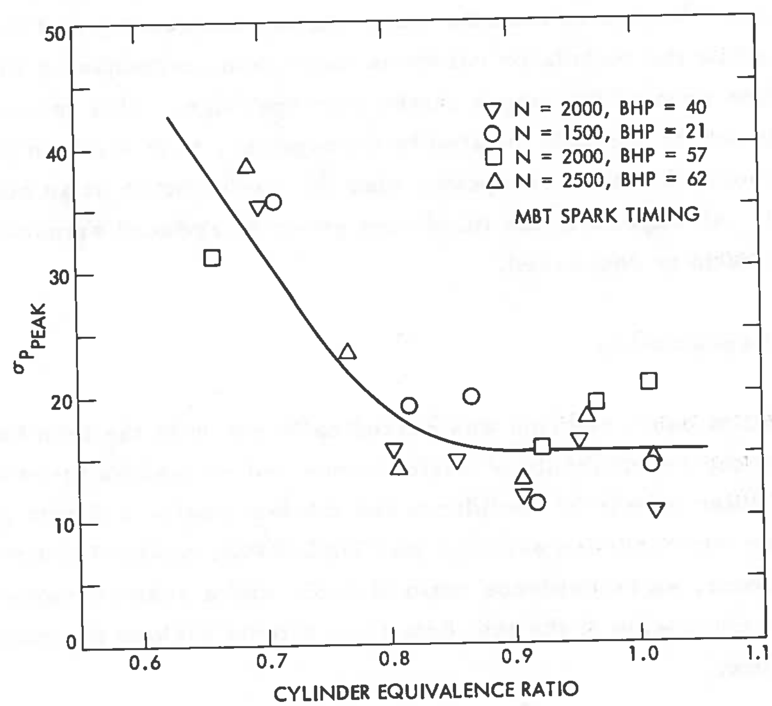


FIGURE 15. PEAK CYLINDER PRESSURE STANDARD DEVIATION VERSUS EQUIVALENCE RATIO

greater than about 0.75. It is not possible to detect any effect of engine speed and load from this data.

3.4 Power Limitations

Several of the engine modifications used in the lean burn engine affect the engine's air breathing capacity and maximum horsepower potential. The turbulator intake valves which were used to provide increased charge turbulence introduce some restriction to air flow; however, the slant plug heads have larger intake valve ports than the stock heads, producing the opposite effect. To determine the combined effect of these modifications on the breathing capacity of the lean burn engine, a series of tests was conducted at wide-open-throttle conditions.

A comparison of the maximum air flow characteristics of the stock and lean burn engines is shown in Fig. 16. The breathing capacity of the lean burn engine is about 5% greater than the stock engine, suggesting that the flow restriction produced by the turbulator valves is more than compensated for by the increased flow area of the larger intake port openings. This increase in breathing capacity can be directly related to horsepower, with the lean burn engine producing about 5% more horsepower than the stock engine at an equivalence ratio of 1.0. As expected, the maximum power is reduced significantly as the equivalence ratio is decreased.

3.5 Data Repeatability

A baseline test condition was periodically run with the lean burn engine to determine the repeatability of performance and emissions measurements and thereby establish a level of confidence for the test engine and data acquisition system. The run condition selected was 2000-RPM, 40-BHP (29.8-kW), level-road-load power, an equivalence ratio of 0.85, and a spark advance of 44° BTDC. A statistical evaluation of the test data from 8 tests yielded the results shown in the table below.

TABLE. STATISTICAL ANALYSIS OF BASELINE DATA

Parameter	Confidence Range at 95% Probability	Percent Deviation at 95% Probability $1.86\ \sigma/\bar{x} \times 100$
RPM	1984 - 2038	1.32
BMEP, psi	44.8 - 45.7	1.03
Φ	0.848 - 0.854	0.33
η_{t_I}	0.360 - 0.366	0.87
NO_x, ppm	2644 - 3624	15.6
HC, ppm	1182 - 1382	17.8
CO, %	0.077 - 0.103	14.5

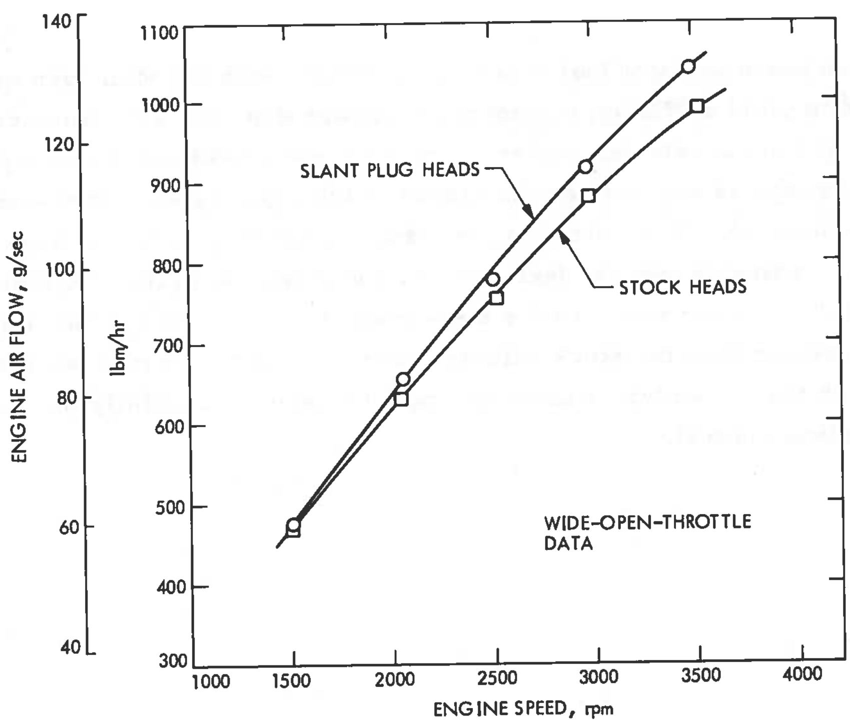


FIGURE 16. EFFECT OF SLANT PLUG HEADS ON WIDE-OPEN-THROTTLE AIR FLOW

Based on a 95% confidence level, the percent deviation in indicated thermal efficiency is ±1%, while the corresponding deviation in the exhaust emissions data is about 15%. These variations are due to a combination of engine setting variations and measurement repeatability.

3.6 Driving Cycle Predictions

Since no vehicle tests were made with the lean burn engine, a computer simulation model of the Federal Driving Cycle (FDC) was used to predict the performance over the cycle. The computer program used fuel consumption and emissions maps obtained from the engine dynamometer data previously discussed. The driving cycle simulation model was used successfully to predict performance of the stock vehicle for which both engine dynamometer data and FDC results were available. All calculations were made for a 4500 lbm (2040 kg) Chevrolet Impala with Turbo-Hydromatic transmission and a 2.73 rear axle ratio.

When tuned for best fuel economy, a vehicle with the lean burn engine is estimated to yield a 22% improvement in mileage over the FDC when compared with the 1973 stock vehicle; however, under these conditions the hydrocarbon and NO_x emissions are greater than those of the stock vehicle with emissions control equipment. When tuned to meet a 2 gm/mi NO_x emission level over the FDC, it is estimated that the lean burn configuration will give 12% more mileage than the stock version. Under these conditions the hydrocarbon emissions are still greater than the stock values; however, exhaust temperatures are high enough that a catalytic converter could be used successfully for hydrocarbon emissions control.

4. CONCLUSIONS

The conclusions of the study are as follows:

1) It has been successfully demonstrated that a lean burn engine tuned for peak thermal efficiency can provide significant improvements in fuel economy over the stock engine; however, this gain is made at the expense of higher NO_x and hydrocarbon emissions.
2) The lean burn engine can be tuned to match or better the NO_x emissions performance of the stock engine with some reduction in the fuel economy gains over the stock engine.
3) The hydrocarbon emissions of the lean burn engine exceed those of the stock engine with emissions control equipment; however, exhaust temperatures are adequate to provide hydrocarbon oxidation in a catalytic converter.
4) Diagnostic data indicate that the performance of the lean burn engine is being limited by problems of ignition delay, cycle-to-cycle pressure variations, and cylinder-to-cylinder distribution. Improvements in these areas should lead to significantly better lean performance.

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