EPA/AA/TDG/92-06

Technical Report

Conversion of Methanol-Fueled 16-Valve, 4-Cylinder Engine To Operation On Gaseous $2H_2/CO$ Fuel - Interim Report IV

by

Ronald M. Schaefer
Fakhri J. Hamady
James C. Martin

September 1992

NOTICE

U. S. Environmental Protection Agency
Office of Air and Radiation
Office of Mobile Sources
Regulatory Programs and Technology Division
Technology Development Group
2565 Plymouth Road
Ann Arbor, MI 48105

UNITED STATES ENVIRONMENTAL PROTECTION AGENCY
ANN ARBOR, MICHIGAN 48105

JAN 8 1993

OFFICE OF
AIR AND RADIATION

MEMORANDUM

SUBJECT: Exemption From Peer and Administrative Review

FROM: Karl H. Hellman, Chief KH
Technology Development Group

TO: Charles L. Gray, Jr., Director
Regulatory Programs and Technology Division

The attached report entitled "Conversion of Methanol-Fueled 16-Valve, 4-Cylinder Engine To Operation On Gaseous $2H_2/CO$ Fuel - Interim Report IV," (EPA/AA/TDG/92-06) describes progress to date on a project to convert a Nissan CA18DE engine previously modified for operation on M100 neat methanol to operation on dissociated methanol ($2H_2/CO$) gaseous fuel. This engine was operated on both M100 and simulated dissociated methanol (67 percent hydrogen and 33 percent carbon monoxide) fuels. This report describes recent modifications made to the engine and fuel delivery system and summarizes the results from recent testing.

Since this report is concerned only with the presentation of data and its analysis, and does not involve matters of policy or regulations, your concurrence is requested to waive administrative review according to the policy outlined in your directive of April 22, 1982.

Concurrence:

Charles L. Gray, Jr., Director, RPT — Date: 1-5-93

cc: E. Burger, RPT

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I. Summary

A 16-valve, 4-cylinder light-duty automotive engine has been converted to operation on a mixture of hydrogen (H_2) and carbon monoxide (CO) gaseous fuel in a 2:1 molar ratio of H_2 and CO. This engine has been used to investigate the difference in emission levels and power output between two different fuels: M100 neat methanol and simulated dissociated methanol gaseous fuel ($2H_2/CO$).

This report contains test results of power output from several engine/fuel system modifications. These modifications were made in an attempt to increase the brake specific power output of the test engine when operated on $2H_2/CO$ fuel in reference to M100 levels. With the gaseous fuel, several intake port/fuel system configurations and fuel injection nozzle designs were evaluated. These modifications were evaluated to determine the effects of fuel pressure, injection location, and fuel delivery methods on engine performance.

Gaseous fuel was delivered to the engine by two different methods, premixed air/fuel mixture in the intake manifold from continuous port injectors (simulated carbureted operation) and simulated direct injection. The simulated direct injection operation was achieved by inserting blockages in four of the intake runners where the M100 fuel injectors were located. As a result, gaseous fuel was supplied through one intake valve and air through the second intake valve.

The largest torque achieved in this evaluation with $2H_2/CO$ fuel was 80 ft-lbs, approximately 80 percent of the maximum torque levels obtained with M100 fuel at the same WOT, 2,000 rpm operating conditions. This torque was obtained in the simulated direct injection configuration. When operating with the premixed mixture, output torque reached approximately 60 ft-lbs with frequent abnormal combustion occurrences, most notably preignition in the intake manifold. Preignition of a hydrogen containing fuel is not an unusual occurrence.[1-5]

Brake-specific emission levels with gaseous fuel testing conducted in this most recent evaluation were significantly lower than levels obtained previously. However, engine-out CO levels were approximately double the level obtained with M100 fuel. HC levels with $2H_2/CO$ fuel operation were very low.

Proposed future efforts will utilize a mixture of M100 and $2H_2/CO$ fuel for engine operation. Emissions, fuel consumption, and engine performance will continue to be monitored for operation on this mixed fuel.

II. Introduction

With recent advances in internal combustion engines and emission control development, new technologies are being directed toward improving combustion efficiency, multi-fuel capability, and reduced emissions. Recent developments in engine technology have enhanced stable burn and reduced emission levels. The use of various alternative fuels is also being addressed to satisfy clean air legislation for the 1990's.

One alternative fuel candidate is the concept of using exhaust waste heat to provide the energy necessary for the dissociation of methanol (CH_3OH) to hydrogen and carbon monoxide. Methanol may be catalytically decomposed to H_2 and CO gases according to the reaction:

$$CH_3OH_{(l)} \text{ ----- } 2H_{2(g)} + CO_{(g)}$$

The decomposition of methanol to this gaseous fuel mixture has been postulated as a more efficient method of using methanol as a light-duty motor vehicle fuel. The major attraction of methanol decomposition is that the resulting gases have a higher heating value per pound than the original liquid methanol. A discussion of the application of dissociated methanol as a light-duty automotive fuel was presented in previous papers.[7,8]

In order to evaluate this concept, EPA modified a Nissan CA18DE multi-valve engine to better utilize the combustion characteristics of dissociated methanol fuel. This engine is a stock model modified by Nissan Motor Corporation for use with liquid methanol. The engine was loaned to EPA by Nissan for use in alternative fuels research. This report summarizes the most recent EPA efforts in the investigation of dissociated methanol as an automotive fuel for this engine.

The simulated dissociated methanol product gas used in this work was a mixture of H_2 and CO gases in the molar ratio $2H_2/CO$. EPA did not possess a methanol dissociation system capable of generating the necessary quantities of gaseous fuel at the time work on this project was started; the engine was therefore tested on a bottled gas mixture of $2H_2/CO$.

III. Description of Test Engine

Several modifications were made to this engine by EPA since its delivery from Nissan; these modifications were detailed in previous EPA technical reports.[6,7] This section will describe the test engine and previous modifications made for operation on $2H_2/CO$ fuel. This was the state of the test engine at the beginning of the work described in this report.

The engine used for this project was a Nissan CA18DE engine with an in-line, 4-cylinder, 1.8-liter capacity. The valve arrangement is a 4-valve per cylinder configuration, consisting of two intake and two exhaust valves per cylinder. The valves are operated by dual-overhead camshafts, one each for the intake and exhaust sides.

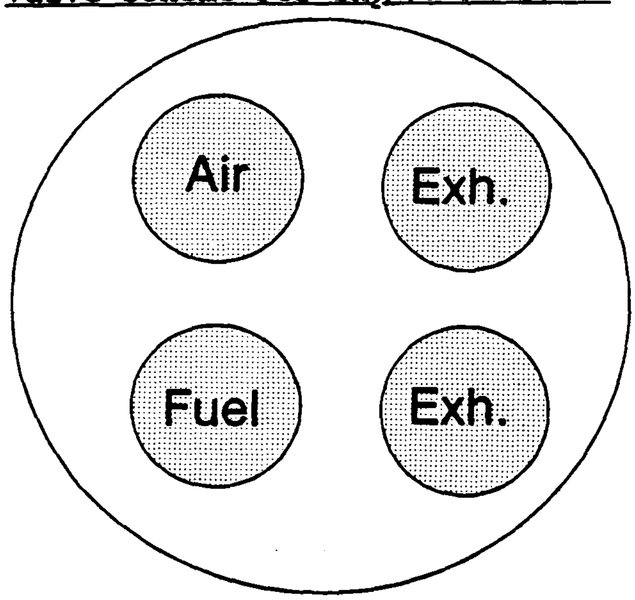
The test engine was initially modified by Nissan to better utilize the qualities of M100 neat methanol over unleaded gasoline. These modifications were discussed in detail in an earlier paper.[6] A summary of the test engine specifications when fueled with M100 neat methanol is included in this report as Appendix A.

EPA then modified this engine for use with simulated dissociated methanol fuel (67 volume percent H_2 and 33 volume percent CO). The first modification was the installation, at Nissan's request, of a thicker head gasket. This thicker gasket raised the clearance between the valve face and the piston crown; this modification was made to improve the durability of the engine. This thicker gasket lowered the compression ratio from 11.0 to 10.5.

With M100 fuel, the engine utilized a 4-valve per cylinder valvetrain configuration (two intake and two exhaust valves per cylinder). This arrangement was modified to allow for admission of air to the cylinder through one intake valve only; the second intake valve supplied the $2H_2/CO$ fuel. The exhaust side valve scheme was not altered (Figure 1). This configuration is hereafter referred to as simulated direct injection operation.

Figure 1

Simulated Direct Injection Operation
<u>Valve Scheme For $2H_2/CO$ Fuel Use</u>



This valve scheme allowed for the admission of gaseous fuel only through one of the intake valves. An intake air control assembly encloses the swirl control valves and is situated between the intake manifold and the combustion chambers on the M100-fueled engine. This assembly controls the air flow so that it is through one intake runner and/or through both intake runners as necessary. This is to control in-cylinder charge motion on the liquid-fueled engine.

When the engine was converted by EPA to operation on $2H_2/CO$ fuel, the control valve slide and actuator were disassembled and the swirl control valves removed. The runners through the valve assembly that contained holes for the fuel injectors were welded shut approximately 1/2-inch upstream from the holes. These seals prevented the admission of air to the ports through which the gaseous fuel passes. The holes in the assembly left by the power valve slide were sealed to prevent leakage of fuel and air between runners.

Fuel injectors were not used to deliver the $2H_2/CO$ fuel. The rail and the individual injectors were removed and 3/8-inch inside diameter stainless steel pipe fittings were used in their place. The stainless steel fittings were threaded and the insides of the aluminum injector holes were then threaded to adopt to these fittings.

IV. Recent Fuel System Modifications

A fuel supply cylinder outside the test cell was used with 22 feet of 1/4-inch stainless steel tubing leading from the pressure regulator on the cylinder to the solenoid valve inside the test cell. After the solenoid valve, the fuel delivery system was completely altered. The fuel line leading from the solenoid valve to a Tylan mass flow controller was 3/8-inch in diameter and measured approximately 27 feet in length. There were only slight bends and no turns in this fuel line so fewer pressure drops in the fuel delivery system would occur.

Approximately 12 inches downstream of the Tylan mass flow controller was a pressure gauge that would measure fuel pressure to the intake manifold. Ten feet downstream of the pressure gauge was another solenoid valve followed by a 2-stage hydrogen flame arrestor.

The cylindrical plenum used previously to distribute fuel flow to each cylinder was replaced with a 1/4-inch diameter fuel rail. The fuel rail would distribute the incoming fuel to each of four flexible fuel lines 3 inches in length leading to the fuel injector ports. The total length from the intake valves to the flame arrestor was approximately 11 inches.

The four fuel lines are connected to the same threaded fittings which are screwed into the fuel injection ports in the valve control assembly. The inside diameter of these fittings (previously 1/4-inch) were replaced with different nozzle designs in an attempt to increase fuel pressure and enhance the fuel distribution. Nozzle openings from 1/2 to 3 millimeters were evaluated to determine the best operating condition. The gaseous fuel is supplied to the combustion chambers by the opening of the fuel valves when the blockages in the intake air control assembly were present. With these blockages, fuel flow to the engine occurred through one intake valve and air through the other. In the remainder of this report, operation in this configuration is called simulated direct injection.

Some testing was also conducted with these blockages removed while still using the same fuel delivery fittings in the fuel injector location. Because the fuel and air were premixed in the intake manifold when operated in this configuration, this operation was called simulated carbureted operation. Different nozzle designs and fuel delivery locations within the intake manifold were also evaluated in this premixed method.

V. Exhaust Measurement Procedure

Both dilute and cylinder-out emission samples were taken during this latest phase of testing. Cylinder-out samples were raw emission levels (not diluted) and were taken for each cylinder from the runner of the exhaust manifold leading from the exhaust valves to the exhaust pipe. Dilute emission samples were engine-out levels and taken downstream of the exhaust manifold.

Dilute engine-out samples were taken as engine exhaust passes from the exhaust pipe to a 2-1/2 inch diameter flexible metal tube. This tube passes the exhaust overhead to a 6-inch rigid tube supported from the test cell ceiling. The rigid tube delivers the exhaust to a Philco Ford 350 cfm constant volume sampler (CVS). Total length of the flexible and rigid tube sections is approximately 40 feet.

A gaseous sample line has been extended through the cell ceiling and connect the mechanical CVS with an electronic display panel in the cell control room. A fitting in the sample line at the control room enables bag sampling at this point. Analysis of bag samples was accomplished at a bank of analyzers located in another test cell. Hydrocarbon (HC) emissions were measured with a Beckman Model 400 flame ionization detector (FID). NOx level determination was conducted on a Beckman model 951 chemiluminescent NOx analyzer. Carbon oxides (CO,CO_2) were measured by infrared technique using a Horiba Model A1A23 infrared analyzer.

Cylinder-out emission samples were taken for each individual cylinder from four different taps in the exhaust manifold. The sampling lines led to a Hankison Model E-4GSS compressed air dryer that cooled the emission sample through an R-12 refrigerant at a rate of 1,215 Btu/hr. The suction pressure of this unit was 33 psig. The emission sample was then collected in a bag. The total length of the sampling line from the exhaust manifold to the collection bag was approximately 8 feet.

The bag samples were then taken to another location which contained a Nicolet Rega 7000 FTIR system. This system allowed for the raw emission sample to be diluted to 10 percent with balance of nitrogen. Because the specific volume of emission sample taken was not known, brake-specific emissions could not be calculated for individual cylinders.

VI. Discussion of Test Results

A. Simulated Carbureted Operation

Simulated carbureted operation was obtained by mixing the gaseous fuel and air in the intake manifold. This premixed charge resulted from removing the blockages that were previously located in one of the intake runners for each cylinder. All of the test results presented here were obtained at steady speed (2,000 rpm) with bottled $2H_2/CO$ fuel.

The first phase of testing described here evaluated the effect of three different nozzle size openings in the stainless steel pipe fittings used to deliver the gaseous fuel. These pipe fittings, again, were located in the fuel injector holes in the intake manifold. Different nozzle designs were fitted in the pipe fittings in an attempt to increase the fuel pressure and fuel distribution in the intake system to improve torque, combustion stability, and emissions.

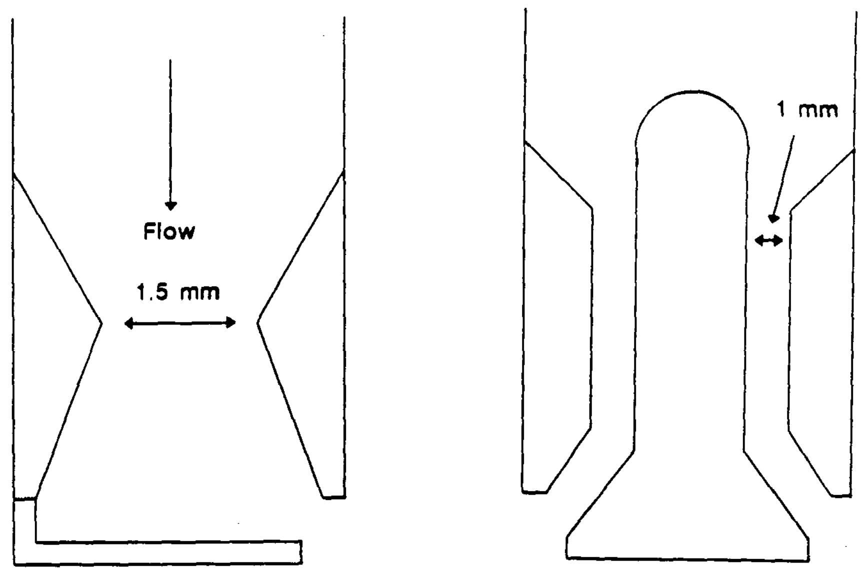
The first nozzle evaluated was 1/2-millimeter in diameter. The maximum fuel flow rate these nozzles would allow was approximately 6.7 cfm. At wide open throttle (WOT) conditions, operation here resulted in an air/fuel ratio (A/F) of 15.4. The engine would not run at this lean condition.

The second nozzle opening utilized was 1-millimeter in diameter. The maximum fuel flow rate with these nozzles was 10.6 cfm, resulting in an A/F of 9.7 at WOT, 2,000 rpm operating conditions. The maximum torque obtained during this testing was 37 ft-lbs. After approximately 1-2 minutes of engine operation, the premixed charge would begin to ignite in the intake manifold and substantially decrease output torque. This happened every time the engine was operated. The engine ran smooth until engine oil temperature reached approximately 140°F. This preignition condition at hotter engine temperatures is quite common when a premixed hydrogen containing fuel is used.[1-5]

The next nozzle investigated here utilized a 1.5-millimeter diameter opening with a diffuser. Figure 2 shows a schematic of two different nozzle designs using this size opening. With these nozzles, the engine was severely hindered by preignition in the intake manifold. This preignition occurred after approximately 1-2 minutes of operation once the engine has warmed. Also, the engine could not be operated at WOT conditions during any testing here. As the throttle was being slowly opened, there was a certain maximum airflow that would allow for smooth engine operation. When the throttle was opened above this point (increasing airflow), the engine would experience frequent preignition. The fuel delivery pressure was approximately 60 psig during this testing.

Figure 2

Simulated Carbureted Operation
Nozzle Openings With Diffusers



(a) Single-Hole Nozzle (b) 4-Hole Nozzle

Table 1 below is a summary of test results obtained with the single-hole 1.5-millimeter nozzle openings at 2,000 rpm. Spark timing for this testing was set at 10°BTDC.

Table 1

Simulated Carbureted Operation
$2H_2$/CO Fuel Operation, 2,000 RPM Conditions

Air Flow (cfm)	Fuel Flow (cfm)	A/F Ratio	Torque (ft-lbs)
38	13.9	7.4	60
25	11.7	5.8	25
20	10.3	5.3	41
25	10.3	6.6	44
32	10.3	8.4	45
35	10.3	9.2	45
15	13.9	2.9	52

The maximum torque obtained during this testing was 60 ft-lbs, approximately 60 percent of the level obtained with M100 fuel at 2,000 rpm, WOT operating conditions. The engine was also severely hindered by preignition in the intake manifold during every test conducted in this premixed combustion method and could not be operated at WOT.

The next phase of testing utilized a different location for the fuel delivery pipe fittings in the intake manifold. The gaseous fuel was now introduced in the intake manifold before the partition of the intake runners. Four holes were drilled in the intake manifold before the partition that led to each intake valve. Therefore, a premixed air/fuel mixture was delivered to each intake valve. Different nozzle sizes were also used during this testing including the use of the 4-hole nozzle. However, testing in this configuration did not increase output torque and also resulted in severe preignition in the intake manifold. Again, this preignition occurred after 1-2 minutes of operation.

B. Simulated Direct Injection Operation

Because of the severe limitations on engine performance due to preignition in the premixed combustion method, the passageway leading to the fuel intake valve was again blocked off. The fuel delivery pipe fittings with the single-hole nozzles were then mounted in the fuel injector holes where the M100 fuel injectors would normally be located. This configuration allowed for mixing of the air and fuel only in the combustion chamber; therefore, simulating direct injection fuel delivery.

The original pipe fitting with a nozzle opening equal to the fitting inside diameter (1/4-inch) resulted in smooth engine operation with a maximum torque of 58 ft-lbs at an A/F ratio of 8.3. The fuel delivery pressure measured here was 52 psi.

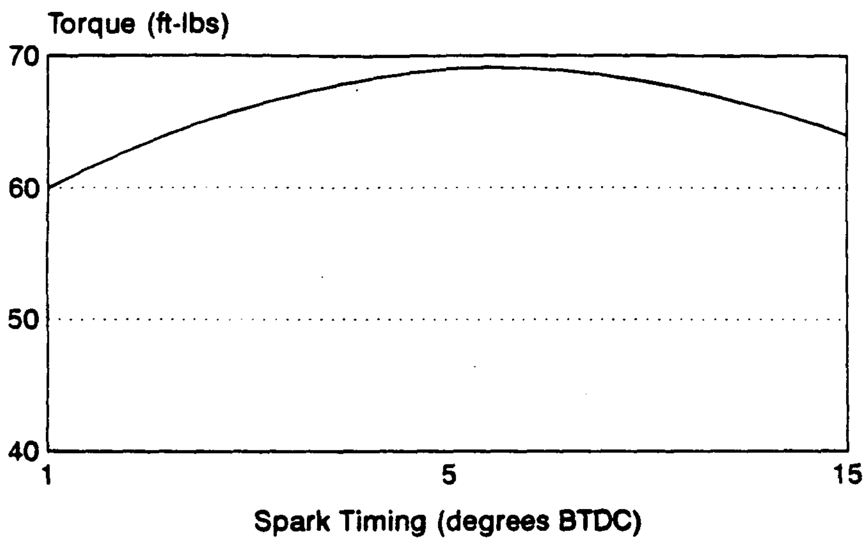
The first phase of testing conducted here utilized nozzles from the previous subsection. The maximum fuel flow attainable with these nozzles varied substantially. The largest fuel flow obtained was 9.9 cfm, resulting in an A/F ratio of 11.0 at WOT, 2,000 rpm operating conditions. The highest torque achieved during this testing was 66 ft-lbs. However, the engine was still affected by a slight preignition in the intake manifold.

The nozzle opening offering the best results here was a 3-millimeter diameter opening with no diffuser. This fuel delivery configuration resulted in very smooth engine operation and the highest output torque obtained to date with this engine is 80 ft-lbs at 2,000 rpm, WOT operating conditions. Although this is still 80 percent of the output torque obtained when using M100 fuel at the same conditions, several literature sources suggest that this may in fact be the output limit when using dissociated methanol fuel.[14,15] These literature sources quantify the maximum obtainable torque when operating on dissociated methanol as 55 percent of the M100 level. The 80 ft-lbs obtained with this engine in this configuration may be close to the maximum attainable output torque at these operating conditions with the gaseous fuel. The engine also ran very smooth, and operation at WOT was not hindered by preignition during any test conducted during this phase of testing.

Several other operating conditions were also evaluated, including fuel delivery pressure measured after the Tylan mass flow controller and before the hydrogen flame arrestor. Another operating condition that was varied was spark timing. Figure 3 below presents a curve fit for the maximum torque obtained at several different ignition timings. All torque valves presented here were maximum values and obtained at 2,000 rpm, WOT conditions. Fuel delivery pressure here was held relatively constant at about 60 psig. The largest torque at these conditions seemed to occur with a spark timing of 5 degrees before-top-dead-center (5°BTDC).

Figure 3

CA18DE Engine, 2H2/CO Fuel
Simulated Direct Injection Operation

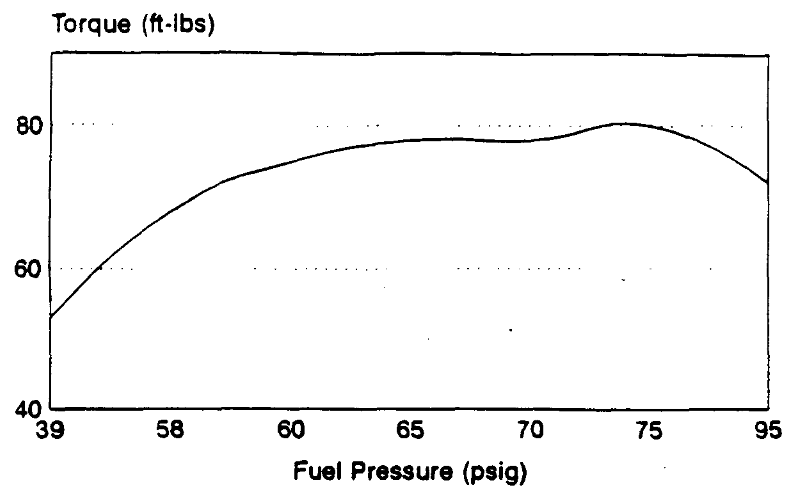


2,000 rpm, WOT, 60 psig fuel pressure.

The next variable operating condition investigated was fuel delivery pressure. This fuel pressure was measured in the fuel delivery system after the Tylan mass flow controller and before the hydrogen flame arrestor. Spark timing was set at 5°BTDC.

The fuel pressure measured here was increased by increasing the regulator pressure at the gas bottle. Figure 4 below presents the results obtained at 2,000 rpm and WOT conditions.

Figure 4
CA18DE Engine, 2H2/CO Fuel
Simulated Direct Injection Operation



2,000 rpm, WOT, 5BTDC spark timing.

The maximum torque obtained during this testing was achieved with a fuel pressure of 75 psig. When the pressure was increased above 75 psig, output torque seemed to taper off. For example, at a fuel pressure of 95 psig, output torque reduced to 72 ft-lbs.

Table 2 below presents all the results obtained from this testing in the simulated direct injection configuration. All torque values presented were maximum. The nozzles used had the single-hole, 3-millimeter diameter openings without a diffuser, and all results were obtained at 2,000 rpm, WOT operating conditions.

Table 2

Simulated Direct Injection Operation
CA18DE Engine, 2,000 RPM/WOT Conditions

Spark Timing (BTDC)	Fuel Pressure (psig)	Air Flow (cfm)	Fuel Flow (cfm)	A/F Ratio	Torque (ft-lbs)
1	58	33	15.5	5.8	60
5	39	30	14.1	5.8	53
5	58	37	16.7	6.0	69
5	60	37	19.2	5.2	75
5	65	39	20.9	5.1	78
5	70	37	24.0	4.2	78
5	75	37	22.5	4.5	80
15	60	37	16.6	6.0	64

Engine operation was kept near to stoichiometric (A/F ratio of 6.5) for most of this testing. Several literature sources again suggest best engine performance occurs at or slightly rich of stoichiometric when operating on hydrogen fuel.[1-5]

C. <u>Emissions and Combustion Analysis</u>

The largest torque obtained during the previous testing with $2H_2/CO$ fuel was 80 ft-lbs (80 percent of the M100 fuel output). As a result, emissions and combustion analysis were performed at similar operating conditions (A/F ratio of 4.5, 2,000 rpm, WOT, and a fuel pressure of 75 psig). The A/F ratio of 4.5 utilized with the gaseous fuel was similar to the closed-loop controlled M100 A/F ratio of 4.7 at the same 2,000 rpm, WOT conditions. Table 3 below details the operating conditions used for both fuels when emissions samples were collected and combustion analysis was performed.

Table 3

Engine Operating Conditions: CA18DE Engine

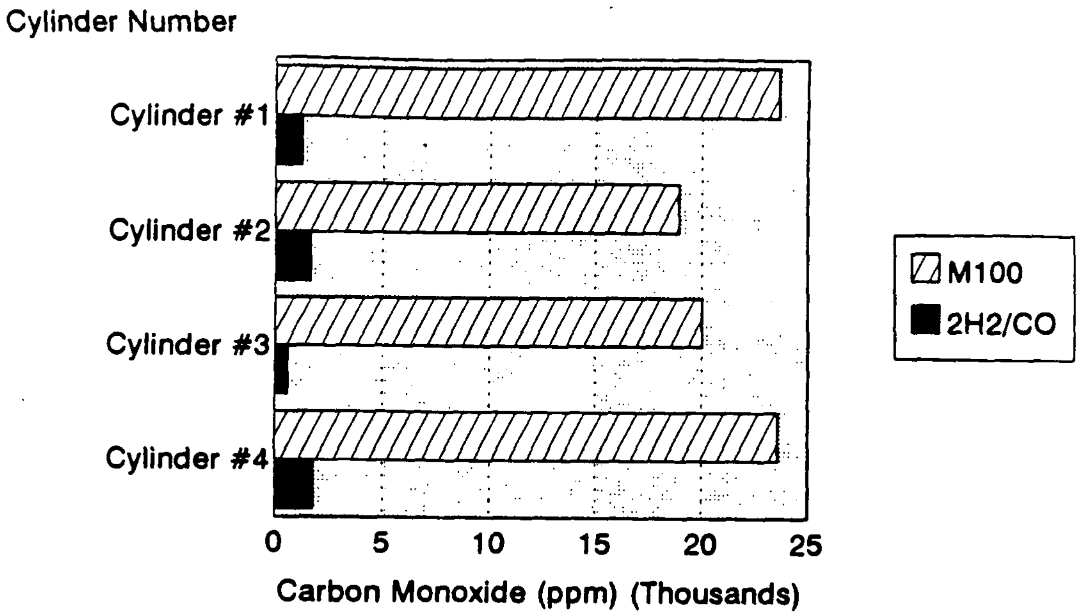
Fuel	**Speed (rpm)**	**Throttle Position**	**A/F Ratio**	**Fuel Pressure (psig)**	**Air Flow (cfm)**	**Torque (ft-lbs)**
M100	2,000	WOT	4.7	45	39	100
$2H_2/CO$	2,000	WOT	4.5	75	37	80

Cylinder-out emission samples were taken for each of the four cylinders. Two samples for each cylinder were taken when operating on each fuel. These were raw (not diluted) emission samples and were analyzed using an FTIR system. Each sample taken was collected over approximately 4 minutes.

Figure 5 below presents cylinder-out carbon monoxide (CO) concentrations when using either fuel. The top bar represents the CO emission concentration leaving cylinder #1 when operated on M100 fuel. Similarly, the bar underneath that represents the CO concentration leaving cylinder #1 when operating on $2H_2/CO$ fuel. These results are the average of two test runs and are presented in parts per million (ppm). Again, these are raw emission levels and not diluted.

Figure 5

Cylinder-Out Carbon Monoxide Levels



2,000 rpm, WOT conditions.

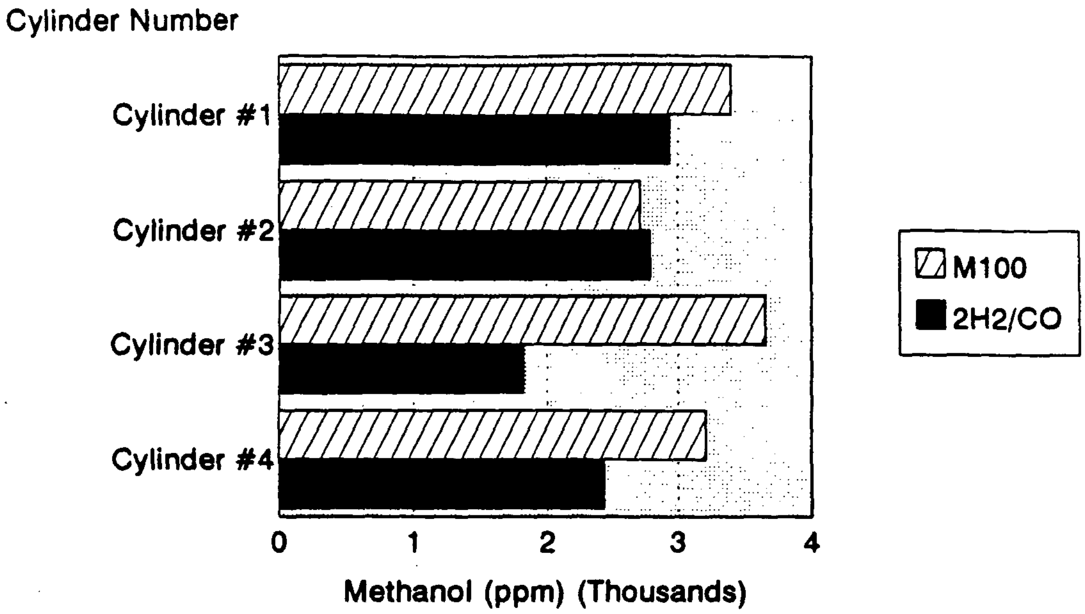
Concentrations of CO when using the gaseous $2H_2/CO$ fuel were much lower than when utilizing M100 fuel. These low CO concentrations, ranging between 590 ppm for cylinder #3 and 1,750 ppm for cylinder #4, seem to indicate that most of the $2H_2/CO$ fuel is combusting and that very little unburned fuel is resulting. This was not the case in previous testing with $2H_2/CO$ fuel when large amounts of unburned fuel were present in the exhaust.[6,13]

From this figure, combustion stability from cylinder-to-cylinder may also be investigated. With M100 fuel, each cylinder is producing approximately equal amounts of CO. The largest variance in cylinder-out CO levels occurs between cylinders #1 and #2, where a 20 percent variance occurs. However, when $2H_2/CO$ fuel is used, cylinder #3 produces a much lower amount of CO than the three other cylinders. For instance, cylinders #3 and #4 differ by 66 percent in CO emission levels.

Figure 6 presents results from methanol emission sampling for the same testing described in Figure 5.

Figure 6

Cylinder-Out Methanol Emission Levels



2,000 rpm, WOT conditions.

When operating on the gaseous $2H_2/CO$ fuel, large concentrations of methanol emissions were produced in each individual cylinder. Also, in cylinder #2, levels of methanol emission concentrations when operating on this fuel were higher than M100-fueled levels.

When operating on M100, combustion stability with regard to unburned fuel in each cylinder was again very good. The largest variance in unburned fuel occurred between cylinders #2 and #3, where a 26 percent difference in methanol emissions occurred. When operating on the $2H_2/CO$ fuel, cylinder #3 levels again varied from the other three cylinders by a substantial margin. However, the largest variance between cylinders when excluding cylinder #3 is only 17 percent.

Table 4 below presents several other emission levels measured during this same testing. All levels are presented in ppm's except for water vapor (H_2O) and carbon dioxide (CO_2), which are presented in a percentage.

Table 4							
2,000 RPM/WOT Conditions, Simulated Direct Injection Operation Cylinder-Out Emission Levels							
Cylinder Number	**Fuel**	**H_2O (%)**	**CO (ppm)**	**CO_2 (%)**	**NOx (ppm)**	**CH_3OH (ppm)**	**HCHO (ppm)**
#1	M100 $2H_2/CO$	12 17	23,650 1,200	1.0 0.9	825 1,800	3,400 2,935	60 7
#2	M100 $2H_2/CO$	11 14	18,930 1,620	1.0 0.9	1,100 1,475	2,715 2,785	45 5
#3	M100 $2H_2/CO$	10 11	20,000 590	0.7 0.5	520 N/A	3,655 1,815	45 4
#4	M100 $2H_2/CO$	11 13	23,595 1,750	1.0 0.8	925 1,540	3,210 2,425	40 5
Average	M100 $2H_2/CO$	11 14	21,544 5,160	0.9 0.8	842 1,605	3,245 2,490	48 5
N/A = Not available							

Formaldehyde (HCHO) emission levels are much lower when operating on the gaseous $2H_2/CO$ fuel for each cylinder. Formaldehyde levels when the $2H_2/CO$ fuel was used were, on the average, 90 percent below corresponding M100 levels for each cylinder. However, emissions of nitrogen oxides (NOx) were substantially larger when the $2H_2/CO$ fuel was used. On the average, NOx levels approximately doubled from M100 levels when the gaseous fuel was used.

Two overall dilute emission samples with each fuel were also taken. The operating conditions were again similar to those described in Table 3 for this testing. These diluted samples were analyzed at a different site using different analyzers. Table 5 below presents the average of two emission samples when operated on each fuel. Brake-specific emissions are presented here (grams/brake horsepower-hour). Methanol and formaldehyde levels were not measured during this testing, and only a total hydrocarbon (HC) value is presented here.

Table 5						
2,000 RPM/WOT Conditions, Simulated Direct Injection Operation Brake-Specific Emission Levels						
Fuel	**BhP**	**A/F Ratio**	**HC (g/BhP-hr)**	**CO (g/BhP-hr)**	**NOx (g/Bhp-hr)**	**CO_2 (g/BhP-hr)**
M100	38.1	4.7	1.37	24.7	2.0	205
$2H_2/CO$	30.5	4.5	0.03	49.0	8.0	450

HC levels when using the gaseous fuel were substantially lower than M100 levels on a brake specific basis. However, brake-specific CO levels approximately doubled from M100 levels with the $2H_2/CO$ fuel. This level of CO is substantially lower, however, than levels obtained in a previous report [6] and is also accompanied with a higher torque of 80 ft-lbs. Although NOx levels with $2H_2/CO$ fuel were much higher, these levels are similar to previous NOx emissions.[6]

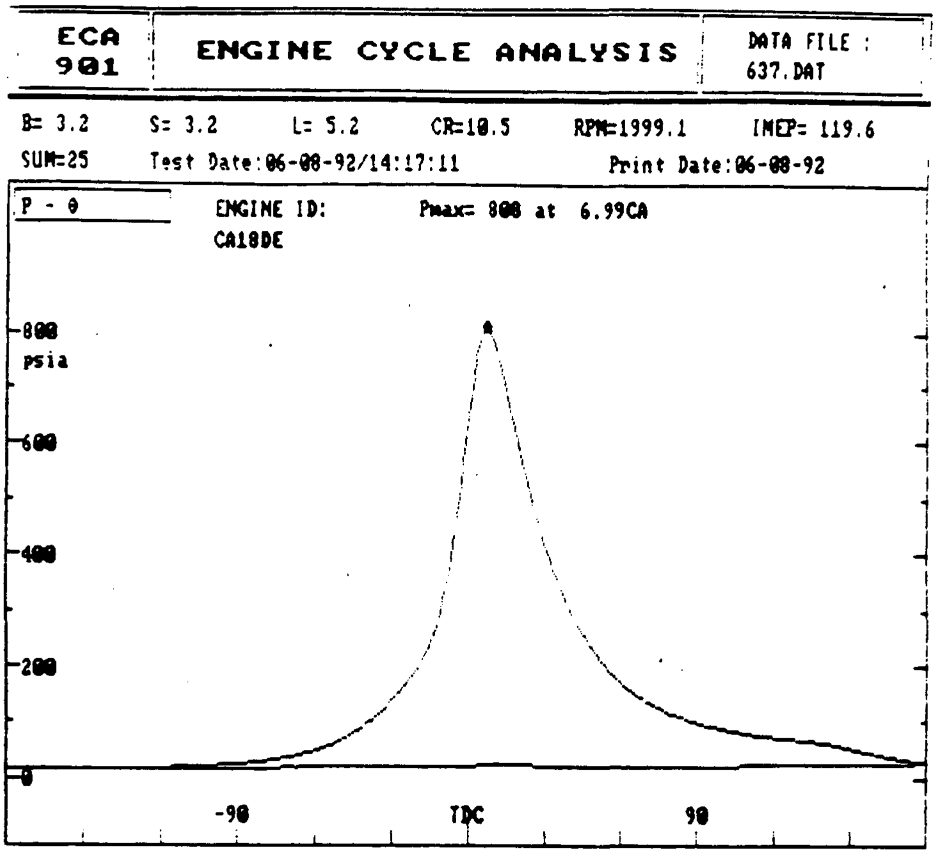
The second phase of testing in this subsection consisted of investigating the in-cylinder combustion process when utilizing each fuel. This investigation was conducted using a Kistler pressure sensor and the test cell data acquisition system. This investigation was performed using a pressure versus crank angle (CA) approach. The computer software program then computes eight other outputs based on the experimental pressure/CA diagram, engine geometry, and operating conditions entered at the beginning of each test. There are four basic categories of functions which result from the further analysis of the pressure/CA data. These include dynamic cylinder pressure and temperature data needed for a thermal stress analysis, heat release rate and cumulative heat release, cycle performance and losses from the indicator diagram, and cycle thermodynamic data.[9] This report will only investigate pressure and heat release data.

When operating on each fuel, pressure data was gathered from each cylinder individually. However, data from each cylinder was very similar; therefore, only one set of data for operation on each fuel will be presented in this report. When M100 fuel was used, 25 consecutive cycles were monitored and averaged by the computer. With $2H_2/CO$ fuel, fifty consecutive cycles were averaged. The same operating conditions for each fuel, presented in Table 3, were again used in this phase of testing. The maximum torque obtained at 2,000 rpm, WOT conditions with M100 fuel was 100 ft-lbs, with $2H_2/CO$, it was 80 ft-lbs.

Figure 7 below presents in-cylinder pressure versus crank angle data when using M100 fuel. The maximum pressure obtained with M100 fuel reached 808 psi. This maximum pressure occurred at approximately 7 degrees after top-dead-center (ATDC). Spark timing with M100 fuel was 22 degrees BTDC. The indicated mean effective pressure (imep) was calculated by the software to be 119.6 psi for this fuel. The standard deviation in maximum pressure values for 25 consecutive cycles was found to be 28 psi.

Figure 7

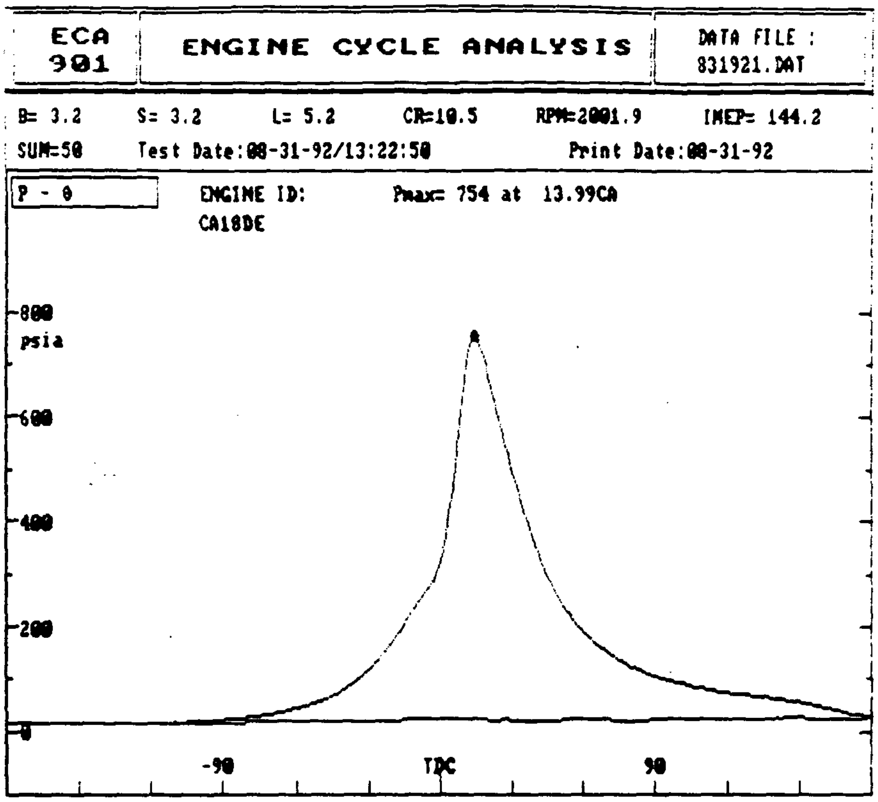
Pressure Trace With M100 Fuel



A similar trace was obtained when the $2H_2/CO$ fuel was used. (Figure 8) Maximum pressure levels reached with this fuel were slightly lower than M100 fuel, at 754 psi. This maximum pressure was achieved at 14 degrees ATDC. Spark timing with this fuel was set at 5 degrees BTDC. Although the maximum output torque achieved with $2H_2/CO$ fuel is approximately 20 percent lower than M100 levels, the calculated imep when using the gaseous fuel is about 20 percent higher. The imep again is a calculated quantity derived from pressure, engine geometry, and operating conditions data. The higher imep values with gaseous fuel operation may be attributed to the following. First, the number of cycles used for M100 operation was different from that of $2H_2/CO$ operation, which may alter the software averaging process. Also, prior to $2H_2/CO$ fuel operation, the optical encoder was removed then remounted on the engine. This may significantly change encoder position and consequently imep measurement. The standard deviation in maximum cylinder pressure with $2H_2/CO$ fuel was 14 psi, denoting less fluctuation and more stable operation than with M100 fuel.

Figure 8

Pressure Trace For $2H_2/CO$ Fuel



One possible way of investigating flame speed/combustion rate is by investigating the heat release rate. Recent literature suggests that this quantity is proportionally related to mass burned rate (energy release), with only a slight difference involved, resulting from neglecting the heat release through the cylinder walls and blowby.[10,11,12] The software uses this quantity because it is much faster to calculate a rate of heat release than a rate of combustion.[9]

Heat release rate, denoted $dX/d\theta$ is calculated from cylinder pressure/volume and crank degrees data by using the following formula[9]:

$$dX/d\theta = [1/(\theta_2 - \theta_1)][1/(n-1)][P_2V_2 - P_1V_1) + (1/2)(P_1 - P_2)(V_2 - V_1)$$

Where:

$dX/d\theta$ = Heat release rate between two crank angles (Btu/degree)

P = Cylinder pressure (psi)

V = Cylinder volume (cubic inches)

n = Polytropic index.

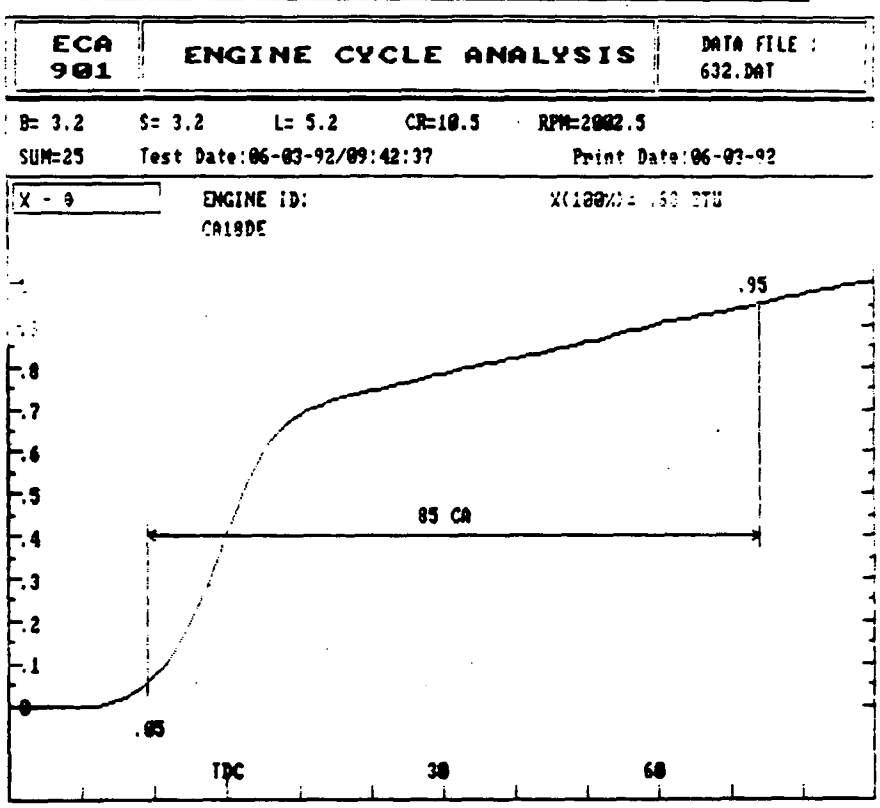
The maximum heat release rate (related to mass burned rate) obtained with M100 fuel was 0.035 Btu/CA. Similarly, the maximum heat release rate with the $2H_2/CO$ fuel was 0.050 Btu/CA.

From the heat release rate values, it is possible to investigate a combustion duration. This duration period can be calculated from the cumulative heat release (cumulative energy release) versus crank angle data. This quantity is proportional to the amount of chemical energy which must be released as thermal energy by combustion to match the measured pressure data.

Figure 9 below presents the trace of cumulative heat release (denoted X) versus crank angle when M100 fuel was used. The y-axis here represents the fraction of the total heat released during an average of 25 combustion cycles. For instance, at TDC, approximately 30 percent of the total heat release when combusting this fuel has occurred. The start of combustion has been selected as X=0.05, or when 5 percent of the total heat release has occurred. Similarly, the end of combustion has been selected as X=0.95, or when 95 percent of the total heat release has occurred. These values were chosen based upon the recommendation of the software developers.[9]

Figure 9

<u>Cumulative Heat Release With M100 Fuel</u>

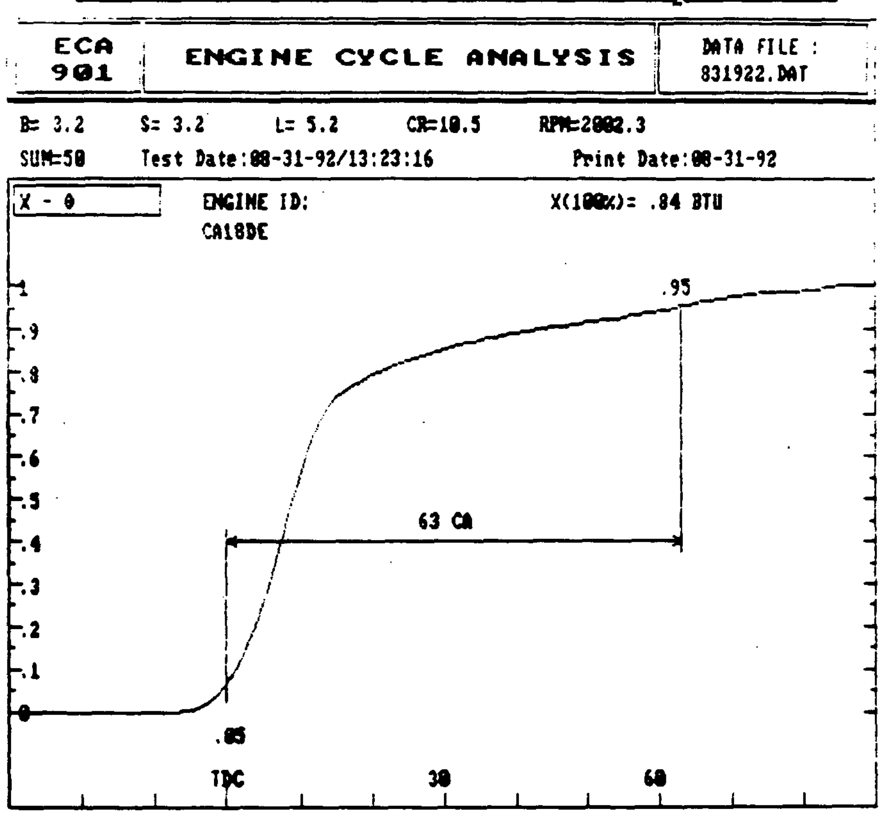


The total heat release duration (related to combustion duration) was 85 crank degrees for M100 fuel. Previously, this duration was calculated to be approximately 60 crank degrees for M100 fuel. However, with the shorter combustion duration, measured output torque was much higher (128 ft-lbs) than the level measured during this testing (100 ft-lbs). It was not determined why output torque dropped off from levels obtained during previous testing with M100 fuel.[13] The heat release trace here reveals a fast-burn rate until approximately 70 percent of the total heat release has occurred. After this point, the heat release rate appears to be very slow and prolonged.

Figure 10 below presents a similar trace for $2H_2/CO$ fuel operation. The total heat release duration with this fuel was approximately 63 crank degrees, substantially faster than the M100 results. A faster burn rate would be expected with a hydrogen-containing fuel. At 20 crank degrees after spark, (15°ATDC) approximately 80 percent of the combustion has occurred. Eighty percent of the total combustion was not reached until 60 crank degrees after spark with M100 fuel. The total amount of heat released during one combustion event with the $2H_2/CO$ fuel was 0.84 Btu. The corresponding value with M100 fuel was 0.63 Btu, approximately 25 percent lower.

Figure 10

<u>Cumulative Heat Release With $2H_2/CO$ Fuel</u>



VII. Summary and Conclusions

This engine conversion project was started to develop a suitable test bed for a practical on-board methanol dissociator. From the engine performance, the following main results can be summarized:

1. The engine currently operates smoothly on dissociated methanol fuel at approximately 80 percent of the M100 output power level. This power output is considerably higher than the maximum power levels reported in another literature source [14], when the engine operated on 100 percent dissociated methanol.

2. In premixed combustion, the engine must run on a lean mixture. Abnormal combustion is likely to occur when the engine warms up. This is considered to be a restriction on the range of dissociated methanol operation.

3. The standard deviation in maximum cylinder pressure with the $2H_2/CO$ fuel is less than M100 fuel deviation. This indicates less fluctuation and more stable engine operation with the gaseous fuel.

4. Dissociated methanol burned much faster than methanol probably resulting from the hydrogen content in the fuel, which has a flame speed over five times that of methanol (see Figures 9 and 10).

5. The lower efficiency with dissociated methanol (as is the case with any gaseous fuels) could be attributed to the increase in heat transfer resulting from higher gas temperatures and the increase of compression work. Gas temperature and compression pressure increases can be explained based on the thermodynamic analysis of the pressure-volume equation of an ideal gas.

6. Emission results were obtained for both fuels, however, definitive conclusions can not be drawn at this stage.

VIII. Future Efforts

A 100 percent efficient methanol dissociator is currently not available. Presently, the best direction for this project may be to evaluate what the $(2H_2/CO)/CH_3OH$ ratio is for best emissions. Running the engine on a mix of dissociated methanol and M100 may yield more power and decrease the uncontrollable combustion and flashback at high loads experienced with $2H_2/CO$ operation. Also, an optimized mixture of vaporized and dissociated methanol may lead to satisfactory cold start performance, enhance stability during idling, and reduce exhaust emissions.

Development and fabrication of a dissociated methanol-assisted injector adaptor is complete. In this case, resistors can be added to the adapters at a small cost to enhance cold start operation if it is needed.

IX. Acknowledgments

The CA18DE engine described in this report was modified for use with M100 neat methanol and loaned to EPA by the Nissan Motor Corporation as support for an effort to investigate the potential of neat methanol as an alternative motor vehicle fuel.

The authors appreciate the efforts of Marc Haubenstricker of the Fuels and Chemical Analysis Branch for FTIR analysis support. The authors also appreciate the efforts of Jennifer Criss and Mae Gillespie of the Technology Development Group for typing and editing support.

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APPENDIX A

TEST ENGINE SPECIFICATIONS, M100 FUEL OPERATION
CONDITION AS LOANED BY NISSAN TO EPA

Manufacturer	Nissan Motor Co., LTD
Basic engine designator	CA18DE
Displacement	1809 cc
Cylinder arrangement	4-cylinder, in-line
Valvetrain	Dual-overhead camshaft
Combustion chamber	Pentroof design
Bore X Stroke	83 mm x 83.6 mm
Compression ratio	10.5
Compression pressure	16.5 kg/square cm (350 rpm, 80°C)
Fuel control system	Electronically controlled fuel injection
EGR	EGR not used
Valve clearance	0 mm (automatically adjusting)
Idle speed	750 rpm
Engine oil	Special formulation supplied by Nissan for methanol engine operation
Fuel	M100 neat methanol
Air/fuel control	Excess air ratio may be varied from 0.5 to 2.0 by means of an external control
Spark advance control	Ignition timing can be varied from 0°BTDC to 54°BTDC by means of an external control