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Technical Report

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

by

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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 ($2H_2/CO$).

Previously, several engine/fuel system modifications were made in an attempt to increase power output when the gaseous fuel was used [1]. Several intake port/fuel system configurations as well as 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. When using $2H_2/CO$ fuel, the largest torque achieved at 2,000 rpm, WOT operating conditions was 80 ft-lb. This represents about 80 percent of the maximum torque levels obtained with M100 fuel at the same WOT, 2,000 rpm operating conditions.

This torque level was considered satisfactory for gaseous fuel operation. It was then necessary to monitor engine performance (output torque, combustion efficiency, and emissions) at several different operating conditions. This report summarizes engine performance when operating on both M100 and $2H_2/CO$ fuels at several different operating conditions. The main goals of this final testing were to demonstrate that there were certain operating conditions where the engine produced more/similar torque, less CO emissions, and had a better combustion efficiency when operating on $2H_2/CO$ fuel.

When the $2H_2/CO$ fuel was used, two different intake camshafts were used. The stock camshaft had the same cam profile for both the air and fuel valves and was also used with M100 operation. This camshaft was used when the torque value of 80 ft-lb was achieved with the gaseous fuel. The modified camshaft had a very short and fast valve lift for the fuel valve and the same air valve event as the stock camshaft. The maximum torque achieved with this camshaft during this testing was 43 ft-lb. Tests were conducted at both 2,000 and 1,500 rpm.

When the stock intake camshaft was utilized with $2H_2/CO$ fuel, torque values were very similar to levels obtained with M100 fuel, even higher at low loads (only varying by 20 percent at WOT). At high loads with the modified camshaft, torque values began to decrease from M100 levels more rapidly, perhaps due to the very short fuel valve event. However, there were several operating conditions when using $2H_2/CO$ fuel that resulted in higher torque values than were achieved with M100 fuel.

The next goal was to operate the engine on $2H_2/CO$ fuel and achieve similar CO emission levels as when fueled with M100. With the stock camshaft during medium-load operation, the engine produced somewhat similar levels of CO when operating on either fuel. ($2H_2/CO$ fuel operation produced about 30 percent higher CO emissions.) However, when the modified camshaft was used during medium-load operation, both fuels produced almost equal amounts of CO emissions.

The last goal of this program was to match combustion efficiencies when using either fuel. During low to medium-load operation, combustion efficiency values for each fuel were very similar at 2,000 rpm. At higher loads, however, efficiency values with the gaseous fuel begin to taper off from M100 levels.

This test program did meet the goals originally set for this engine conversion program. First, the engine was successfully converted to operation on simulated dissociated methanol fuel. Also, after much engine and fuel system optimization work, the engine did perform comparably to M100 operation when the $2H_2/CO$ fuel was used. Similar torque values, CO emissions, and combustion efficiencies were noted during low to medium-load operation for both fuels.

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 reports.[2,3]

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 was 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 and Fuel System Modifications

Several modifications were made to this engine by EPA since its delivery from Nissan; these modifications were detailed in previous EPA technical reports.[1,2,4] 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.

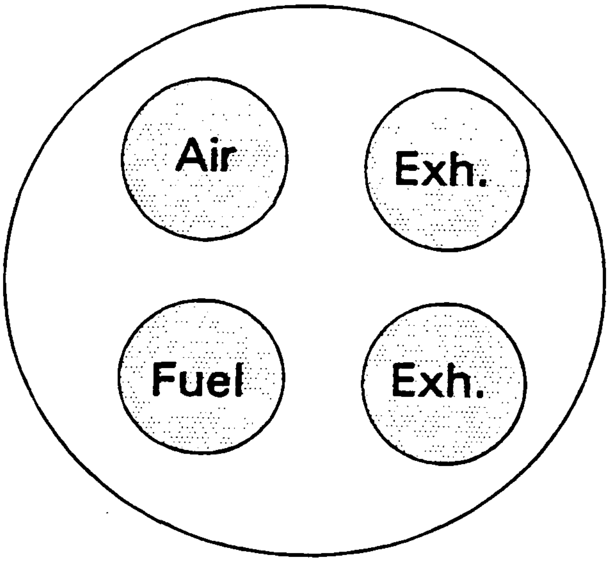
The test engine was initially modified by Nissan to better utilize the qualities of M100 neat methanol over unleaded gasoline. These modifications were discussed in detail in an earlier report.[5] A summary of the test engine specifications when fueled with M100 neat methanol is included in this report in 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 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). When operating on the gaseous fuel, 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 was used for all testing described in this report with the $2H_2/CO$ fuel.

Figure 1

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



This valve scheme allowed for the admission of gaseous fuel through only 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 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 hole in the assembly left by the power valve slide was sealed to prevent leakage of fuel and air between runners.

With $2H_2/CO$ fuel operation, two different intake camshafts were utilized. The first was the stock M100 camshaft with similar air and fuel valve events. The valve lift here was 0.335 inches with a total valve event of 225°. The modified camshaft utilized different valve events for both air and fuel. The air valve scheme remained the same as the stock camshaft. The lobes on the fuel valves were altered to a total lift of 0.200 inches and a total valve event of 100°. The development of this modified camshaft was discussed in a previous report.[5]

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 adapt to the fittings.

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. The fuel leading from the solenoid valve to a flame arrester 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 that few pressure drops in the fuel delivery system would occur.

Before the flame arrestor in the fuel line 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 mass flow controller used previously to control $2H_2/CO$ fuel was eliminated from the fuel delivery system. This caused the engine to operate at rich conditions when gaseous fuel was used.

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 posts in the valve control assembly. The inside diameter of these fittings (previously 1/4-inch) were replaced with different nozzle designs to increase fuel pressure and enhance the fuel distribution. A nozzle opening of 3 millimeters was used in this testing. 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.

IV. Exhaust Measurement Procedure

Dilute emission samples were taken during this latest phase of testing. Dilute emission samples were engine-out levels and taken downstream of the exhaust manifold. These 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 hung from ceiling supports. 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.

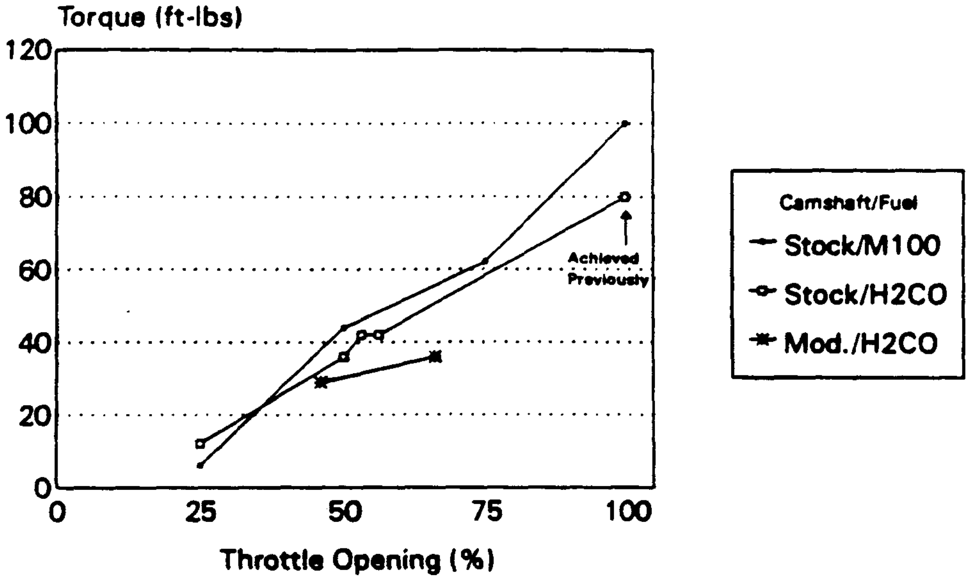
A gaseous sample line has been extended through the cell ceiling and connects the mechanical CVS with an electronic display panel in the cell control room. A fitting in the sample line at the control room enables bag sampling at this point. Analysis of bag samples is accomplished at a bank of analyzers located in another test cell. Hydrocarbons (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.

V. Discussion of Test Results

The goal of this last phase of testing was to operate the engine on M100 neat methanol and simulated dissociated methanol ($2H_2/CO$) fuels over a wide range of operating conditions to determine if there were operating conditions on $2H_2/CO$ fuel that were comparable to M100 power output, CO emissions, and combustion efficiency levels. First, testing was conducted at both 2,000 and 1,500 rpm when the engine load was varied from idle to wide-open-throttle (WOT) with A/F ratio also being varied. The engine was then converted to operation on $2H_2/CO$ fuel while utilizing the stock intake camshaft. Because each cylinder of fuel would operate the engine for about 10-15 minutes, only selected operating conditions were used here. Similarly, only a limited test sequence was followed when the stock intake camshaft was replaced with the modified version for $2H_2/CO$ fuel use. The spark timing for testing on M100 fuel was kept at 20°BTDC; similarly, with $2H_2/CO$ fuel, the spark timing was kept at 5°BTDC. Fuel pressure with M100 was 45 psig, and 75 psig with $2H_2/CO$. Only selected operating points were selected where the engine ran very smooth when $2H_2/CO$ fuel was used.

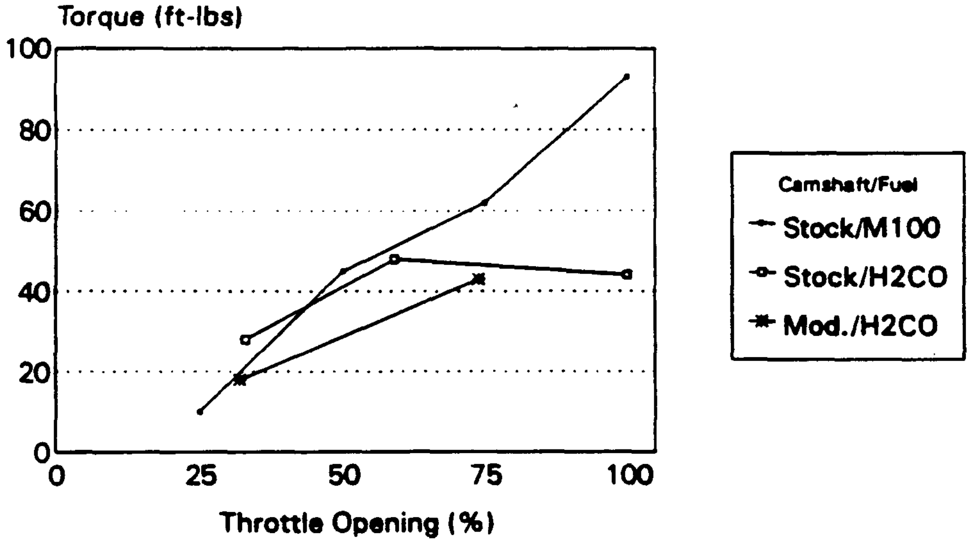
Figure 2 below presents brake specific torque results while using either fuel at 2,000 rpm. The horizontal axis represents the percentage of throttle opening (100 percent throttle opening represents WOT). Also, this data was obtained under rich conditions of Lambda equal to 0.7. From this plot, brake specific torque during low load operation (25 percent throttle opening) is higher with the $2H_2/CO$ fuel than with M100. During medium load operation (40-65 percent throttle opening), only a slight torque deviation from M100 levels results when operating on the gaseous fuel with the stock intake camshaft. Using the modified camshaft results in a larger drop in brake specific torque at higher engine loads. At WOT, there is about a 20 percent difference in power output between the two fuels using the stock camshaft. Operation with the modified camshaft resulted in leaner operating conditions because of the reduced fuel valve lift and duration.

Figure 2
Brake Specific Torque Values
Lambda = 0.7, 2000 rpm Operating Conditions



These trends in brake specific torque were also noted during similar testing at 1,500 rpm. Again, these results were obtained at rich conditions of a Lambda equal to 0.7. Also, at lighter loads below 30 percent throttle opening, brake specific torque levels with the gaseous fuel and the stock intake camshaft were larger than corresponding M100 levels. At these conditions, the modified camshaft testing also yielded similar torque levels as M100 fuel. During medium load testing, torque values associated with $2H_2/CO$ fuel use begin to fall below M100 levels. With the stock camshaft, torque values are only slightly lower than M100 levels, but using the modified camshaft has an even more detrimental affect on torque. Above 75 percent throttle opening, torque values associated with $2H_2/CO$ fuel fall well below M100 fuel levels. At WOT, the stock camshaft testing yielded a torque approximately 53 percent below M100 levels. This reduction in torque may be attributable to not adjusting the spark timing for changes in air/fuel richness at higher load conditions.

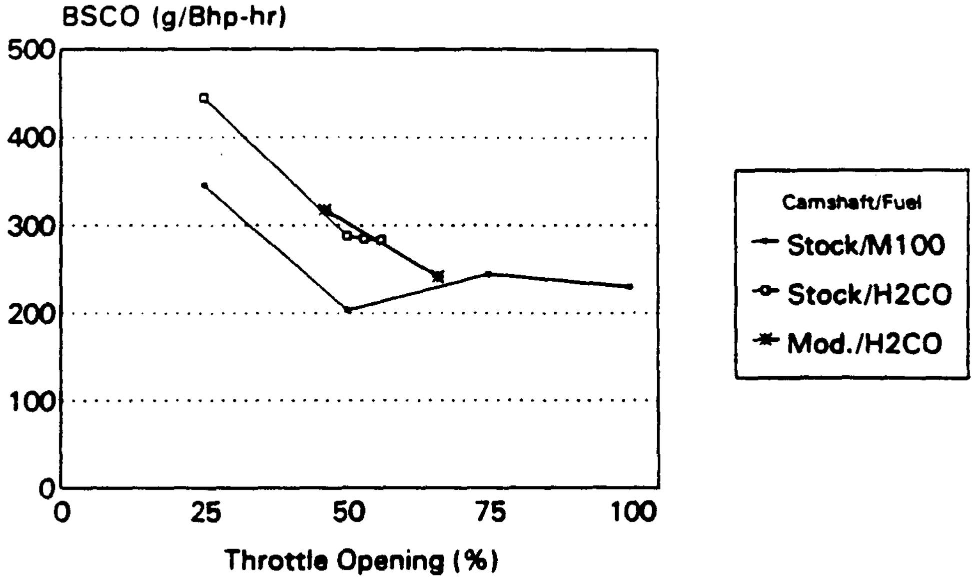
Figure 3
Brake Specific Torque Values
Lambda = 0.7*, 1500 rpm Operating Conditions



* Operated slightly leaner with modified camshaft

The second goal of this test program was to determine if there were operating conditions with $2H_2$/CO fuel that produced less CO emissions than with M100. As was the case with torque output, CO emission levels were indeed comparable between $2H_2$/CO fuel operation and M100 at some operating conditions. Figure 4 below presents CO emission levels in grams/BHP-hr over a wide range of load operation for each fuel at 2,000 rpm.

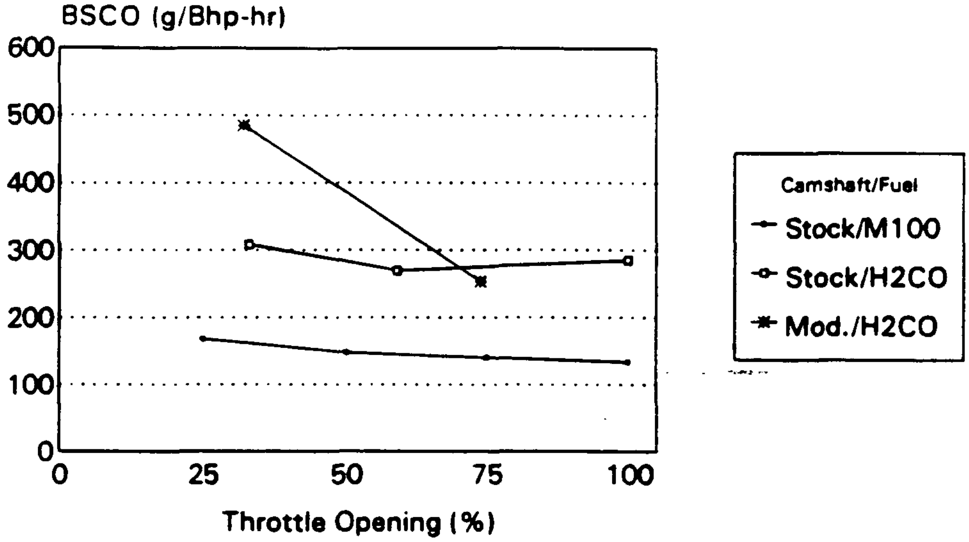
Figure 4
Brake Specific Carbon Monoxide Emissions
Lambda = 0.7, 2000 rpm Operating Conditions



At light loads (25 percent throttle opening), both fuels produced a large amount of CO emissions. However, when using the gaseous fuel, CO emissions at light loads were about 29 percent higher than M100 levels. During medium load operation, CO levels with $2H_2/CO$ fuel dropped below the light-load M100 levels. However, with the stock intake camshaft, the difference in CO levels between M100 and $2H_2/CO$ fuels at each operating point remained similar to the difference during light-load operation (about 30 percent). However, when the modified intake camshaft was used, CO levels between the two fuels were approximately equal at a throttle opening of 66 percent.

Figure 5 presents CO emission levels from testing at a 1,500 rpm engine speed. During this testing, operating on $2H_2/CO$ fuel resulted in higher CO emission levels for every operating condition investigated. CO levels with M100 fuel at 1,500 rpm were also much lower than when operating at 2,000 rpm. With the stock intake camshaft and $2H_2/CO$ fuel, CO emission levels remain approximately twice as large as corresponding M100 levels over the entire engine load range investigated. Below 75 percent throttle opening, the use of the modified camshaft resulted in higher CO emissions than the stock camshaft.

Figure 5
Brake Specific Carbon Monoxide Emissions
Lambda = 0.7*, 1500 rpm Operating Conditions

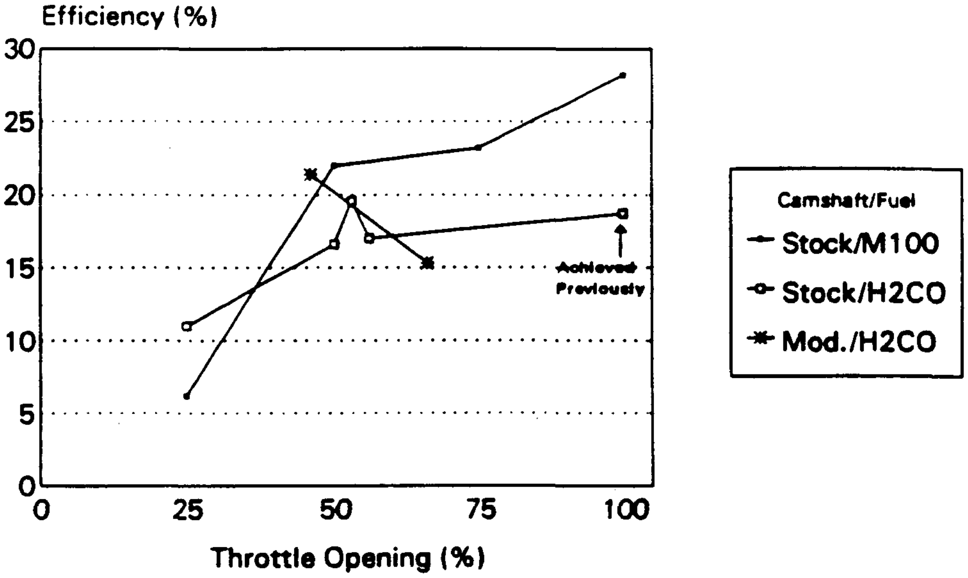


* Operated slightly leaner with modified camshaft

Obtaining lower CO levels with $2H_2/CO$ fuel than with M100 seems to be very difficult. Although torque levels were greater at some operating points with $2H_2/CO$ fuel, it was not possible to obtain lower CO emissions with this fuel when compared to M100 levels. This could be partly attributable to unburned fuel and incomplete combustion, both contributing to CO formation when $2H_2/CO$ fuel is used, especially at rich conditions. With M100, only incomplete combustion contributes to this phenomenon. Also, during testing with the gaseous fuel, it was not possible to fully warm the engine prior to emissions testing because of the limited amount of fuel present in the gas bottle. The engine oil temperature during emissions testing with M100 fuel was about 180-200°F; the corresponding temperature with $2H_2/CO$ fuel was 90-120°F. Therefore, the combustion chamber was much colder, resulting in greater amounts of unburned fuel and incomplete combustion, both contributing to higher CO emissions.

The last goal in this test program was to operate the engine on $2H_2/CO$ fuel at a greater combustion efficiency than when operated on M100. Figure 6 below presents combustion efficiency values over the same engine load range at 2,000 rpm and Lambda = 0.7 conditions.

Figure 6
Brake Thermal Efficiency Values
Lambda = 0.7, 2000 rpm Operating Conditions



Combustion efficiency was calculated based on the brake horsepower output, fuel consumption, and heat of combustion of the fuel used by the following formula:[6]

$$\eta = [BHP \times K_p/J]/M_f Q_c$$

Where:

η = Brake thermal efficiency
BHP = Brake horsepower
K_p = Value of 1 hp expressed in (force x length/time) units
J = Joule's law constant
M_f = Mass of fuel supplied per unit time
Q_c = Heat of combustion of a unit mass of fuel.

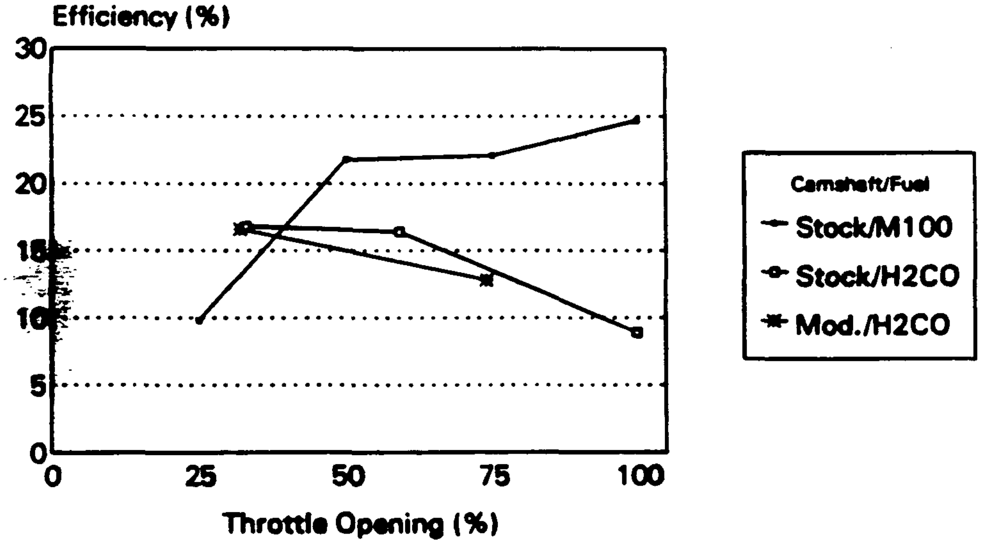
Low load operation below 30 percent throttle opening using the $2H_2/CO$ fuel with the stock intake camshaft resulted in higher brake thermal efficiency values than when operating on M100. At greater loads with the stock intake camshaft, brake efficiency values begin to diverge below corresponding M100 levels, resulting in about a 10 percent thermal efficiency difference at WOT operating conditions. Again, this plot is only for rich operating conditions of Lambda = 0.7.

Testing with the modified intake camshaft also provided operating points where the engine was more efficient operating on gaseous fuel than with M100. At about half throttle, the brake thermal efficiency when operating on $2H_2/CO$ fuel was slightly greater than the M100 efficiency curve. However, as the engine load was increased, efficiency values with the modified intake camshaft dropped substantially. However, there again were operating conditions at Lambda = 0.7, 2,000 rpm where the engine was more efficient operating on $2H_2/CO$ fuel.

Figure 7 presents similar traces of brake thermal efficiency for each fuel, however, when operating at 1,500 rpm. Again, below a throttle opening of about 40 percent (low load), brake thermal efficiencies were greater when operating on $2H_2/CO$ fuel rather than M100. This was the case for both the stock and modified camshaft. During middle to full load operation, brake efficiency values dropped off considerably from corresponding M100 levels. This efficiency drop may again be attributed to not adjusting the spark timing when the engine was operated at leaner conditions here. At WOT, a thermal efficiency difference of 15 percent results between operation on the two fuels. However, at lower loads, the goal of higher brake thermal efficiencies on dissociated methanol fuel operation was realized.

Figure 7

Brake Thermal Efficiency Values
Lambda = 0.7*, 1500 rpm Operating Conditions



* Operated slightly leaner with modified camshaft

Again, all the previous figures' results were obtained at rich operating conditions (Lambda = 0.7). This was the case because the fuel controller used previously during $2H_2/CO$ fuel operation was eliminated from the fuel delivery system. An air/fuel ratio was used to monitor richness with both fuels. The stock intake camshaft always allowed the engine to operate at a Lambda value of 0.7 at a fuel delivery pressure of 75 psig. However, when the modified intake camshaft was used, engine operation became much leaner, varying anywhere between a Lambda value of 0.8 to 1.1, depending on throttle position. The air/fuel richness when operating on M100 was controlled by an external controller provided to us by Nissan. The air/fuel value would be changed by turning the dial and monitoring the Lambda value on the air/fuel ratio meter.

Appendix B of this report contains all the power output, efficiency, and emissions data from testing on M100 fuel at both 2,000 and 1,500 rpm conditions. The test sequence started with selecting the WOT condition, and emission bag samples and torque values were taken at three different fuel richness points: rich (Lambda = 0.7), stoichiometric (Lambda = 1.0), and lean (Lambda = 1.4). Four throttle openings were evaluated; 100 percent (WOT), 75 percent, 50 percent, and 25 percent throttle openings. All emissions values are presented in brake specific form (grams per brake horsepower-hour) and are engine-out levels; there was no catalyst present in the exhaust system.

When operating on M100, the engine produced much higher emissions levels of hydrocarbons and CO at rich conditions (Lambda = 0.7). At rich conditions, these levels were approximately ten times higher than when operating at stoichiometric. The largest brake specific horsepower was obtained at WOT and rich conditions and was measured at 39.2 at 2,000 rpm. The greatest brake thermal efficiency on M100 fuel was 36.3 percent at WOT, lean operation (Lambda = 1.3), and 2,000 rpm. Similarly, the highest efficiency at 1,500 rpm was also noted at this same operating point.

Appendix C presents similar data when operating on $2H_2/CO$ fuel with the stock and modified intake camshafts. Again, the limited amount of fuel in the gas cylinders allowed for data collection at only a few test points (indicated by percent throttle opening). The overall goals of operating on $2H_2/CO$ fuel at higher efficiencies and torque were realized at low load operation. This, however, was not the case at higher loads. Operating on $2H_2/CO$ fuel while producing lower levels of CO emissions were more difficult to attain. Because of the CO content in the fuel itself, operating on dissociated methanol always produced more CO emissions than M100 at the same operating points. However, through the recent engine and fuel system modifications described previously, CO levels produced during $2H_2/CO$ fuel operation here were much lower than those produced anytime previously.[1]

In the prior interim report,[1] the combustion process was monitored using an in-cylinder pressure transducer. Previously, engine operation on dissociated methanol fuel was smoother and offered less fluctuation in maximum cylinder pressure values (a more stable combustion process). It was again attempted to monitor this data, however, error and resonance in the pressure transducer did not allow for accurate data presentation.

VI. Future Efforts

A 100 percent efficient methanol dissociator is currently not available. The initial goal of this engine conversion project was to convert an M100-fueled engine to operation on simulated dissociated methanol fuel. The engine is currently able to operate on $2H_2/CO$ fuel at higher power output and greater thermal efficiency than when fueled with M100 at certain operating conditions. EPA does not plan any additional engine testing until an acceptable methanol dissociator becomes available.

VII. 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 Jennifer Criss and Mae Gillespie of the Technology Development Group for word processing support.

VIII. References

1. "Conversion Of Methanol-Fueled 16-valve, 4-Cylinder Engine To Operation On Gaseous $2H_2/CO$ Fuel-Interim Report IV," Schaefer, R. M., et al., EPA/AA/TDG/92-06, September 1992.

2. "Conversion Of Methanol-Fueled 16-Valve, 4-Cylinder Engine To Operation On Gaseous $2H_2/CO$ Fuel-Interim Report II," Piotrowski, G. K. and J. Martin, EPA/AA/CTAB/89-02, March 1989.

3. "Resistively Heated Methanol Dissociator For Engine Cold Start Assist-Interim Report," Piotrowski, G. K., EPA/AA/CTAB/88-02, March 1988.

4. "Conversion of Methanol-Fueled 16-Valve, 4-Cylinder Engine To Operation On Gaseous $2H_2/CO$ Fuel-Interim Report III," Schaefer, R. M. et al., EPA/AA/CTAB/91-01, April 1991.

5. "Conversion of Methanol-Fueled 16-Valve, 4-Cylinder Engine to Operation On Gaseous $2H_2/CO$ Fuel-Interim Report," Piotrowski, G. K., EPA/AA/CTAB/88-06, June 1988.

6. The Internal-Combustion Engine in Theory and Practice, Volume 1, Taylor, C. F., The M.I.T. Press, 1985.

APPENDIX A

TEST ENGINE SPECIFICATIONS, M100 FUEL OPERATION CONDITION AS LOAN 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

APPENDIX B

M100 TEST RESULTS

2,000 RPM Conditions							
		g/BHP-hr					
Lambda	Torque (ft-lb)	BSHC	BSCO	BSNOx	BSCO	Effic. (%)	BHP
Wide-Open Throttle:							
0.7	103	10.37	229.7	3.9	399	28.2	39.2
1.0	94	0.45	37.5	9.1	448	35.2	35.8
1.3	78	0.64	3.9	4.6	466	36.3	29.7
75 Percent Throttle Opening:							
0.7	62	15.64	244.6	0.9	404	23.2	23.6
1.1	58	1.42	22.0	6.6	557	31.9	22.1
1.4	40	1.62	3.5	1.2	589	27.9	15.2
50 Percent Throttle Opening:							
0.7	44	2.66	203.3	0.2	404	22.0	16.8
1.1	36	0.41	6.3	3.6	599	26.7	13.7
1.4	22	3.63	6.9	1.0	776	21.0	8.4
25 Percent Throttle Opening:							
0.7	6	8.64	346.2	0.8	1595	6.2	2.3
1.0	6	1.32	22.6	1.2	1762	8.4	2.3
1.4	0	NA	NA	NA	NA	0.0	0.0
NA = Not available							

M100 TEST RESULTS

1,500 RPM Conditions							
		g/BHP-hr					
Lambda	**Torque (ftlb)**	**BSHC**	**BSCO**	**BSNOx**	**BSCO$_2$**	**Effic. (%)**	**BHP**
Wide-Open Throttle:							
0.7	93	25.35	133.8	5.5	428	24.7	26.6
0.1	85	2.88	16.9	10.1	457	30.7	24.3
1.4	69	1.18	2.7	3.2	461	35.0	19.7
75 Percent Throttle Opening:							
0.7	62	26.44	140.3	1.9	428	22.1	17.7
1.0	58	1.43	16.6	7.2	510	28.3	16.6
1.4	40	1.40	2.4	1.9	561	27.3	11.4
50 Percent Throttle Opening:							
0.7	45	3.87	147.9	0.3	393	21.8	12.9
1.0	37	1.08	7.2	4.2	576	24.4	10.6
1.4	25	1.87	1.8	0.9	536	23.0	7.1
25 Percent Throttle Opening:							
0.7	10	5.01	168.1	0.4	851	9.8	2.9
1.0	11	0.94	14.1	1.2	1024	14.3	3.1
1.4	2	16.60	15.0	1.9	4177	3.9	0.6

APPENDIX C

2H$_2$/CO TEST RESULTS

Stock Intake Comshaft Testing							
%WOT	Torque (ftlb)	g/BHP-hr				Effic (%)	BHP
		BSHC	BSCO	BSNOx	BSCO$_2$		
2,000 rpm							
25	12	0.22	445.0	0.2	524	11.0	4.6
50	36	0.07	288.0	0.8	415	16.6	13.7
53	42	0.02	284.9	0.2	394	19.6	16.0
56	42	0.04	283.3	0.4	443	17.0	16.0
1,500 rpm							
33	28	0.01	308.2	0.1	336	16.8	8.0
59	48	0.04	269.5	0.7	394	16.4	13.7
100	44	0.03	285.1	0.2	358	8.9	12.6

Modified Intake Camshaft Testing								
Lambda	%WOT	Torque (ftlb)	g/BHP-hr				Effic (%)	BHP
			BSHC	BSCO	BSNOx	BSCO$_2$		
2,000 rpm								
1.1	46	29	0.03	317.9	17.1	676	21.4	10.9
0.9	66	36	0.02	241.3	13.1	624	15.3	13.7
1,500 rpm								
1.1	32	18	0.06	485.2	4.6	688	16.6	5.1
0.8	74	43	0.05	253.1	21.7	618	12.8	12.3