EPA/AA/CTAB-91-01

Technical Report

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

by

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Technical Reports do not necessarily represent final EPA decisions or positions. They are intended to present technical analysis of issues using data which are currently available. The purpose in the release of such reports is to facilitate the exchange of technical information and to inform the public of technical developments which may form the basis for a final EPA decision, position or regulatory action.

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UNITED STATES ENVIRONMENTAL PROTECTION AGENCY

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OFFICE OF
AIR AND RADIATION

MEMORANDUM

SUBJECT: Exemption From Peer and Administrative Review

FROM: Karl H. Hellman, Chief KH
Control Technology and Applications Branch

TO: Charles L. Gray, Jr., Director
Emission Control 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 III," (EPA/AA/CTAB/91-01) 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 has been operated on both M100 and simulated dissociated methanol (hydrogen and carbon monoxide) gaseous fuels. This report describes modifications made to the engine and summarizes the results of recent testing. Further work on this project will be described in a future technical report.

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: ______________________ Date: 5-6-91
Charles L. Gray, Jr., Dir., ECTD

Nonconcurrence: ______________________ Date: __________
Charles L. Gray, Jr., Dir., ECTD

cc: E. Burger, ECTD

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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 to CO. This engine has been used to investigate the difference in emission levels and lean limit operation between two different fuels: M100 neat methanol and simulated dissociated methanol gaseous fuel ($2H_2/CO$).

The work described in this report contains results of recent emission testing begun in July 1990 with the engine equipped with stock and modified intake camshafts. These results are compared to results obtained during previous testing with the modified camshaft. This previous work was completed in March 1989 and is described in EPA/AA/CTAB-89-02. [1] The present report also contains a summary of fuel system/engine experiments conducted to more accurately measure gaseous fuel flow to the test engine and increase the power output of the engine.

Table 1 is a summary of our test results. The first entry refers to testing conducted in March 1989. Previously, the test engine was operated at lean conditions with the $2H_2/CO$ fuel. Under load, the maximum output torque achieved using the gaseous fuel was 26.6 ft-lbs at 2,000 rpm and an A/F ratio of 11.9:1. (The stoichiometric A/F ratio for operation on the gaseous fuel is the same as it is for M100 operation, 6.4:1.)

Recent testing consisted of the engine operating under load at wide-open throttle (WOT), fueled with gaseous $2H_2/CO$, and with a spark timing of 0° before top dead center (BTDC). Spark timing was limited to near BTDC because severe engine backfire or knock resulted when spark timing was advanced. Separate tests were performed with the engine equipped with both modified and stock intake camshafts. Fuel flowrate here was calculated from the change in weight of the gas bottle during each test, the density of the fuel, and the time duration of each test. A meter which measured the remainder of the gas bottle contents in standard cubic feet was also used.

During testing with the modified camshaft, an output torque of 60 ft-lbs at an A/F of 7.89:1, under 2,000 rpm, WOT conditions was measured. This torque was more than twice the highest value measured during the previous testing performed under both WOT/throttled and lean operating conditions.

Output torque increased during engine warmup. When the engine was cold started, the output torque was usually 15 ft-lbs below the maximum value recorded later. The torque reached its maximum value after approximately 5 minutes into each test. It remained constant at that value for the remainder of the test until the engine was shut off.

Table 1

Summary of Test Results
Nissan CA18DE Engine, $2H_2/CO$ Fuel

Test Number	Date	Camshaft	RPM	Torque ft-lbs	BHP	Air/Fuel
1	03/01/89	Modified	2,000	27	10.30	11.90
2	10/30/90	Modified	2,000	55	20.94	8.33
3	12/04/90	Modified	2,000	60	22.85	7.89
4	01/28/91	Stock	2,000	65	24.75	N/A
5	01/30/91	Stock	2,000	65	24.75	N/A
6	02/20/91	Stock	1,750	67	22.32	6.07
7	02/21/91	Stock	1,750	66	21.99	7.36
8	02/25/91	Stock	2,000	70	26.28	N/A

N/A Not available.

Note: All tests at WOT; torque values are maximum.

Approximately 13 percent of the fuel was unburned and passed through the exhaust with the modified camshaft. The stock intake camshaft was then placed in the test engine to enhance mixing of the air/fuel charge in the chamber prior to combustion. The result was the highest torque measured on a single test (70 ft-lbs) under the same 2,000 rpm, WOT operating conditions. The engine ran much smoother with the stock camshaft at these same conditions, with no fluctuation in torque during each entire test. Higher amounts of unburned fuel at 1,750 rpm (41.7 percent was not burned) under richer conditions were noted.

The 70 ft-lbs torque value obtained here is approximately 38 percent below the maximum value (110 ft-lbs) obtained by Nissan with an M100-fueled, 12.0 compression ratio CA18DE engine. However, with recent engine/fuel system modifications, the power output has been increased 160 percent when compared to results obtained from previous testing with $2H_2/CO$ fuel.[1]

A direct comparison of emissions test results from the engine when it is alternately fueled with M100 and $2H_2/CO$ fuels is not possible at this time. The previous testing with the M100 fuel utilized a catalytic converter in the exhaust stream while $2H_2/CO$ fuel results are engine-out emissions. Also, the limited amount of gaseous fuel in the T-cylinder storage bottles did not permit starting and warming to true steady-state conditions prior to emission testing.

Higher emissions of CO and NOx were measured during this testing. High CO emissions may result from the substantial amount of unburned fuel passing out of the exhaust. H_2 in the exhaust was also measured and was consistently twice the CO levels, this proportion being the same as that of the components in the gaseous fuel.

Future efforts will utilize a recently acquired mass flow controller to accurately control and measure fuel flow. The large amounts of unburned fuel in the exhaust will be investigated. An in-cylinder pressure sensor will also be used to investigate the combustion event.

II. Introduction

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.[1,2]

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, a stock model modified by Nissan Motor Corporation for use with liquid methanol, was loaned to EPA by Nissan for use in alternative fuels research. This report summarizes the most recent EPA efforts to investigate dissociated methanol as an automotive fuel with this engine.

III. Description of Test Engine

The base engine used for this project was a Nissan CA18DE engine. The stock engine is an in-line, 4-cylinder, 1.8-liter capacity powerplant. 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 two overhead camshafts, one each for the intake and exhaust sides.

The test engine was 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.[1] A summary of them is included in this report in Appendix A.

EPA also modified the engine to use simulated dissociated methanol fuel (66 volume percent H_2 and 33 volume percent CO). The most significant modification made here was the replacement of the intake camshaft with a specialty camshaft. The specialty cam admits air only through one intake valve and gaseous fuel only through the second valve. These EPA modifications were also discussed in a previous report [1] and are detailed here in Appendix B.

EPA installed, at Nissan's request, a thicker head gasket for the testing mentioned in this report. This thicker head gasket raised the clearance between the valve face and the piston crown; this modification was made to improve the durability of the engine. The effect of this modification was to lower the compression ratio from 11.0 to 10.5. The current testing referred to in this report made use of this modification.

IV. Exhaust Analysis

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 40 feet.

A gaseous sample line and electronic ties have 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 is accomplished at a bank of analyzers located in another test cell. Emissions measured as hydrocarbons (HC) are measured on Beckman model 400 flame ionization detector (FID). NOx level determination is conducted on a Beckman model 951 chemiluminescent NO/NOx analyzer. CO is measured by infrared technique using a Horiba model A1A23 infrared analyzer. Hydrogen gas in the exhaust was measured using a Gow-Mac model 550 gas chromatograph calibrated with a 40 percent hydrogen span gas.

V. Recent Engine/Fuel System Modifications

A significant period of time, in excess of one year, had elapsed from the date of the last report [1] to the start of the testing commented on here. It was necessary, therefore, to first perform leak checks on the entire fuel system and a leakdown check on the engine cylinders to determine their integrity. No leaks were detected when the fuel system was pressurized, and the greatest loss of pressure in a cylinder was only 2 percent, indicating that the compression had not significantly deteriorated.

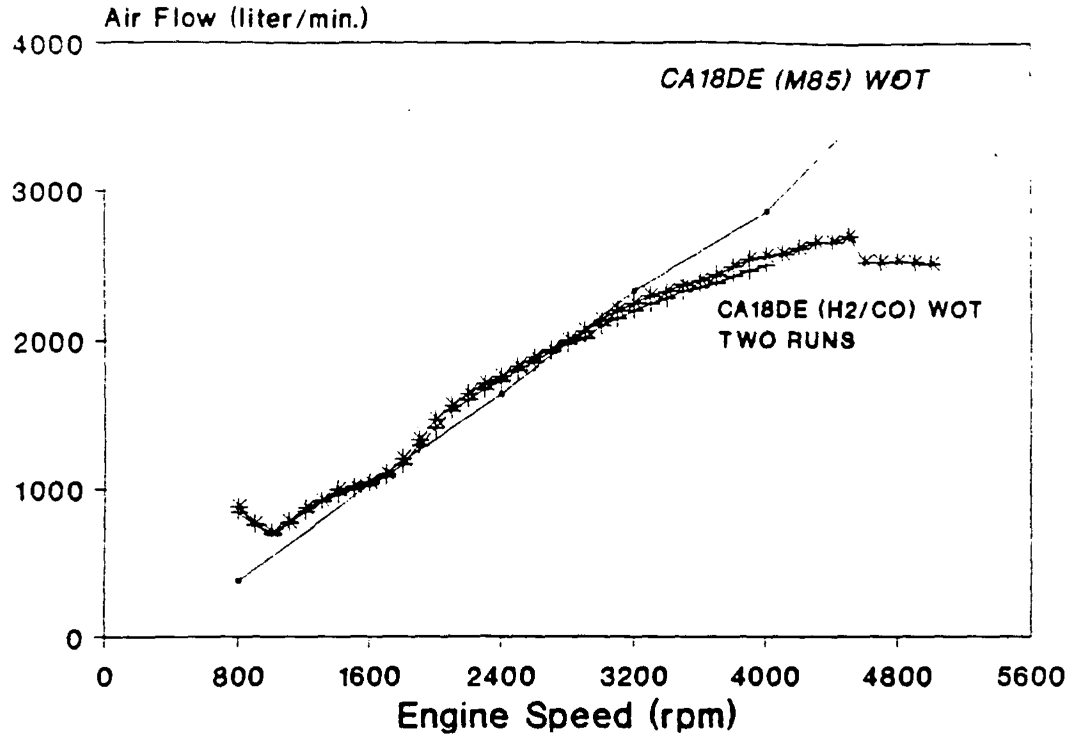
Next, the fuel valve lifters were removed and the fuel system was pressurized to 60 psi using bottled nitrogen. The engine was then motored by the dynamometer. No flow in the flowmeter was observed, indicating no leakage in the fuel delivery system.

When the modified intake camshaft was installed, gaps were noted in the gasket that mates the swirl control valve housing to the engine head. These gaps made possible the transfer of fuel and air between the separated fuel and air intakes in each runner. These gaps were plugged to limit inaccuracies in air/fuel flowrate measurement.

Air flowrates through the engine at WOT conditions over a range of engine speeds were determined and are plotted in Figure 1. Air flowrates over the same conditions with the engine as modified by Nissan for M85 liquid fuel use are also plotted for comparison. (The air intake passages were significantly modified when the special intake cam was added to restrict air flow to one of the intake valves in each cylinder.) The objective of these measurements was to determine the effect of the restriction on air flow.

Figure 1

Air Flow Rate



The two sets of data follow a roughly similar trace until approximately 3,000 rpm, where restriction of air flow to only one runner may have significantly affected the volumetric efficiency. However, for the engine speed range 800-2,500 rpm the air flowrates are similar. In this range, volumetric efficiency values are approximately 80 percent.

The Dwyer 10 cfm rotameter used during the previous testing may have provided inaccurate fuel flowrate measurements as it was not calibrated for the gaseous fuel. With varying engine speeds at constant air flowrates, the rotameter indicated only slight differences in fuel flowrates. During one test, an audible leak of the gaseous fuel through the rotameter control valve was detected. A less-audible leak may not have been detected as the result of background noise from the ventilation system in the test cell.

Other fuel flowrate measurement devices were tried, however, none of these methods provided consistently accurate measurements. To date, three different fuel measurement devices have been used when operating the engine on the gaseous fuel (rotameter, electronic/calibrated orifice metering system, and a dry gas meter). Each should have been capable of measuring fuel flow into the engine. However, in each case the methods provided conflicting flowrate information when the engine was operated under steady-state conditions.

One indirect check of fuel consumption for this gaseous fuel system is to use the change in mass of the cylindrical fuel tank as a measure of fuel flowrate. The fuel tank was weighed immediately prior to and following an engine test; the change in fuel tank mass indicates fuel consumed. The engine was brought to test conditions immediately after start, and these conditions remain unchanged during testing. This is not quite steady-state testing due to the warmup of the engine that occurs after start. The time duration of the test was recorded, and the mass of fuel consumed was converted to standard cubic feet of gas. This method, used in the testing discussed below, provide accurate, consistent measurements of gaseous fuel flowrate.

A scaled pressure gauge which correlates bottle pressure with standard cubic feet of gas present in the cylinder was also used as a check on the weighing method referred to above.

Finally, it was necessary to vent the crankcase to the test cell ventilation system. This action was taken as the result of a severe oil leak at the oil filter caused by overpressure in the crankcase. This situation was alleviated by this rerouting of the PCV system.

VI. Discussion of Test Results

All of the testing described here was performed using $2H_2/CO$ bottled gas, simulating dissociated methanol fuel. Table 2 is a summary of the emission results. Test numbers here correspond to the same test numbers in Table 1. In test numbers 5 and 7, two bag samples were taken during one test run, hence the a and b distinctions.

The first entry in Table 2 refers to testing conducted during March 1989. Air/fuel ratios at that time were calculated to be 11.9:1, very lean of stoichiometric. This value may be high; subsequent experiments at similar conditions have suggested that the rotameter supplied fuel flowrates were too low. Only 27 ft-lbs of brake torque were generated under these conditions.

The remainder of the tests described in Table 2 were conducted as part of the recently completed testing. The head gasket, which reduced the compression ratio to 10.5, from 11.0 previously, was in place for this testing. The rotameter which previously measured fuel flowrate was also removed from the fuel circuit; any backpressure therefore caused by the meter was removed. Test numbers 2 and 3 were conducted with the modified intake camshaft in place, as in test number 1. At WOT conditions, torque increased to 55 ft-lbs, a considerable gain over the value noted previously. Fuel flowrate was calculated by running at a single set of conditions and noting the change in the mass of the fuel bottle and the duration of the experiment. H_2 and CO emission concentrations were measured. Because intake and dilution air was also metered, the rate of unburned fuel could be calculated. The rate of unburned fuel was calculated to be approximately 13 percent.

Table 2

Emission Test Results
Nissan CA18DE Engine, $2H_2/CO$ Fuel

Test Number	Torque (ft-lbs)	UBF (%)	A/F	Brake Specific Emissions (g/BHP-hr) HC	CO	CO2	NOx
1	27	N/A	11.90	0.130	2.27	426	8.57
2	55	13.4	8.33	0.011	29.23	349	0.79
3	60	13.8	7.89	0.008	33.62	349	6.69
4	65	N/A	N/A	0.004	79.66	455	10.87
5a	65	N/A	N/A	0.007	57.25	376	9.33
5b	65	N/A	N/A	0.006	78.09	355	6.96
6	67	24.8	6.07	0.004	76.37	261	4.09
7a	66	41.7	7.36	0.006	125.40	256	2.35
7b	66	41.7	7.36	0.005	119.20	239	2.24
8	70	N/A	N/A	0.015	111.90	452	7.64

N/A Not available.

a,b Two bag samples for one test run.

Several other experiments were conducted in order to accurately determine the fuel flowrate. Only H_2 and CO emissions were measured during these experiments; the values measured during that work were similar to those given to the second and third tests in Table 2.

Spark ignition was timed at 0 degrees before top dead center (BTDC). Advancing the spark even slightly caused an audible knocking condition. At 2,000 rpm, part throttle also caused an audible knock. It was not possible to vary fuel flowrate with a high degree of accuracy using the current fuel system.

The modified intake cam was then replaced with the stock camshaft and hydraulic lifter system. This camshaft would enable fuel and air to mix in the cylinder earlier, as the stock camshaft permits the introduction of fuel earlier in the combustion cycle. The stock control slide housing, however, did not replace the housing modified to separate the valves into fuel and air admission passages. Fuel continued to be admitted through a single valve while air only was admitted through the other valve.

Tests 4 through 5b in Table 2 were conducted at 2,000 rpm, WOT conditions with the stock intake camshaft. The admission of fuel was accomplished in the same manner as in previous experiments, and was not closely controlled.

The higher cam profile and extended valve open period caused a richer mixture, richer than stoichiometric. Torque increased to 65-70 ft-lbs at the richer condition. This maximum value was still below the value of 110 ft-lbs experienced by Nissan with an M100-fueled, 12.0 compression ratio CA18DE engine. No backfire was experienced at WOT. The amount of fuel unburned was unknown because of a calibration problem with the engine air sensor.

NOx emission levels were similar in magnitude between the testing conducted with the modified camshaft and the stock camshaft. CO emissions, a component of unburned fuel, more than doubled to approximately 75 g/BHP-hr, with the stock camshaft.

Engine speed was then reduced to 1,750 rpm at WOT conditions. Brake torque was measured at approximately 67 ft-lbs, similar to the values recorded at 2,000 rpm. Air/fuel ratio went slightly lean during this testing to approximately 7.3:1. Fuel efficiency decreased significantly, however, as unburned fuel rose sharply to a calculated value of 40 percent. CO levels, representing unburned fuel, rose sharply as engine speed was reduced. H_2 was measured at roughly twice the volumetric concentration of CO during this testing.

A pressure sensor was installed in the spark plug well of the number 1 cylinder in the test engine. An attempt was made to monitor cylinder pressure during the combustion event and use this in an attempt to relate the pressure pattern to the onset of lean misfire. It was hoped that the shape of the pressure versus crank angle curve would provide information concerning abnormal combustion occurrences. The pressure sensor ultimately failed, however, due to pressure wave resonation in the cavity which housed the sensor.

It is not yet possible to determine to what extent air/fuel mixing within the cylinder is a problem and whether the lower torque values are associated with mixing. At WOT, in all cases, when fuel flowrate was reduced, output torque at constant engine speed was also reduced. For air/fuel ratios in excess of 6.4:1, enough air should have been present to complete the combustion of the fuel present.

VII. Highlights From Current Testing

1. Restricting the air flow to the engine to one of the two intake valves per cylinder did not appreciably affect the maximum air flowrate possible in the engine speed range 800-2,500 rpm at WOT conditions.

2. The rotameter used to measure fuel flowrate was eliminated from the fuel system, and the compression ratio was reduced to 10.5 through the installation of a new head gasket. At 2,000 rpm engine speed, WOT conditions brake torque was measured at 55-60 ft-lbs. This was a considerable increase above the levels measured previously at these conditions.

3. The testing mentioned in 2. above, was conducted at an air/fuel ratio of approximately 8.0:1 (lean of stoichiometric). Thirteen percent of the gaseous $2H_2/CO$ fuel was passed through the exhaust as unburned fuel during this testing. CO emissions, an indicator of unburned fuel, increased sharply during this testing, to approximately 30 g/BHP-hr.

4. Testing was also conducted with the stock intake camshaft in place of the specialty camshaft. At 2,000 rpm, WOT conditions, brake torque increased slightly to 65 ft-lbs. The amount of CO in the exhaust more than doubled, from levels measured with the specialty camshaft.

When the engine speed was reduced and held constant at 1,750 rpm the percentage of unburned fuel rose sharply to about 40 percent. Brake torque rose only slightly to 67 ft-lbs. CO (unburned fuel) increased to levels exceeding 100 g/BHP-hr.

VIII. Future Efforts

This engine conversion project was begun to develop a suitable test bed for a practical, onboard methanol dissociation system. Further development of this engine concept will be structured to accommodate this goal. Immediate plans concern development of two measures of engine performance:

1. Emissions/fuel economy; and

2. Engine performance at lean operating conditions.

Further emissions testing at various engine speed/load operating conditions will be conducted to characterize the emissions profile of this engine when operated on both M100 and dissociated methanol. A/F ratio at these various test points will also be determined. A/F ratios will now be controlled by varying fuel flowrate to the engine with the use of a Tylan General Corporation mass flow controller. This equipment will enable accurate metering and control of the gaseous fuel.

Previous testing with M100 liquid methanol was conducted with a catalytic converter in the exhaust stream. Testing will be conducted with the catalytic converter removed in order to provide engine-out emissions data with M100 fuel.

Spark timing was limited to near TDC in these tests by knock. Spark timing will continue to be adjusted as air/fuel conditions are changed with the new fuel controller in order to better utilize this operating parameter.

One way to determine the proximity to the lean misfire limit at various engine operating conditions is to obtain a quantifiable measure of increasing engine roughness as the air/fuel mixture is leaned out. A measure of proximity to lean misfire limit may be obtained directly, through measurement of changes in cylinder pressure during the combustion effort. An indirect method might involve the measurement of the variability in successive crank rotation times as leanness increases. The test engine is not equipped with a knock sensor; it should therefore be possible to obtain a quantifiable measure of engine performance as the lean misfire limit is approached when the engine is fueled with the gaseous $2H_2/CO$ blend.

EPA has obtained a cylinder pressure transducer less susceptible to the resonation problem experienced with the earlier generation sensor used in the present work. This sensor will be used in an effort to optimize the combustion event.

IX. Acknowledgments

The CA18DE test 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 A. Criss and Leslie A. Cribbins of CTAB/ECTD for typing, formatting, and editing this report.

X. References

1. "Conversion of Methanol-Fueled 16-Valve, 4-Cylinder Engine to Operation on Gaseous $2H_2/CO$ Fuel - Interim Report II," Piotrowski, Gregory K., James Martin, EPA/AA/CTAB-89-02, March 1989.

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

3. "Internal Combustion Engines and Air Pollution," Obert, E. F., Harper and Row, New York, NY, 1973.

4. "Standards for Emissions From Methanol-Fueled Motor Vehicles and Motor Vehicle Engines: Final Rule," Federal Register, U.S. Environmental Protection Agency, April 11, 1989.

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	11.0, 10.5*
Compression pressure	16.5 kg/square cm (350 rpm, 80 degrees Celsius)
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 degrees BTDC to 54 degrees BTDC by means of an external control

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* Reduced to 10.5 for testing referred to in this paper.

APPENDIX B

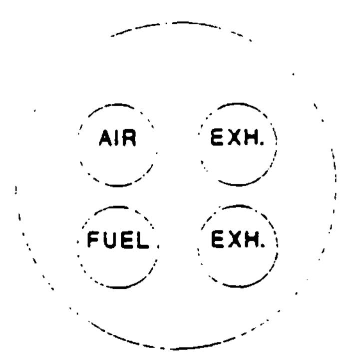
PREVIOUS ENGINE/FUEL SYSTEM MODIFICATIONS FOR $2H_2/CO$ FUEL OPERATION

The simulated dissociated methanol product gas used in this work is 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 $2H_2/CO$.

The Nissan CA18DE engine utilizes a 4-valve per cylinder valvetrain configuration; both the stock gasoline and M100 methanol modified versions utilize 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 gaseous fuel. The exhaust-side valve scheme was not modified (Figure 1).

Figure 1

Valve Scheme
$2H_2/CO$ Fuel Conversion
<u>Nissan CA18DE Engine</u>



VALVE SCHEME

2H2/CO FUEL CONVERSION

NISSAN CA18DE ENGINE

APPENDIX B (CONT'D)

PREVIOUS ENGINE/FUEL SYSTEM MODIFICATIONS FOR $2H_2/CO$ FUEL OPERATION

The advantages of structuring the intake process this way are threefold. First, air flow into the engine may be less restricted if the fuel, already in the gaseous state, is introduced into only one of the intake runners. Second, there may be less chance of flashback and a resulting manifold ignition if fuel exclusively, and not a combustible fuel/air mixture, is introduced at an intake valve. Finally, fuel may enter the combustion chamber at the designer's discretion, rather than at the same time the air needed for combustion is admitted.

It was necessary to alter the fuel and air intake system in order to allow 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 liquid-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. The control valve slide and actuator were disassembled and the swirl control valves removed. The runners through the valve assembly that contained wells for fuel injectors were welded shut approximately 1/2-inch upstream from the well holes. These seals prevent 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. A metal impregnation technique was used to seal the holes. The sealed holes were then coated with a layer of epoxy.

Fuel injectors are not used to feed the gaseous state 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 wells were then threaded to accept the fittings.

APPENDIX B (CONT'D)

PREVIOUS ENGINE/FUEL SYSTEM MODIFICATIONS FOR $2H_2/CO$ FUEL OPERATION

The $2H_2/CO$ fuel is a gaseous blend with a composition of 67 and 33 volume percent H_2 and CO respectively. This fuel is stored in compressed gas cylinders (T-sized) at 2,000 psig. A fuel supply cylinder is located outside the test cell, approximately 5 feet from the cell wall during testing. The bottle, fitted with a regulator and pressure gauge, is opened by a hand valve prior to testing. The fuel line from the bottle is 1/4-inch stainless steel tubing, approximately 22 feet in length from bottle to cell wall.

The stainless steel fuel line enters the cell through a hole drilled through the concrete wall. A Gould electrically controlled solenoid valve is located in the line immediately after the wall. An electrical signal from the control room controls the opening of the valve.

The fuel line from the cell wall to a fuel flow regulator measures approximately 54-1/2 feet. This regulator is a Twin Bay Model TB-100. Gas flow through this regulator is controlled by a flexible diaphragm. The diaphragm is opened proportionally to the pressure exerted by a stream of air provided by a tank of compressed air; the pressure exerted by this airstream is controlled by a valve located in the cell control room.

The final stage of the fuel system supplies the gaseous fuel to the combustion chamber ports. The fuel passes to a cylindrical plenum, this plenum serving as a header to four flexible fuel lines. Inserted in each of the four fuel lines approximately 17 inches from each cylinder is a 2-stage H_2 flame arrestor. The fuel lines are connected to threaded fittings which are screwed into the fuel injection ports in the valve control assembly. The $2H_2/CO$ fuel is supplied to the combustion chambers by the opening of the fuel valves.

APPENDIX C

AIR/FUEL RATIO CALCULATION WITH $2H_2/CO$ FUEL

Given:

Air flowrate, 29 standard cubic feet/minute (scfm)
Change in weight of gas bottle, 4.5 lbs
Density of fuel, 0.02969 lbs/cubic feet
Time of test, 10.0 minutes (min)

Fuel flowrate = (4.5 lbs)/(0.02969 lbs/cubic feet)(10.0 min)

Fuel flowrate = 15.16 scfm

Air/fuel ratio is defined [3] as:

$$\frac{\text{Mass flowrate of air}}{\text{Mass flowrate of fuel}}\text{, dimensionless} \qquad (1)$$

Molecular weight of air, 28.89, approximately

Calculate molecular weight of fuel, $2H_2/CO$:

2/3 (molecular weight of H_2, 2) = 1.333

1/3 (molecular weight of CO, 28) = <u>9.333</u>

Molecular weight of fuel, approx. 10.666

At standard conditions for gases:

$$PV = nRT \qquad (2)$$

Where:

P = pressure (atmosphere)

V = volume (cubic feet)

n = pound moles

R = gas constant (0.7302 atm-cubic feet/lb mol-degrees R)

T = Temperature (degrees R)

At standard conditions:

T = 492 degrees R

P = 1 atmosphere

APPENDIX C (CONT'D)

AIR/FUEL RATIO CALCULATION
WITH $2H_2/CO$ FUEL

Mass flowrate may then be defined as:

(PVt/RT)(molecular weight) = nt(molecular weight) (3)

Where:

Vt = cubic feet/minute

nt = lb moles/minute

Calculate A/F ratio:

Mass flowrate of air from (3)

(1 atm)(29 scfm)(28.89 lb/lb mol)/(0.7302)(492 degrees R)

= 2.332 lb/minute of air

Mass flowrate of fuel using (3)

(1 atm)(15.16 scfm)(10.67 lb/lb mol)/(0.7302)(492 degrees R)

= 0.4501 lb/minute of fuel

From (1):

A/F = (2.332 lb/minute air)/(0.4501 lb/minute fuel)

A/F = 5.18

APPENDIX D

EXHAUST MASS EMISSION CALCULATION WITH $2H_2/CO$ FUEL

The procedure for calculating mass emissions for methanol-fueled light-duty engines is defined in the Federal Register,[4] and that same method was utilized in this report.

From exhaust gas bag analysis, you are given the following dilute concentrations: HCe, COe, CO2e, and NOxe. It is now necessary to correct for background concentrations. All concentrations are in parts per million (ppm) except for CO_2 concentrations which are in a percentage (percent). This is done with the use of the following formula.

$$Yconc = Yex - Yb(1 - 1/DF) \qquad (4)$$

Where:

Yconc = emission concentrations corrected for background

Yex = dilute concentrations measured in bag samples

Yb = background concentrations

DF = dilution factor.

The values used for the background concentrations of each emission were:

HCb = 1.0 ppm

COb = 0.0 ppm

CO_2b = 0.04 percent

NOxb = 0.0 ppm

The dilution factor is calculated by:

$$DF = 100[x+y/2+3.76(x+y/4-z/2]/CO2ex+(HCex+COex)0.0001 \qquad (5)$$

This formula is provided for methanol-fueled vehicles where fuel composition is CxHyOz. Therefore, for our fuel (2H2/CO), x = 1, y = 4, z = 1.

APPENDIX D (CONT'D)

EXHAUST MASS EMISSION CALCULATION
WITH $2H_2/CO$ FUEL

Once each emission is corrected for background concentrations, it is now possible to calculate the mass of each emission contained in the bag sample with the following formula:

$$Ymass = Vmix \times Dy \times (Yconc/1{,}000{,}000) \tag{6}$$

Where:

Ymass = mass value of each emission, grams

Vmix = total dilute exhaust volume, cubic feet

Dy = density of each emission, grams/cubic feet

Yconc = corrected concentration of each emission, ppm.

The densities of each emission are also listed in the Federal Register. When finding the correct mass of NOx, there is an additional NOx factor (Kh) that must also be applied to equation 6. This value was calculated as follows:

$$Kh = 1/[1 - 0.0047(H - 75)] \tag{7}$$

Where:

Kh = NOx factor

H = Absolute humidity, grams of water per kilogram of dry air.

The NOx factor used for all calculations was 0.9.

Once the mass of each emission in the bag sample is known, it is now possible to calculate a brake specific emission value so that direct comparisons of different tests could be made. First the brake horsepower needs to be calculated by:

$$BhP = Tn/5252.1 \tag{8}$$

Where:

BhP = brake horsepower

T = engine output torque, ft-lbs

n = engine speed, rpm.

APPENDIX D (CONT'D)

EXHAUST MASS EMISSION CALCULATION WITH $2H_2/CO$ FUEL

By knowing the time of each bag sample, brake specific emission values can be found by:

$$BSY = Ymass/(BhP)(t) \tag{9}$$

Where:

BSY = brake specific emissions, grams/BhP-hour

Ymass = mass value of each emission, grams

BhP = brake horsepower

t = time of emission bag sample, hours.