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Evaluation of Dissociated and Steam-Reformed Methanol as Automotive Engine Fuels

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ABS:  Dissociated and steam reformed methanol were evaluated as automotive
      engine fuels. Advantages and disadvantages in using methanol in the
      reformed rather than liquid state were discussed. Engine dynamometer tests
      were conducted with a four cylinder, 2.3 liter, spark ignition automotive
      engine to determine performance and emission characteristics operating on
      simulated dissociated and steam reformed methanol (2H2 + CO and 3H2 + CO2
      respectively), and liquid methanol. Results are presented for engine
      performance and emissions as functions of equivalence ratio, at various
      throttle settings and engine speeds. Operation on dissociated and steam
      reformed methanol was characterized by flashback (violent propagation of a
      flame into the intake manifold) which limited operation to lower power
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Evaluation of Dissociated and Steam-Reformed Methanol as Automotive Engine Fuels

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NOMENCLATURE

A/F	Air-fuel ratio
Bhp	Brake horsepower
Bmep	Brake mean effective pressure
Bsfc	Brake specific fuel consumption
BTDC	Before top dead center
Bte	Brake thermal efficiency
C	Carbon
CO	Carbon monoxide
CO_2	Carbon dioxide
CR	Compression ratio
EGR	Exhaust gas recirculation
FID	Flame ionization detector
gm	Grams
HC	Hydrocarbon
HCHO	Formaldehyde
Ihp	Indicated horsepower
in. Hg	Inches of mercury
lbf	Pounds force
lbm	Pounds mass
LFE	Laminar flow element
LHV	Lower heating value
LMF	Liquid mass flowmeter
$\dot{m}$	Mass flow rate
M	Molecular weight
MBT	Minimum spark advance for best torque
NDIR	Non-dispersive infrared

NOMENCLATURE

(cont'd)

NO_x	Oxides of nitrogen
O_2	Oxygen
ΔP	Pressure drop
PCV	Positive crankcase ventilation
Rpm	Revolutions per minute
SA	Spark advance (BTDC)
SI	Spark ignition
T	Temperature
WOT	Wide open throttle
X	Mole fraction
ϕ	Equivalence ratio
μ	Absolute viscosity
ρ	Density

1. INTRODUCTION

Recently, there has been considerable interest in alternatives to gasoline as automotive fuels (refs. 1-11). Methanol has received attention as an alternate fuel for the dwindling supply of petroleum based fuels because it can be synthesized from coal, oil shale, tar sands, and renewable resources (ref. 12). Also, methanol could be distributed through the present fuel distribution network with some minor modifications. Hydrogen has also been investigated as an automotive fuel because of its potential for efficient combustion and low exhaust emissions (refs. 1, 2, 13, 14). The two fuels are linked in that methanol can be catalytically reformed at relatively low temperatures to yield a hydrogen-rich gas (refs. 13-18). Thus, methanol can be used as a liquid or as a hydrogen-rich reformed gas. Further, the conbined use of liquid and reformed methanol in a single system may be possible (but was not investigated in this study).

Two means of catalytically reforming methanol are: dissociation and steam reforming. Dissociation occurs according to the chemical reaction:

$$CH_3OH \rightleftharpoons 2H_2 + CO.$$

The steam reforming reaction equation for equal molar percentages of water and methanol is:

$$CH_3OH + H_2O \rightleftharpoons 3H_2 + CO_2$$

The composition of the gas leaving a reformer is determined by residence time, temperature, pressure, feed gas composition and catalyst used. In an actual reformer the existing gas would be composed of H_2, CO, CO_2, H_2O, and unreacted methanol, as well as dimethyl ether, methane, and other organic compounds (ref. 17).

Several possible ways that methanol may be used are shown in the Test Matrix in Figure 1. The column titled "Dry", lists the different forms in which methanol may be used—as a liquid, vaporized, or dissociated. The last entry in the first column represents incomplete conversion which is the realistic case for dissociated methanol, that is the fuel that could be expected using an actual reformer. The last two columns are for the cases where water is added to methanol to produce steam reformed methanol ($3H_2$ + CO_2). The "Equimolar" column represents the ideal methanol-water mixture from which steam reformed methanol could be obtained. The "Excess Water" column represents the case for which additional water is added to the methanol-water mixture to improve the hydrogen yield of the steam reforming process (ref. 14).

Some disagreement exists in the literature as to the meaning of the term "reformed methanol". In this report reformed methanol will be used for both dissociated and steam reformed methanol. The terms "dissociated" and "steam reformed" methanol will be used for gaseous $2H_2$ + CO and $3H_2$ + CO_2, respectively.

Several advantages are expected by reforming methanol prior to combustion. These result from: (1) the endothermic nature of the reforming reaction, (2) the possibility of recovering exhaust gas energy and (3) the possibility of burning very lean air-fuel mixtures. In addition, there may be advantages in the area of emissions.

The reformer products could have a greater heating value than liquid methanol resulting from the endothermic reforming reaction (refs. 14-16, 19, 20). When compared to the lower heating value (LHV) of the liquid fuel that would be stored in the tank, there is a 20% increase in the LHV for dissociated methanol, and a 22% increase for steam reformed methanol. Some

Water Content Fuel Conditioning	Dry	Equimolar (1:1 Ratio)	Excess Water
None	*1 Liquid Methanol	N.A.	N.A.
Vaporized but Not Reacted	Vaporized Methanol	Vaporized Methanol and Steam	N.A.
Vaporized and Reacted (n_f=100%)	* 2 $CO+2H_2$ (Dissociated)	* 3 CO_2+3H_2 (Steam Reformed)	N.A.
Vaporized and Reacted (n_f<100%)	N.A.	N.A.	N.A.

* 1,2,3 indicate the cases studied

Figure 1: Test Matrix

disagreement exists as to the amount of this increase for steam reformed methanol. Finegold, et al. (ref. 16) reported a 15% increase in the LHV for steam reformed methanol which is the amount of increase when the LHV of the methanol-water mixture is computed based solely upon the methanol in the mixture.

The use of a catalyst may result in low enough temperatures and reaction times so that waste energy in the engine exhaust may be recovered and used for the reforming process. Also, because of its wide flammability limits and high flame speeds, the hydrogen-rich fuel lends itself readily to ultra-lean combustion and should allow the use of higher compression ratios (higher compression ratio was not investigated in this study). Combining the increase in heating value, the recovery of waste energy from the engine exhaust, lean operation, and higher compression ratios provides a potentially high increase in thermal efficiency for the reformed fuel over that for liquid methanol.

Another advantage of burning the hydrogen-rich fuels in an engine may be lower exhaust emissions. Because of the lack of hydrocarbons in the fuel and the potential for ultra-lean operation to reduce NO_x, reformed methanol is potentially a cleaner burning fuel with lower pollutants in comparison to hydrocarbon fuels (refs. 14, 15).

Although reformed methanol has received serious attention only recently, little information has been found in the literature documenting multicylinder engine performance using steam reformed methanol as a fuel. An attempt was made in Sweden to lower the lean limit of gasoline by adding a small amount of steam reformed methanol to the incoming charge (ref. 13). Methanol was mixed with a portion of the exhaust stream in a catalytic reformer to produce a gas consisting primarily of H_2 and CO_2. As a result, lower emissions were obtained with higher energy efficiencies.

The use of dissociated methanol as an engine fuel was investigated as early as 1971 when a group at the University of Santa Clara investigated the addition of dissociated methanol as a means of extending the lean limit of neat methanol (ref. 21). In that study, bottled H_2 and CO were used to simulate dissociated methanol to fuel a single cylinder CFR engine. The effects of air-fuel ratio, compression ratio, and percent of dissociation of methanol (i.e., mixtures of methanol and dissociated methanol) on the performance and emissions from the engine were investigated while operating at one engine speed (950 rpm). Other researchers have investigated the addition of hydrogen to extend the lean limit of gasoline (refs. 13, 22-24).

In the late 1970's the Nissan Motor Company of Japan (refs. 15, 20) and the Solar Energy Research Institute (SERI) in the United States (refs. 16, 25) conducted experimental work with reformer systems. Performance maps were obtained with four cylinder automotive engines operating on dissociated methanol supplied by an actual catalytic reforming system. Improvements in efficiency and emissions were reported with the dissociated methanol, and in both cases, the engines were modified (increased compression ratio) to take advantage of the characteristics of the hydrogen rich gas. A direct comparison of the SERI and Nissan dissociated methanol results to liquid methanol or gasoline results is difficult because it is not clear what contribution the individual engine modifications made to the overall improvements. Additionally, lack of knowledge about, and constancy of the composition of the fuel gases from the reformer presents still another variable with unknown effects. Consequently, it remains to be determined whether there are advantages to be gained from the use of steam reformed or dissociated methanol as a fuel for an unmodified multicylinder automotive engine.

An experimental research program was conducted at Texas A&M University to further evaluate reformed methanol as an automotive engine fuel. The work was accomplished through a grant from the NASA-Lewis Research Center and was part of the DOE funded Alternative Fuels Utilization Program. The objective of the investigation was to study the advantages and disadvantages in terms of power, efficiency, and emissions of using reformed methanol (dissociated and steam reformed methanol) compared to liquid methanol as an automotive engine fuel. The fuels investigated were liquid methanol and the complete conversion cases of dissociated and steam reformed methanol ($2H_2$ + CO and $3H_2 + CO_2$). These are indicated with the case numbers 1, 2 and 3 on Figure 1.

Performance of an unmodified spark ignition (SI) engine operating on reformed methanol was determined by conducting stationary dynamometer tests. Engine modifications were held to a minimum so that the reformed methanol results could be directly compared to the engine's performance with liquid methanol. Premixed bottled gases were used to simulate the gaseous fuels so that the composition of the fuels would be known and controlled, and would represent the best possible reformer output. The results from operation on the simulated reformed fuels were then compared to the baseline results obtained with liquid methanol.

In addition to the reformed methanol and liquid methanol results, data were taken with gaseous methane and natural gas. These data were obtained to provide still another baseline dataset for gaseous fuels in a carbureted multi-cyclinder engine. These results are presented in Appendix B.

As a means of direct comparison of results to those of others, stationary dynamometer tests were also conducted to obtain data such that the energy consumption and emissions for liquid methanol and methane over the EPA urban driving cycle could be predicted using the three point approximation method of Baker and Daby (ref. 26). These results are presented in Appendix A.

2. APPARATUS

The experimental system was designed to conduct basic experiments with low uncertainty to determine engine performance in terms of power, efficiency, and emissions with both liquid and gaseous fuels. The test setup is briefly described and the results of the uncertainty analysis are presented.

2.1 Description of System Components

The experimental apparatus consisted of four major subsystems: the engine-dynamometer; the fuel-air supply and measurement system; the emissions measurement system; and the data acquisition system. The relationships among these subsystems is illustrated on the schematics in Figures 2 and 3.

A schematic of the basic test setup designed for use with liquid fuels is shown in Figure 2. For gaseous fuel experiments, the engine carburetor was replaced by one designed for propane use, and the gaseous fuel supply system shown in Figure 3 was added to the basic experimental setup. The test engine used for the investigation was a 1977 model, 2.3 liter, four cylinder Ford (Pinto) engine with a compression ratio of 9.0:1. A summary of specifications for the stock engine configuration is given in Table 1. The only external accessories driven by the engine were the water pump and the mechanical fuel pump. Stock positive crankcase ventilation (PCV) and exhaust gas recirculation (EGR) were disconnected for most of the testing, but retained for use with certain tests. The engine crankcase was ventilated by forcing a small stream of air into the engine at the breather

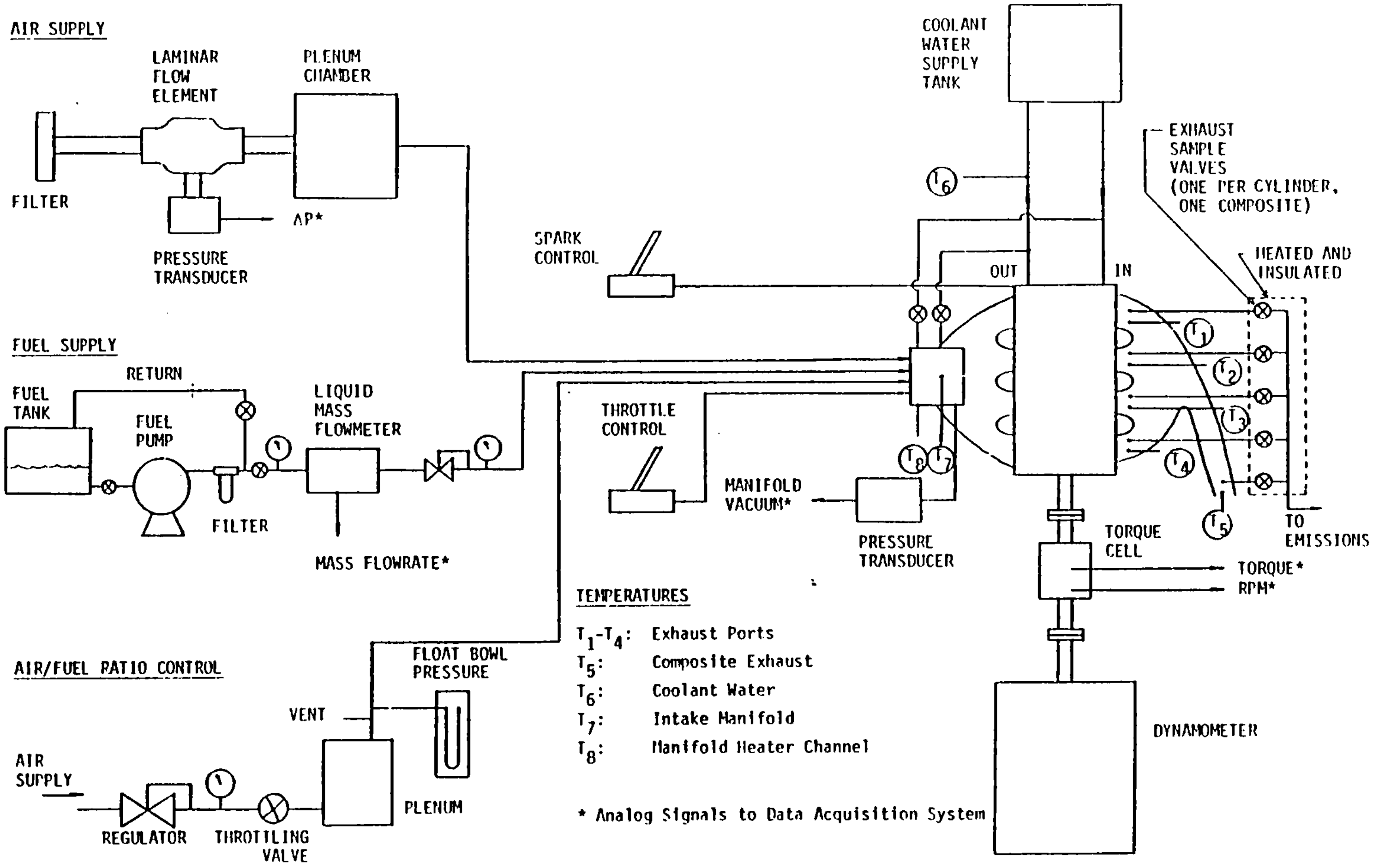

Figure 2: Experimental Test Setup.

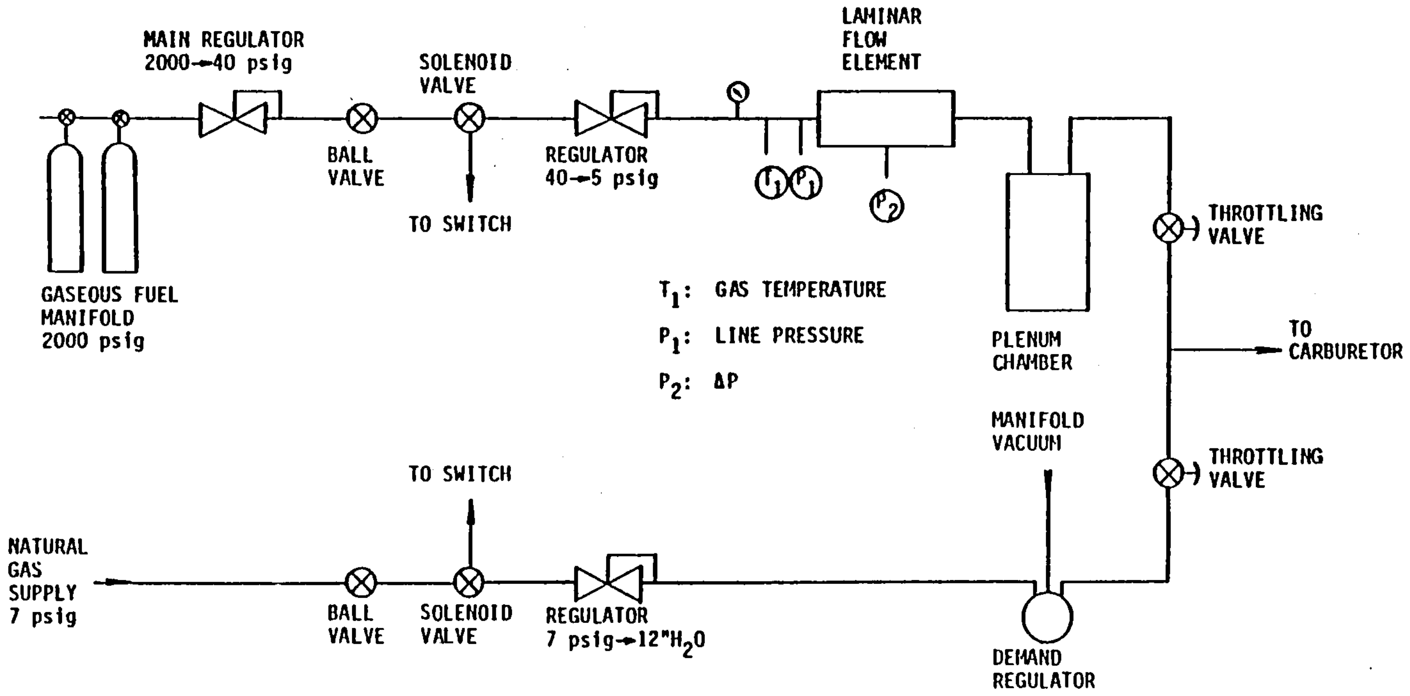

Figure 3: Gaseous Fuel Supply and Measurement System.

Table 1: Stock Engine Configuration Specifications

Displacement, cu. in.	140
Maximum horsepower, bhp, 5,000 rpm	92
Maximum torque, lb-ft, 3,000 rpm	120
Bore and stroke, in.	3.78 x 3.126
Configuration	in-line 4 cylinder
Compression ratio	9.0:1
firing order	1-3-4-2
Ignition timing at idle speed, °BDTC 900 rpm	20
Block material	cast iron
Head material	cast iron
Number of crankshaft main bearings	5
Number of compression rings/piston	2
Number of oil rings/piston	1
Cam drive type	belt and sprocket
Valve lift:	
Intake, in.	0.3997
Exhaust, in.	0.3997
Valve timing:	
Intake opens, °BTDC	22
Intake closes, °ABDC	66
Exhaust opens, °BBDC	64
Exhaust closes, °ATDC	24
Valve overlap, degrees	46
Spark plug gap, in.	0.032
Engine weight, lb	250
Exhaust gas recirculation system:	
Valve type	tapered stem
Control signal	ported vacuum
Point of discharge	carb. spacer
Crankcase emission control:	
Control method	PCV
Point of discharge	intake manifold
Carburetor specifications:	
Type	2V downdraft
Manufacturer	Holley-Weber
Number	D7EE-JA

cap and venting the gases from the engine block where the PCV valve normally would be connected.

The engine was loaded with a 155 horsepower electric dynamometer. The dynamometer was used for starting as well as motoring the engine to measure engine friction power. Engine spark timing was manually set and displayed using an electronic spark controller protractor.

The intake air flow rate was measured with a laminar flow element. Two independent systems were constructed to supply and measure the flow rates of liquid and gaseous fuels. The liquid fuel mass flow rate was measured with a liquid mass flowmeter (LMF), and the gaseous fuel flow rate was measured using a laminar flow element.

An emissions bench, shown in Figure 4, was constructed to enable the continuous monitoring of hydrocarbons (HC), carbon monoside (CO), carbon dioxide (CO_2), oxygen (O_2), and oxides of nitrogen (NO_x) in the engine exhaust. Aldehydes were measured using the MBTH method (refs. 9 and 28) which does not allow for continuous monitoring of aldehyde concentrations in the exhaust.

A micro-computer based data acquisition system was developed and used to display and record the data and perform statistical analysis and reduction of the data.

2.2 Results of Uncertainty Analysis

An uncertainty analysis was performed during the planning stages to aid in the selection of instrumentation. Later, uncertainties in the measured and calculated values were determined to provide an assessment of the validity of the experimental results. The uncertainties associated with the results were low enough such that valid inferences and conclusions could be drawn

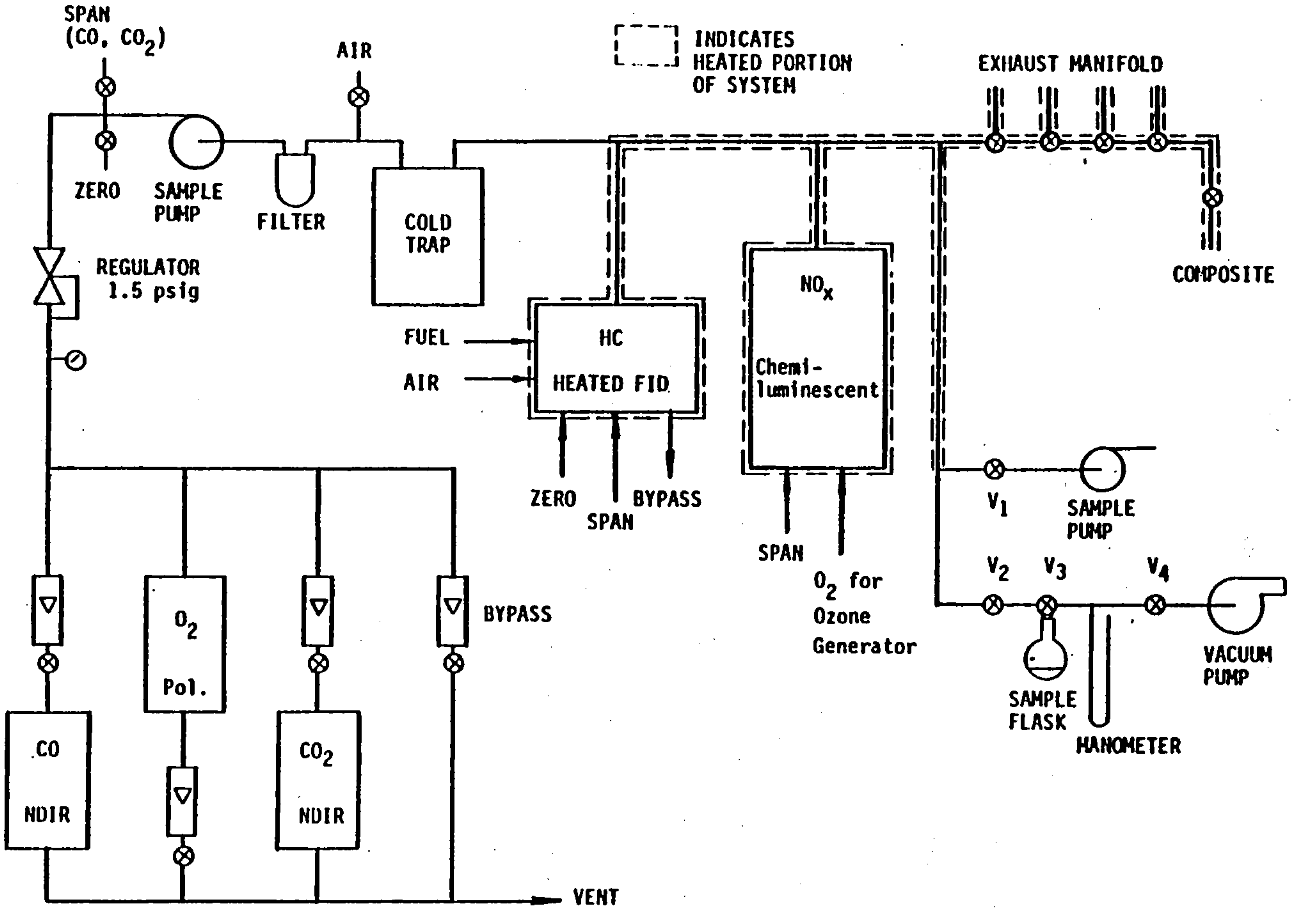

Figure 4: Exhaust Emissions Sampling and Measurement System.

3. EXPERIMENTAL PROCEDURE

Preliminary procedures, completed prior to the actual experimental program are described, followed by descriptions of the general test procedures and procedures specific to individual experiments.

3.1 Preliminary Procedures

Preliminary procedures were required for initial engine inspection, engine break-in, determining engine friction power, and calibration of the gaseous fuel flow meter and the hydrocarbon analyzer.

The engine used in the investigation was new, with only a few hours of operation, therefore, it had to be broken-in to prevent a shift in baseline performance after a few initial hours of operation. During break-in, the engine was operated on pump gasoline and tuned to stock specifications. After operation at each of the break-in conditions for the specified time, the flow rate of the blow-by gas was measured at an operating condition of 1900 rpm and 18 in. Hg. The variation of the blow-by flow rate with hours of operation is shown in Figure 5. About two-thirds of the way through the break-in schedule, the blow-by flow rate decreased and became constant, from which it was concluded that the engine was broken-in.

Engine friction power was needed for the calculation of indicated power from brake power data. The friction power was determined by motoring the engine with the dynamometer while engine oil and coolant were at normal operating temperatures. The variation of friction power with manifold vacuum at various engine speeds is shown in Figure 6. The indicated power was calculated by summing the brake and motored friction powers (see ref. 30).

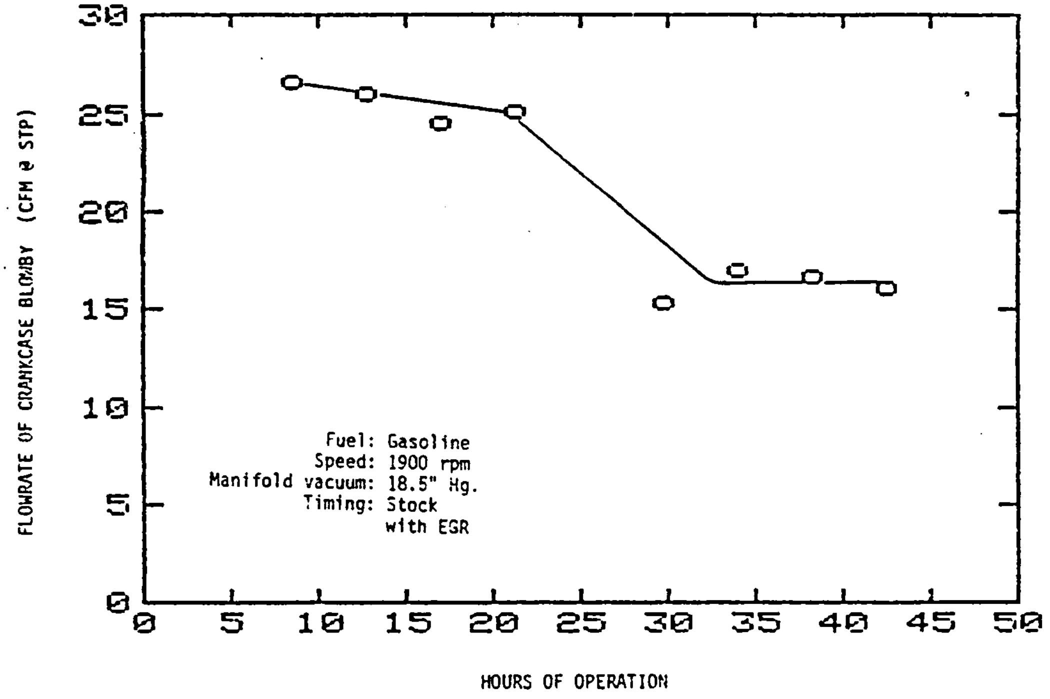

Figure 5: Blow-by as a Function of Operating Hours During Engine Break-in.

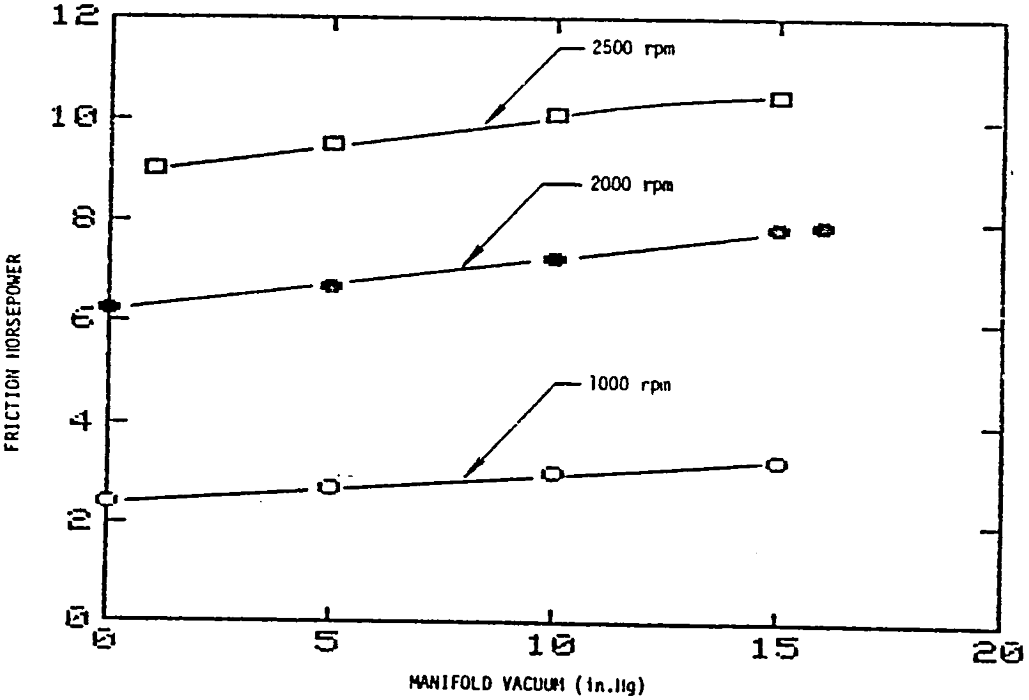

Figure 6: Motored Friction Horsepower as a Function of Manifold Vacuum at Various Engine Speeds.

The laminar flow meter used to measure the gaseous fuel flow rates was supplied with calibration data for air. Information also supplied with the flowmeter indicated that the calibration was valid for any gas as long as the density and viscosity of the gas were known. The density of the reformed gas mixtures was found using the ideal gas law, and the viscosity of the mixtures was calculated using equations from reference 31. Preliminary testing indicated that the calibration was incorrect for the reformed methanol gases, and the flowmeter was calibrated on each gas mixture using a bell-type gasometer as the calibration standard.

The flame ionization detection (FID) hydrocarbon anayzer did not respond completely to methanol and was calibrated by sampling gas mixtures with known concentrations of methanol vapor and recording the response of the meter. The measured response of the hydrocarbon analyzer to methanol was determined to be 79%.

3.2 General Procedures

The general procedures for conducting constant speed tests and recording and reducing data are presented.

All instrumentation was turned on and allowed to warm up and the ambient conditions were recorded. Output analog signals from various instruments were zeroed and spanned with the data acquisition system. Emission meters were zeroed and spanned with reference gases.

After the instrumentation was calibrated, the engine was started and allowed to reach normal operating temperature. The engine was warmed up on gasoline during liquid fuel tests and on natural gas during gaseous fuel tests. After the engine reached oeprating temperature, operation was switched to the test fuel, and the engine allowed to run for more than 10 minutes before beginning the test. This was to insure that the system had been purged of the warm up fuel, and was operating solely on the test fuel.

Dynamometer tests were conducted during which the throttle and load were adjusted to maintain constant engine speed and manifold pressure while the equivalence ratio was varied. The equivalence ratio, denoted by the symbol ϕ, is defined as the ratio of the stoichiometric air-fuel ratio to the actual air-fuel ratio. An equivalence ratio less than one indicates a fuel-lean mixture, and an equivalence ratio greater than one indicates a fuel-rich mixture. Spark timing was set at MBT (minimum spark advance for best torque) at each point. MBT was set by advancing the spark setting while holding engine speed, equivalence ratio, and manifold pressure constant until the torque output peaked or until there was no significant change in torque with additional spark advance. The manifold pressures tested corresponded to wide open throttle (WOT) and 3/4, 1/2, and 1/4 of wide open throttle. The manifold temperature was not controlled during the testing.

Levels of hydrocarbons, NO_x, CO, CO_2, and O_2 in the composite exhaust were recorded at each test condition. In addition, exhaust gases from the individual cylinders were sampled during some tests to detect maldistribution among the cylinders. Equivalence ratios were determined from exhaust emissions data. As a check, the equivalence ratio was also determined from measurements of air and fuel flow rates. The two methods resulted in values which were, in general, within 2% of each other. Aldehydes were measured at regular intervals to establish their range and determine the effects of load and equivalence ratio on their formation.

Aldehydes in the composite exhaust were sampled by slowly filling an evacuated flask, containing a small amount of absorbing reagent, with exhaust gas. The sample was later analyzed following the MBTH method described in references 9 and 28. The emissions meters were recalibrated after four to five data points. To facilitate comparison with others' results the hydrocarbon and NO_x emissions were converted to specific values by dividing the mass emissions by the corresponding indicated horsepower.

When engine operation was stabilized performance and emissions data were recorded using the data acquisition system. Emissions data and MBT spark advance were recorded by hand. Several sets of data were taken at each operating condition and averaged over a 5 to 10 minute time period.

3.3 Procedures Specific to Individual Tests

Constant speed tests were conducted with reformed methanol, liquid methanol, indolene, methane, and natural gas to determine engine performance on these fuels. Tests were also conducted to estimate EPA energy consumption and emissions projections for liquid methanol, methane, and natural gas.

Constant speed tests were conducted with reformed methanol at engine speeds of 1000, 2000, and 2500 rpm without EGR or PCV, and with the fuel entering the engine at ambient temperature. A 3000 rpm test was originally planned, but preliminary testing indicated that engine operation at this speed would not be possible. For these tests, the equivalence ratio (ϕ) was first set at the leanest ϕ at which the desired engine speed and throttle setting could be attained. At this point, the brake power being produced by the engine was that necessary to rotate the dynamometer at the test engine speed. Data were taken at this point and at successively higher equivalence ratios until flashback occurred at that throttle setting. Violent propagation of a flame into the intake manifold was designated flashback. If flashback did not occur, the equivalence ratio was increased to fuel-rich stoichiometry. The equivalence ratio was varied by throttling the flow of reformed methanol to the carburetor.

Baseline liquid methanol and indolene constant speed tests were conducted at 2000 rpm with and without EGR or PCV. An additional liquid methanol test was conducted at 1000 rpm without EGR or PCV. The equivalence ratio was reduced from near stoichiometry to a value at which the hydrocarbon emissions from the engine began to increase rapidly. This latter point was defined as the lean misfire limit (ref. 32). The equivalence ratio was varied by varying the liquid fuel flow rate. This was accomplished by using undersized main metering jets in the carburetor and externally controlling the float bowl pressure such that it was always above atmospheric even for lean operation.

Additional testing was conducted to estimate the EPA driving cycle energy consumption and emissions projections for liquid methanol and methane in order to have results to compare with those of other investigations. In these tests, the engine was operated at three prescribed (ref. 26) steady-state engine speeds and load conditions while holding the equivalence ratio constant. The results from the three points, when weighed properly, simulate the EPA urban driving cycle. Each operating condition was tested with the equivalence ratio equal to 1.0 and 0.9. Details of the procedure, operating conditions, and the results of the testing are presented in Appendix A.

4. RESULTS AND DISCUSSION

Detailed results are presented in Appendix B. The purpose of this section is to discuss and interpret the results.

Included in this section are comparisons of dissociated and steam reformed methanol to liquid methanol, and of the two reformed fuels to each other. Also presented are discussions of the effects of engine operating variables on (1) performance and emissions from reformed methanol, and (2) flashback. The section concludes with two subsections which are a discussion of the energy in the exhaust available to reform the methanol, and a comparison of the results of this investigation to those obtained by other researchers.

4.1 Comparison of Reformed Methanol to Liquid Methanol

The reformed methanol results (both dissociated and steam reformed) are compared to those of liquid methanol to determine relative merits of using reformed methanol and liquid methanol. The three major areas of comparison are the brake power, the brake thermal efficiency, and exhaust emissions.

4.1.1 Brake Power. Operation on reformed methanol (both dissociated and steam reformed) was limited to lean equivalence ratios by flashback, limiting the maximum power output of the engine as compared to liquid methanol at the same engine speed. A comparison of the variation of brake power with equivalence ratio at various throttle settings for reformed methanol and liquid methanol at 2000 rpm is shown on Figures 7 and 8. These figures were obtained by including all of the reformed and liquid methanol results on the same plot. The brake power results presented were not

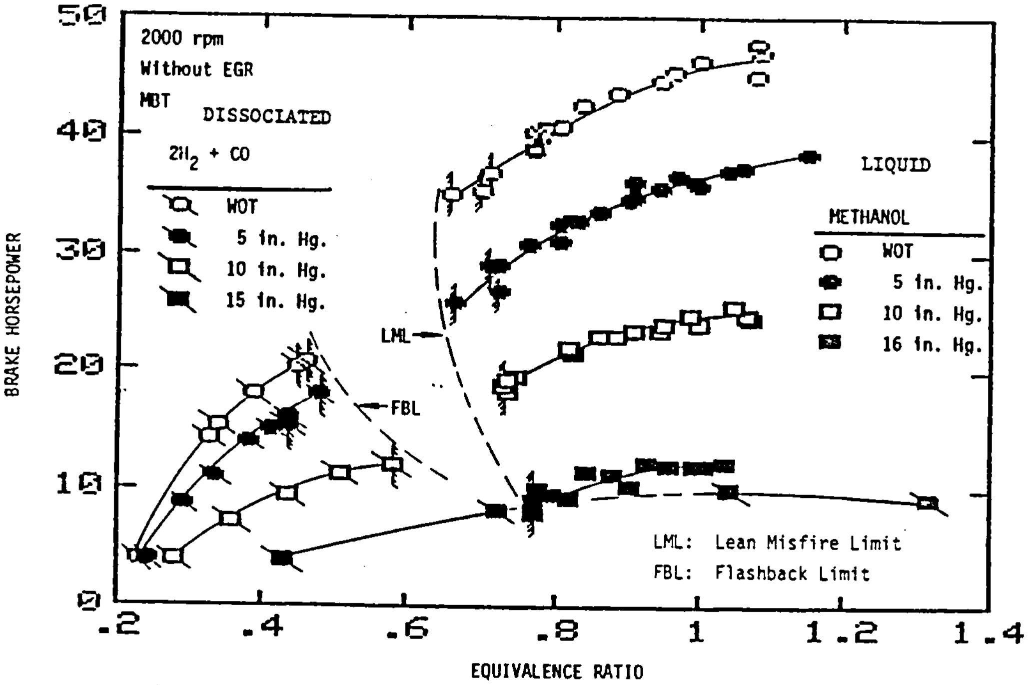

Figure 7: Variation of Brake Horsepower with Equivalence Ratio at Various Throttle Settings for Dissociated and Liquid Methanol at 2000 rpm, without EGR.

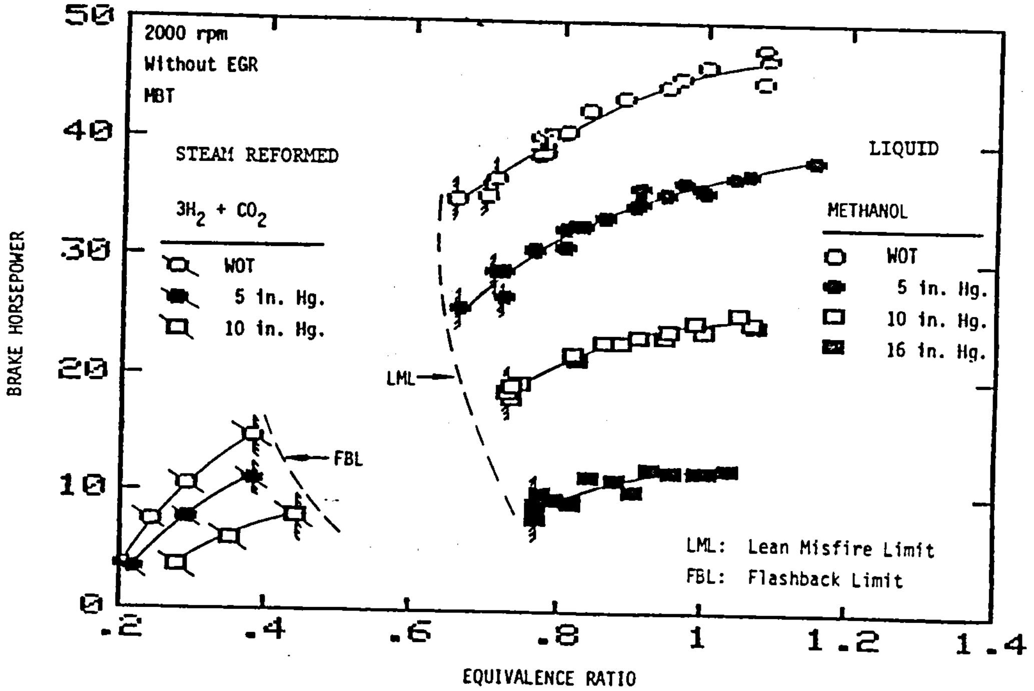

Figure 8: Variation of Brake Horsepower with Equivalence Ratio at Various Throttle Settings for Steam Reformed and Liquid Methanol at 2000 rpm, without EGR.

corrected to standard conditions because the correction factors were small (less than 2%). As can be seen from the figures, the brake power output increased with increasing equivalence ratio at constant manifold pressure and speed for all three fuels and decreased as the throttle was closed at constant equivalence ratio. Maximum power at 2000 rpm was lower by about 50% for dissociated methanol, and 65% for steam reformed methanol, compared to liquid methanol at the same speed. This reduction in maximum power output resulted from the limitation to operation at lean equivalence ratios by flashback. The equivalence ratio at which flashback occurred (for the reformed methanol fuels) was richer as the manifold pressure was reduced. The reason for this is discussed later (section 4.4). For liquid methanol operation, lean misfire also occurred at richer equivalence ratios as the manifold pressure was reduced.

The reformed methanol curves in figure 7 and 8 indicate that even if the flashback limit were avoided and the engine could operate at richer equivalence ratios, there still would be a reduction in maximum power, compared to liquid methanol. This can be visualized by extending the WOT brake power curves for reformed methanol to stoichiometry ($\phi = 1$). This reduction in maximum power results from the displacement of air by the gaseous fuel (reducing the volumetric efficiency). This reduction would be greater for steam reformed methanol than for dissociated methanol because of the presence of an inert component in the gaseous fuel mixture.

Since power output is severely limited by flashback with reformed methanol compared with liquid methanol, there are no advantages to using reformed methanol rather than liquid methanol based upon peak power output. However, power outputs equivalent to those obtained at part throttle using liquid methanol can be obtained with reformed methanol. At these low powers, there are advantages in using reformed methanol rather than liquid methanol. These advantages are discussed in the following two parts of this section. A system that incorporates a reformer as well as provision for liquid fuel supply may very well be required to obtain the maximum power levels required and still take advantage of the reformed fuel. However, such a hybrid system was not investigated during this program.

4.1.2 Brake Thermal Efficiency. At low power outputs, the brake thermal efficiency (Bte) for the reformed methanol fuels was higher than for liquid methanol as shown in Figures 9 and 10. It is important to keep in mind that the efficiency values shown for the reformed methanol fuels were obtained using simulated reformer product mixtures that would only result under the optimum conditions of: (1) 100% conversion in the reformer, (2) sufficient waste energy available at high enough temperature to accomplish 100% conversion, and (3) a temperature of the gaseous fuel that was lower than it would leave an actual reformer. The figures show the variation of brake thermal efficiency with brake power output at constant equivalence ratios and engine speeds. They were obtained by cross plotting the curves of brake power and brake thermal efficiency as functions of equivalence ratio. All the efficiency-power curves terminate at the greatest power (WOT) attained for each equivalence ratio shown. For the reformed methanol fuels these were the WOT conditions at which flashback occurred. The thermal efficiency of liquid methanol was based upon the constant pressure LHV of liquid methanol, and that for the reformed fuels was based upon the constant pressure LHV of the liquid fuel mixture that would be stored in the fuel tank and subsequently reformed into the gaseous fuel. That is, the efficiency for dissociated methanol was based upon the LHV of liquid

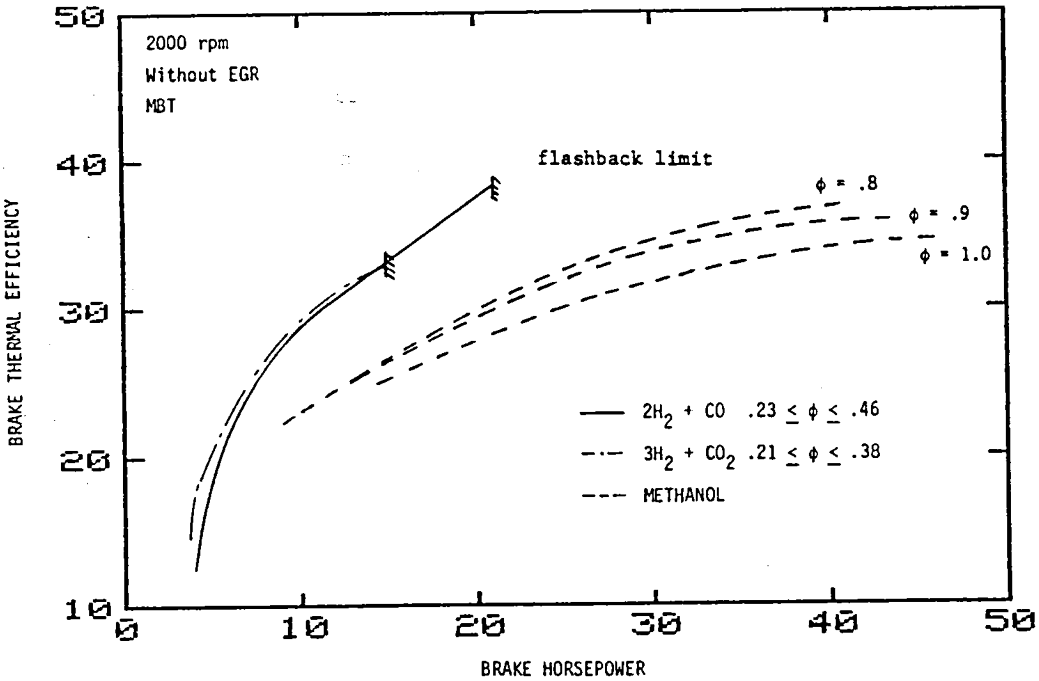

Figure 9: Comparison of the Variation of Brake Thermal Efficiency with Brake Horsepower for Reformed and Liquid Methanol at 2000 rpm.

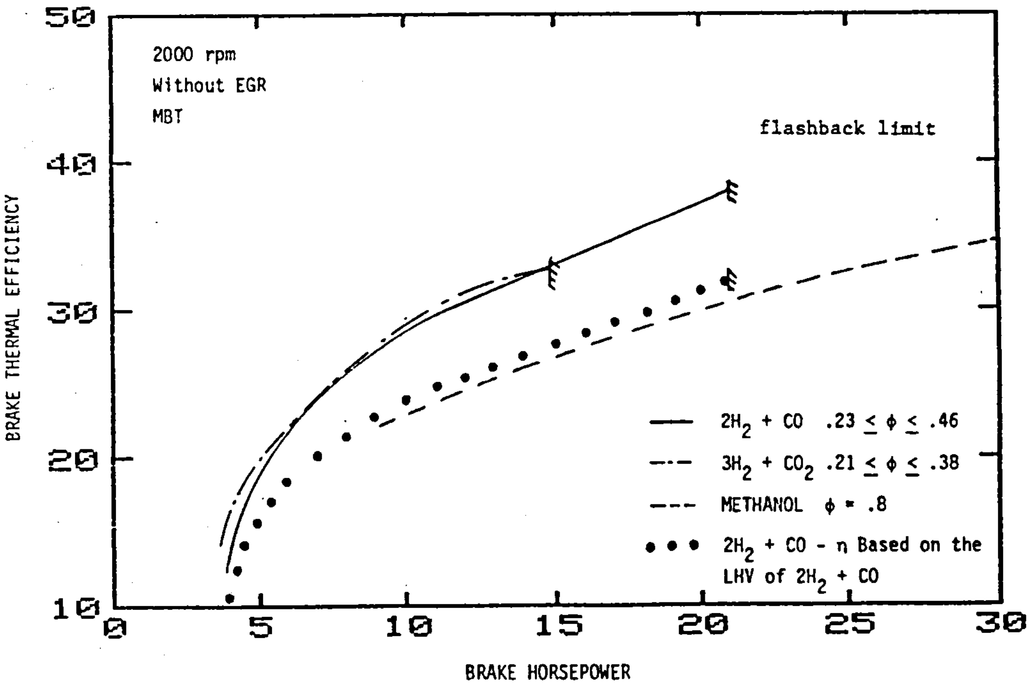

Figure 10: Comparison of the Variation of Brake Thermal Efficiency with Brake Horsepower for Reformed and Liquid Methanol at 2000 rpm and Low Loads.

methanol, and the efficiency for steam reformed methanol was based upon the LHV of an equimolar methanol-water mixture. This basis for evaluating the efficiencies was chosen to provide an optimum case comparison of the output of the engine to the energy which would be input to the fuel tank. It is an optimum case because the gaseous fuel mixtures used simulated the most optimistic situation, that is 100% conversion of the liquid fuel to gaseous dissociated or steam reformed methanol before introduction into the engine. Presenting the efficiences in this manner also provided a means for evaluating the upper limit effect of the endothermic reforming reaction on the overall system efficiency, and for comparison to the results of other studies (refs. 15 and 25) in which complete systems including actual reformers were investigated.

The brake thermal efficiency results for liquid methanol were cross plotted for equivalence ratios of 0.8, 0.9, and 1.0, with the highest efficiencies occurring at an equivalence ratio of 0.8. The brake thermal efficiency increased by about 9% when the equivalence ratio was reduced from 1.0 to 0.8 at a particular value of power output (Fig. 9). This increase in efficiency primarily resulted from opening the throttle more in order to produce the same power output at the lower equivalence ratio. The increase in efficiency as brake power output increased for a particular value of equivalence ratio resulted from opening the throttle more to increase the brake power, thereby reducing the pumping losses.

The brake thermal efficiency curves for the reformed methanol fuels at the various constant equivalence ratios were found to overlap to form a single continuous curve. The average of these overlapping curves is shown in Figures 9 and 10 along with the range of equivalence ratios for which the data was cross plotted. As can be seen, there was little or no measurable difference in the brake thermal efficiency between dissociated and steam reformed methanol over the range for which equal power was produced with each fuel. However, higher power resulted from dissociated methanol and, therefore, resulted in higher efficiency than steam reformed methanol. This can be seen from the fact that the curve for dissociated methanol (solid line) extends beyond the line for steam reformed methanol. This resulted because, with this engine, the flashback limit for dissociated methanol occurred at a higher equivalence ratio than for steam reformed methanol.

As noted earlier and is evident from Figure 9, the only engine operating region in which the reformed and liquid methanol thermal efficiency results can be compared is at the low power outputs. To facilitate the comparison, the low power region of Figure 9 was expanded and is shown as Figure 10. In Figure 10, the equivalence ratios greater than 0.8 are not shown for liquid methanol so that a comparison between the reformed fuels' efficiency results and the highest efficiencies attained with liquid methanol can be made. As shown in Figure 10, there was about a 25% greater brake thermal efficiency with reformed methanol compared to liquid methanol in the 10 to 20 Bhp range. As explained above the efficiencies for the reformed fuels were based upon the LHV of the liquid fuel mixture that would be stored in the fuel tank and therefore, represent the upper limit; that is, efficiencies for a system with an actual reformer would probably be lower.

The dotted line on Figure 10 shows the efficiency for dissociated methanol computed based upon the LHV of the actual gaseous fuels used rather than that of the liquid fuel in the tank. Although not shown, the efficiency for steam reformed methanol would overlap this dotted line up to the point at which flashback occurred. As can be seen there was little or no difference between the efficiencies for the gaseous fuels computed in this way and that for liquid methanol. Thus, the difference between the efficiency for the reformed fuels and that for liquid methanol must result primarily from the endothermic reforming reaction increasing the heating value of the fuel. For an actual engine reformer system, for which the product would be something less than 100% converted, the overall efficiency for the reformed fuels would lie somewhere between the dotted line and solid line as the increase in heating value in the reformer would not be as great. Therefore, the difference between the efficiencies for the reformed fuels and liquid methanol shown on Figure 10 is composed of two parts. One part, approximately 80% (20 of the 25 percentage points), of the increase in efficiency determined for reformed (based upon the LHV of the liquid fuel mixture in the tank) over liquid methanol results from the larger heating value of the reformed fuel. The remainder of the increase (5 percentage points) resulted from lean combustion. The increase in thermal efficiency from lean combustion was primarily the sum of two parts: an increase in efficiency resulting from opening the throttle, and a decrease in efficiency resulting from operation at very lean equivalence ratios. For most fuels, the brake thermal efficiency is a maximum at slightly lean equivalence ratios, and operation at very lean equivalence ratios results in a decrease in thermal efficiency (ref. 30).

At low power outputs, the thermal efficiency can be increased by lowering the engine speed for both reformed and liquid methanol (or any other fuel) as shown in Figures 11 and 12. This increase in efficiency results primarily from the necessity to open the throttle more to produce

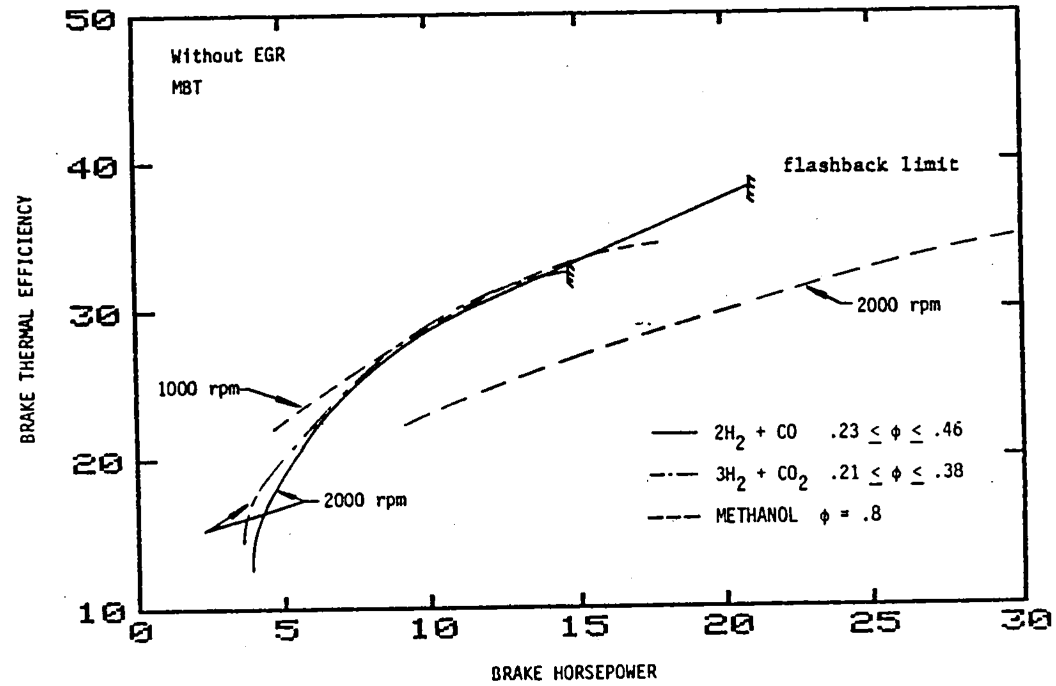

Figure 11: Comparison of the Variation of Brake Thermal Efficiency with Brake Horsepower for Reformed Methanol at 2000 rpm and Liquid Methanol at 1000 and 2000 rpm.

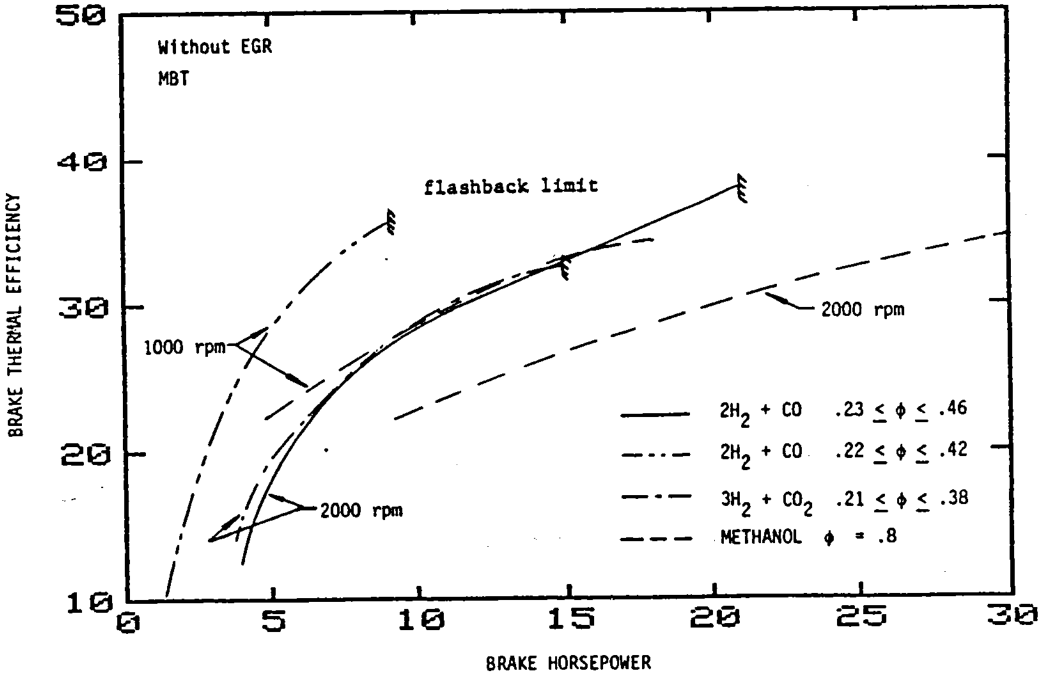

Figure 12: Comparison of the Variation of Brake Thermal Efficiency with Brake Horsepower for Reformed and Liquid Methanol at 1000 and 2000 rpm.

the required power at the lower speeds thus lowering throttling losses. Engine friction is also lower at the lower speeds. Figure 11 is the same as Figure 10 with the addition of an efficiency curve for liquid methanol at 1000 rpm. For clarity only one curve is included for liquid methanol at 1000 rpm representing the best case, that is, the highest efficiency (ϕ =0.8). As can be seen, the high efficiencies at low powers obtained with reformed methanol at 2000 rpm could also be obtained with reformed methanol at 2000 rpm could also be obtained with liquid methanol by reducing the engine speed to 1000 rpm (possibly with gearing). However, lowering the engine speed to 1000 rpm with dissociated methanol also resulted in an increase in efficiency over that for liquid methanol at the same speed as shown in Figure 12. An efficiency curve for steam reformed methanol at 1000 rpm is not shown because operation at this speed was limited by flashback to extremely lean equivalence ratios and very low powers (about 3 Bhp). The amount of the increase in the efficiency at 1000 rpm for dissociated over that for liquid methanol was approximately the same as that found at 2000 rpm, with the most of the increase resulting from the increase in the heating value of the fuel.

4.1.3 Exhaust Emissions. In this section, the exhaust emissions from reformed methanol at 2000 rpm are compared to those from liquid methanol at the same engine speed. The emissions that are compared are hydrocarbons, oxides of nitrogen (NO_x), carbon monoxide (CO), and aldehydes (as formaldehyde).

Hydrocarbon emissions from reformed methanol were lower than those from liquid methanol by a factor of 3 to 10 on a mass basis at 2000 rpm as shown in Figures 13 and 14. On a volume basis, the hydrocarbon emissions from reformed methanol were never greater than 50 ppm. In these figures, the

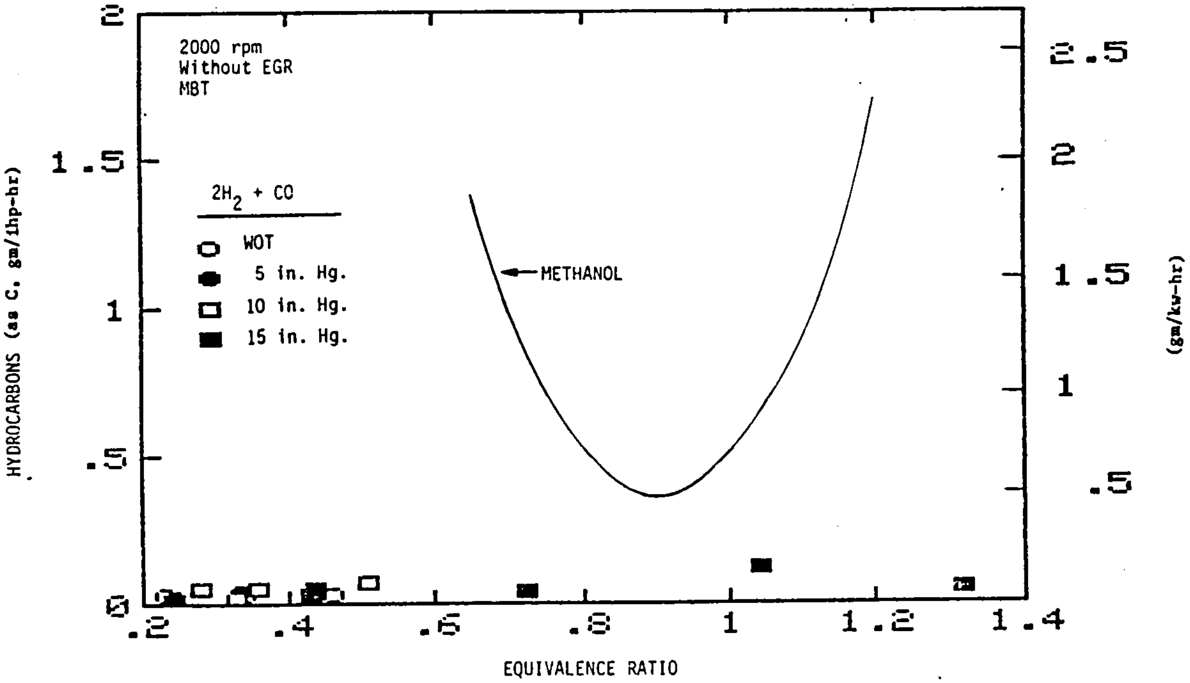

Figure 13: Comparison of Hydrocarbon Emissions from Dissociated and Liquid Methanol at 2000 rpm.

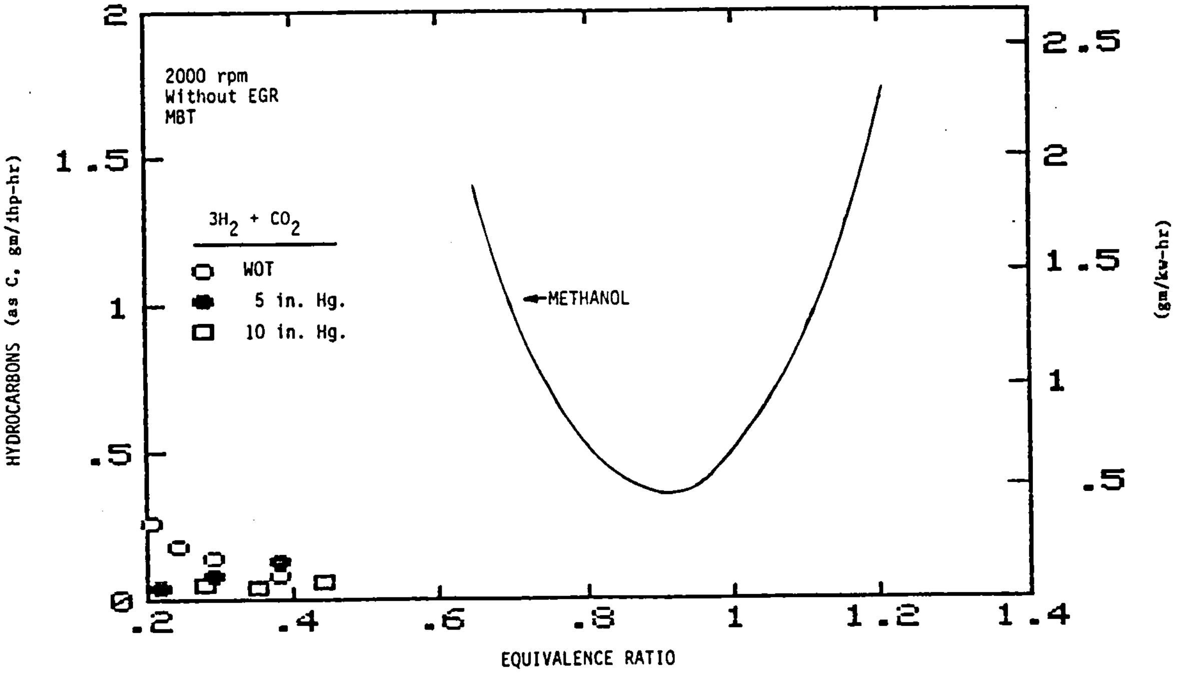

Figure 14: Comparison of Hydrocarbon Emissions from Steam Reformed and Liquid Methanol at 2000 rpm.

test data taken from operation on reformed methanol are plotted, and the best fit curve of the liquid methanol hydrocarbon data is shown as a solid line. These data are based on measured carbon emissions, and are not unburned fuel.

Hydrocarbon emissions from reformed methanol must have been from the engine lubrication oil, since there are no hydrocarbon compounds in either form of reformed methanol. The hydrocarbon emissions from reformed methanol were not affected by the equivalence ratio as they were for operation on liquid methanol. This would be expected since these emissions do not result from the fuel itself.

The point should be made that the magnitude of the hydrocarbon emissions from reformed methanol is only a function of the engine design and the condition of the engine since they are not from the fuel. Engine oil consumption increases with increased engine wear. This should be less of a concern operating on reformed methanol than for operation on liquid methanol or gasoline since engine wear should be reduced when using a gaseous fuel (ref. 30). Also, the engine would have to be in very poor condition before the hydrocarbon emissions from an engine operating on reformed methanol would exceed the regulated maximum value of 0.4 gr/mile, at which time there would be a noticeable decrease in engine performance.

Oxides of nitrogen emissions from reformed methanol while operating at 2000 rpm were also very low (less than 0.4 gm/ihp-hr) at the very lean equivalence ratios. It was only at the lean equivalence ratios where operation without flashback at large throttle openings was possible. A comparison between NO_x emissions from reformed methanol and liquid methanol at 2000 rpm is shown in Figures 15 and 16. The NO_x emissions from liquid methanol at all throttle settings tested are represented by the shaded region on the figures. The largest NO_x emissions from liquid methanol occurred at WOT with these values forming the upper boundary of the region shown. The peak NO_x emissions measured from dissociated methanol occurred

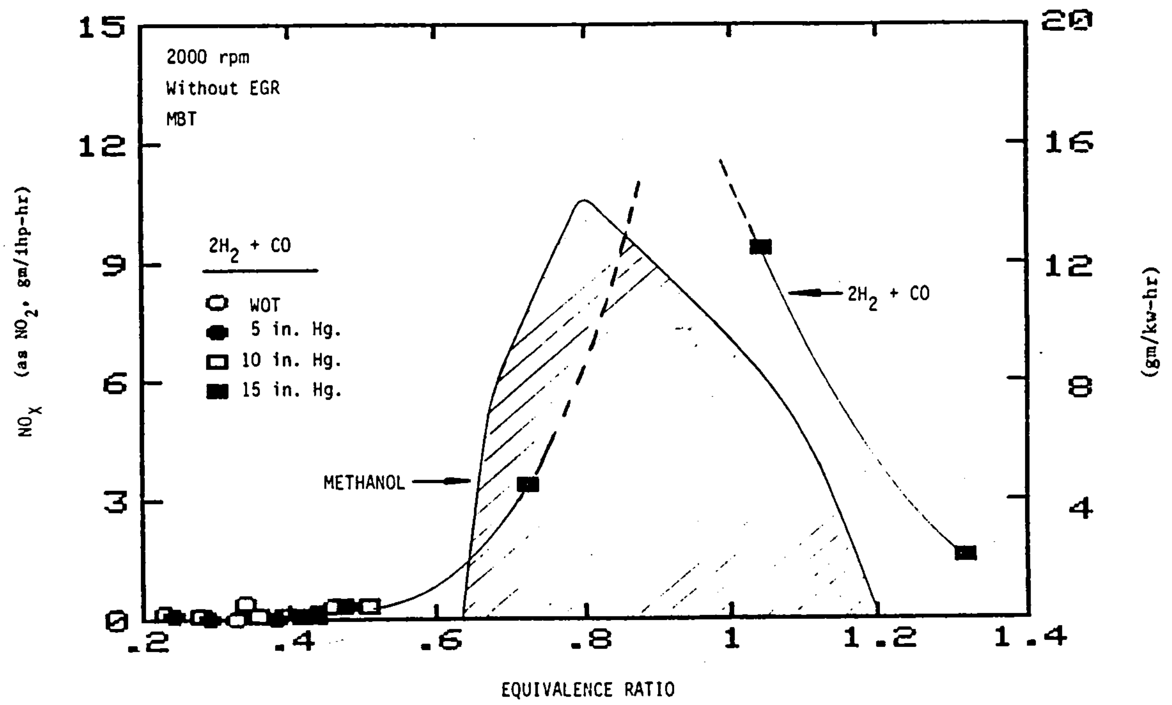

Figure 15: Comparison of Oxides of Nitrogen Emissions from Dissociated and Liquid Methanol at 2000 rpm.

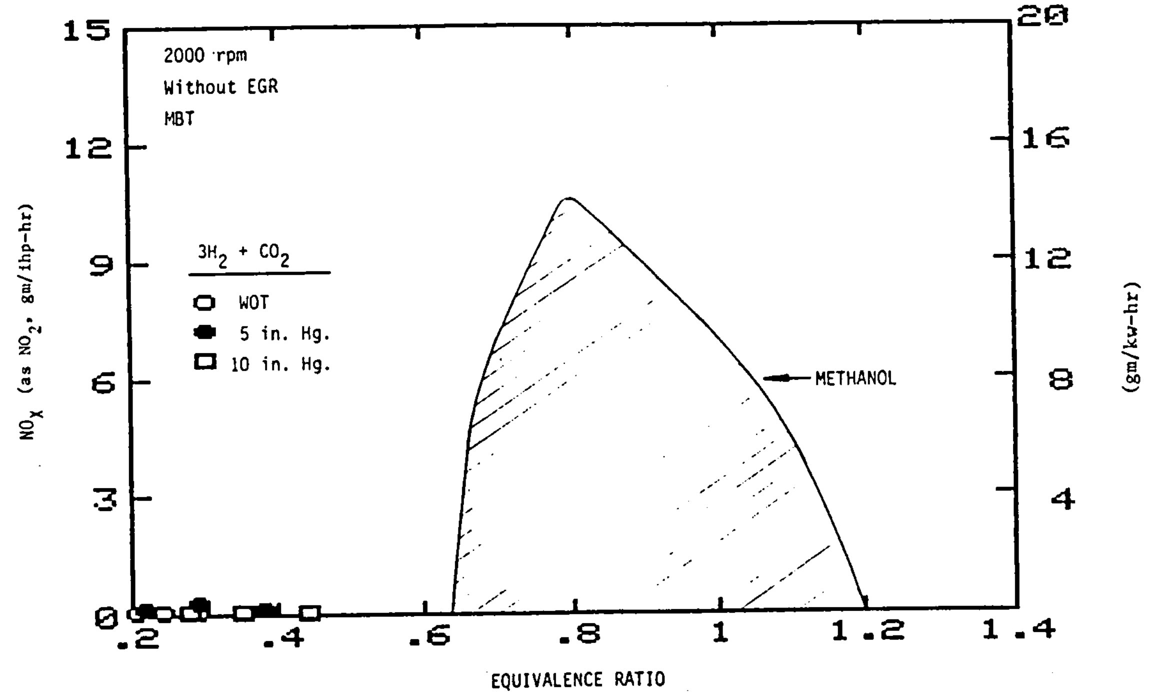

Figure 16: Comparison of Oxides of Nitrogen Emissions from Steam Reformed and Liquid Methanol at 2000 rpm.

at the lowest throttle opening tested (a manifold vacuum of 15 in Hg) and are slightly greater than those from liquid methanol at WOT. Comparing NO_x emissions at the lower throttle openings (low manifold pressures), the NO_x emissions from dissociated methanol were much greater than those from liquid methanol. The higher NO_x emissions resulted from operation on dissociated methanol near stoichiometric operation because it burns with a higher flame temperature than liquid methanol thus promoting greater NO_x formation.

The effect of equivalence ratio on CO emissions for operation on dissociated methanol and liquid methanol at 2000 rpm is shown in Figure 17. CO emissions from dissociated methanol were high at rich and very lean equivalence ratios. The lean equivalence ratio at which the CO emissions started to increase was about $\phi = 0.35$ and was independent of engine speed and manifold pressure. Near stoichiometry, the CO emissions from dissociated methanol were about 60% lower than those from liquid methanol (at least for on data point shown).

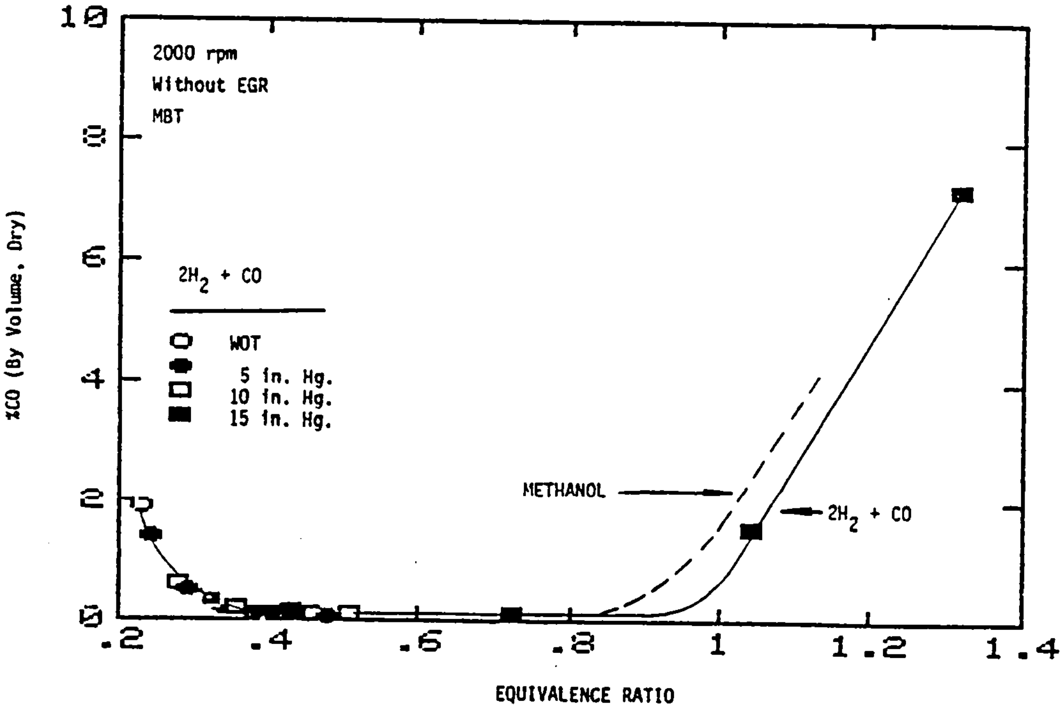

Figure 17: Comparison of Carbon Monoxide Emissions from Dissociated and Liquid Methanol at 2000 rpm.

Pefley, et al. (ref. 11) showed that, as expected, CO emissions from liquid methanol begin increasing at richer equivalence ratios (the position of knee of curve shifts to the right) as the maldistribution of the fuel-air mixture from cylinder to cylinder is reduced. The result of maldistribution in a multicylinder engine is that some of the cylinders receive a richer (and some leaner) fuel-air mixture than the average that is supplied to the engine, the consequence being higher CO emissions at lower overall equivalence ratios. If maldistribution with liquid methanol is reduced, the CO emissions from dissociated and liquid methanol might be about the same at equivalence ratios near and greater than stoichiometry.

Carbon monoxide emissions from dissociated methanol represent unburned fuel in the exhaust. At the lean equivalence ratios where the CO emissions were high, the CO in dissociated methanol ($2H_2 + CO$) was not completely burning. The incomplete burning of the CO in the fuel places a limit on how lean an engine can be operated on dissociate methanol with low CO emissions. Therefore, dissociated methanol has a "lean limit" similar to hydrocarbon fuels because of the incomplete burning of the CO in the fuel. The value of the equivalence ratio defining the "lean limit" would depend upon an allowable level of CO emissions from the engine.

There were no CO emissions from steam reformed methanol in the range of equivalence ratios and throttle settings tested, so there are no plots of CO as a function of equivalence ratio presented. However, at one point during testing, the engine was operated at a very rich equivalence ratio ($\phi > 1$) and low engine speed resulting in high CO emissions (greater than 10%). The CO resulted from the shift toward equilibrium of the excess hydrogen in the exhaust, the water produced during combustion, and the CO_2 in the exhaust (from the fuel). Although it is unlikely that an engine would be operated on steam reformed methanol at extremely rich equivalence ratios, the results of operating at the above condition were included to make the point that CO emissions are possible from steam reformed methanol.

The aldehyde emissions (as formaldehyde) from reformed methanol and liquid methanol at 2000 rpm are shown in Figure 18. As can be seen, the aldehyde emissions from reformed methanol were about an order of magnitude lower than from liquid methanol possibly because of the lack of hydrocarbon compounds in the fuel that could be partially oxidized to form an aldehyde.

In the range of equivalence ratios from 0.21 to 0.46, the hydrocarbon and NO_x emissions from reformed methanol were low, and there would probably be no need for an exhaust emission catalyst to pass present emission regulations. However, operation on dissociated methanol could result in high CO emissions (up to about 2% by volume in the exhaust on a dry basis)

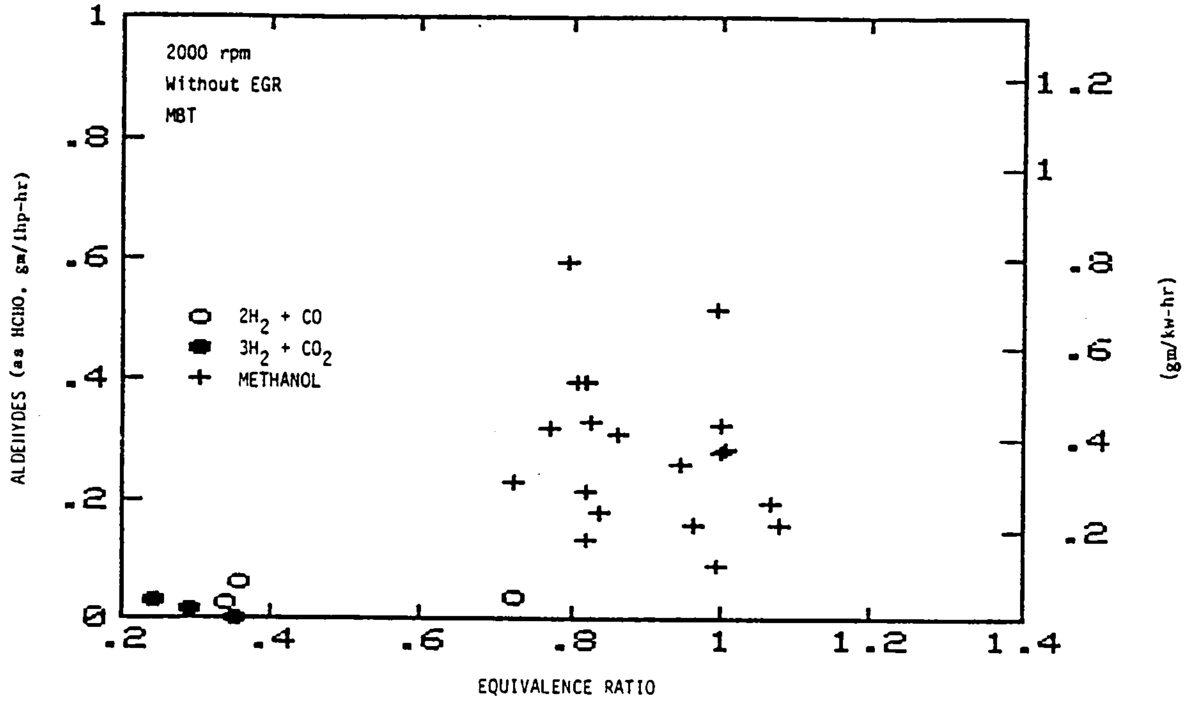

Figure 18: Comparison of Aldehyde Emissions from Reformed and Liquid Methanol at 2000 rpm.

if the engine is operated below $\phi = 0.35$. To keep CO emissions low at the low powers, the engine needs to be operated at equivalence ratios above $\phi = 0.35$, and the power output lowered by throttling the engine.

4.2 Comparison of Dissociated Methanol to Steam Reformed Methanol

4.2.1 Engine Performance. In Figures 19 and 20, the variation of brake power and brake thermal efficiency with equivalence ratio at 2000 rpm and WOT are shown. Dissociated methanol resulted in higher maximum power produced and higher thermal efficiency than did steam reformed methanol because flashback occurred at a higher equivalence ratio for dissociated methanol. Maximum power produced from dissociated methanol was about 40% higher, and peak brake thermal efficiency was about 15% higher than for steam reformed methanol. The power and efficiency curves cross at an equivalence ratio of about 0.3, with operation on steam reformed methanol resulting in higher powers and efficiences at the lower equivalence ratios.

For equivalence ratios above $\phi = 0.3$, power outputs and efficiencies were lower with steam reformed methanol than with dissociated methanol because of the inert CO_2 present in steam reformed methanol resulting in lower combustion temperatures (ref. 30). Power outputs from dissociated methanol were lower than from steam reformed methanol at equivalence ratios below $\phi = 0.3$ because some of the CO in the fuel was not burning as was apparent from the CO emissions (Figure 17). Although not all of the H_2 in both fuels was burned completely at very lean equivalence ratios (ref. 2), a larger percentage of the incoming fuel's chemical energy exited the engine in the exhaust in the form of unburned fuel for dissociated methanol than for steam reformed methanol. Since there was less chemical energy converted into useful work while operating on dissociated methanol at extremely lean

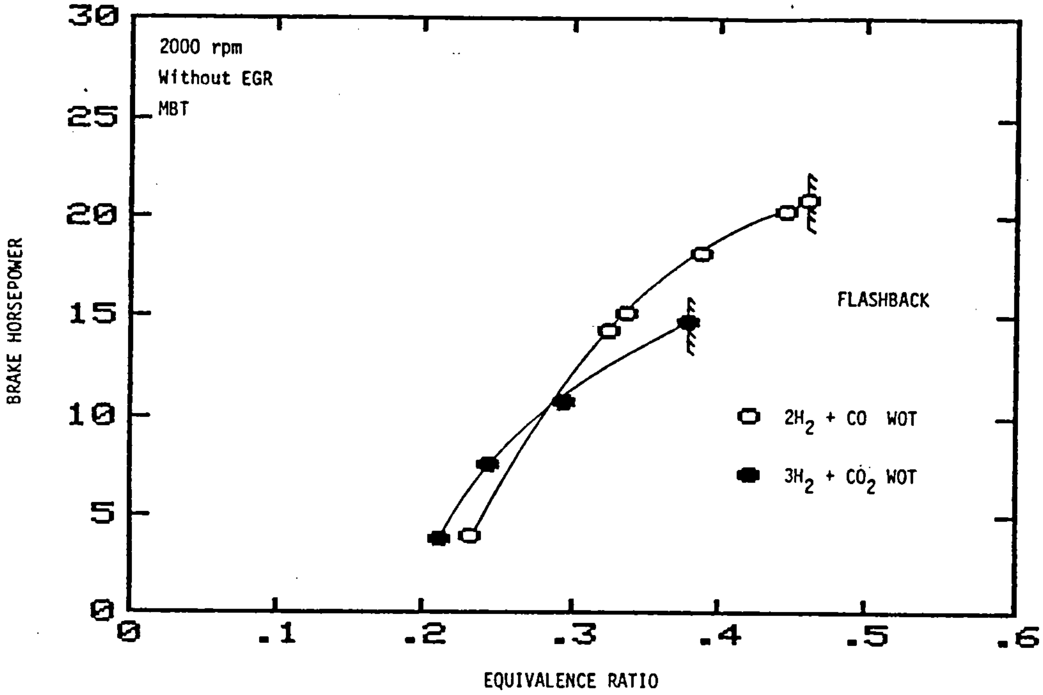

Figure 19: Comparison of WOT Brake Horsepower for Dissociated and Steam Reformed Methanol at 2000 rpm.

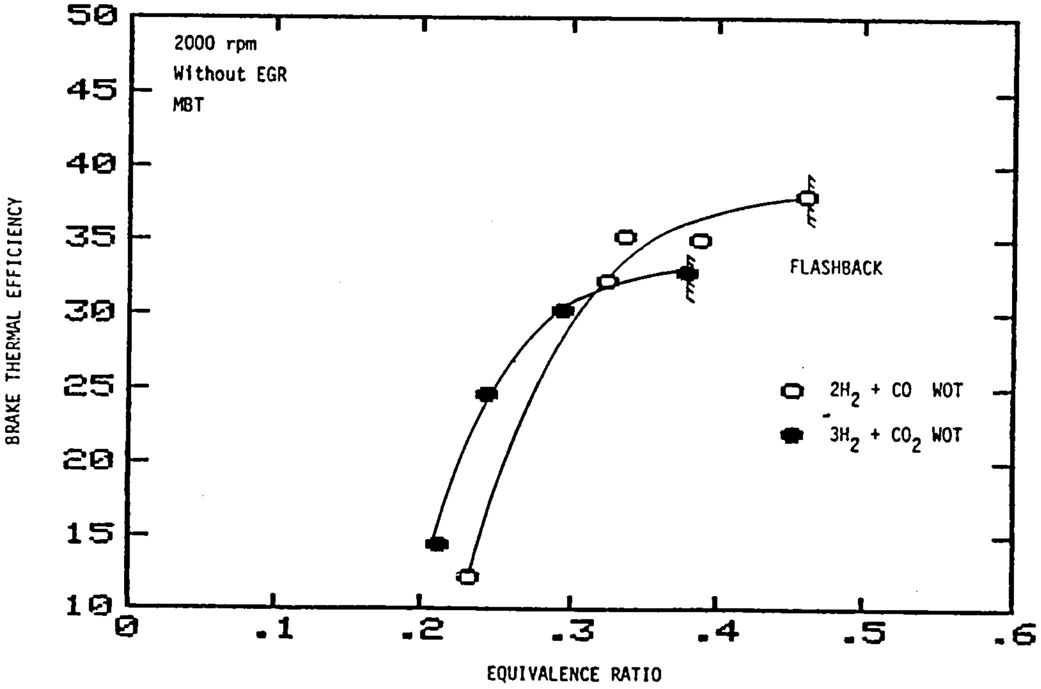

Figure 20: Comparison of WOT Brake Thermal Efficiency for Dissociated and Steam Reformed Methanol at 2000 rpm.

equivalence ratios, the power output and thermal efficiency were lower than those for steam reformed methanol.

4.2.2 Exhaust Emissions. Hydrocarbon, NO_x, and aldehyde emissions from dissociated methanol and steam reformed methanol were about the same at lean equivalence ratios. Hydrocarbon emissions were about the same because they did not originate from the fuel, but from the engine, and NO_x emissions were about the same at lean equivalence ratios because combustion temperatures from operation on both reformed fuels were not high enough to result in substantial amounts of NO_x.

The large difference between the two reformed fuels, based upon emissions, was in the CO emissions. While absent from lean operation on steam reformed methanol, CO emissions from dissociated methanol were high at very lean equivalence ratios (Figure 17), and still present (although low) at moderately lean equivalence ratios.

4.3 Effect of Engine Operating Variables on Performance and Emissions

Various engine operating parameters affect the performance and emissions from the engine while operating on reformed methanol. The effects of equivalence ratio, engine speed, and throttle opening are discussed in the following sections.

4.3.1 Effect of Equivalence Ratio on MBT Spark Advance. The effect of equivalence ratio on MBT spark advance is shown in Figure 21 for reformed methanol at 2000 rpm. As expected, the amount of spark advance for MBT decreased in value as the equivalence ratio was increased. The MBT spark advances for the two reformed fuels were about the same at the same equivalence ratio and were lower than that for liquid methanol at the richer equivalence ratios.

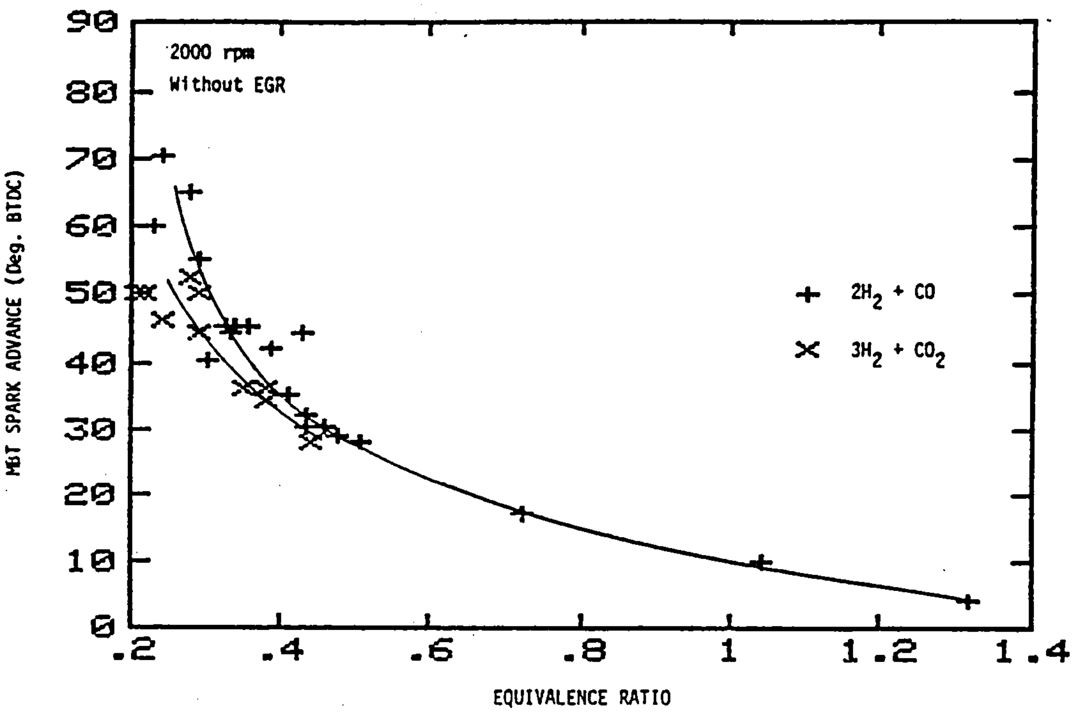

Figure 21: Effect of Equivalence Ratio on MBT Spark Advance for Dissociated and Steam Reformed Methanol at 2000 rpm, without EGR.

4.3.2 Effect of Engine Speed. Variations in engine speed affected the NO_x emissions and brake thermal efficiency operating on either reformed methanol fuel.

Changes in engine speed did not affect NO_x emissions operating on steam reformed methanol in the range of equivalence ratios where operation without flashback was possible. However, speed did affect NO_x emissions operating on dissociated methanol at a manifold vacuum of 15 in. Hg, representing 1/4 throttle (where operation at rich equivalence ratios was possible). Increasing the engine speed increased the NO_x emissions in the exhaust as shown in Figure 22. The increase in NO_x emissions with increasing engine speed probably resulted from a decrease in the percent heat loss with increased engine speed (ref. 30). Lowering the percent heat loss probably resulted in higher cylinder temperatures, thus promoting higher NO_x formation.

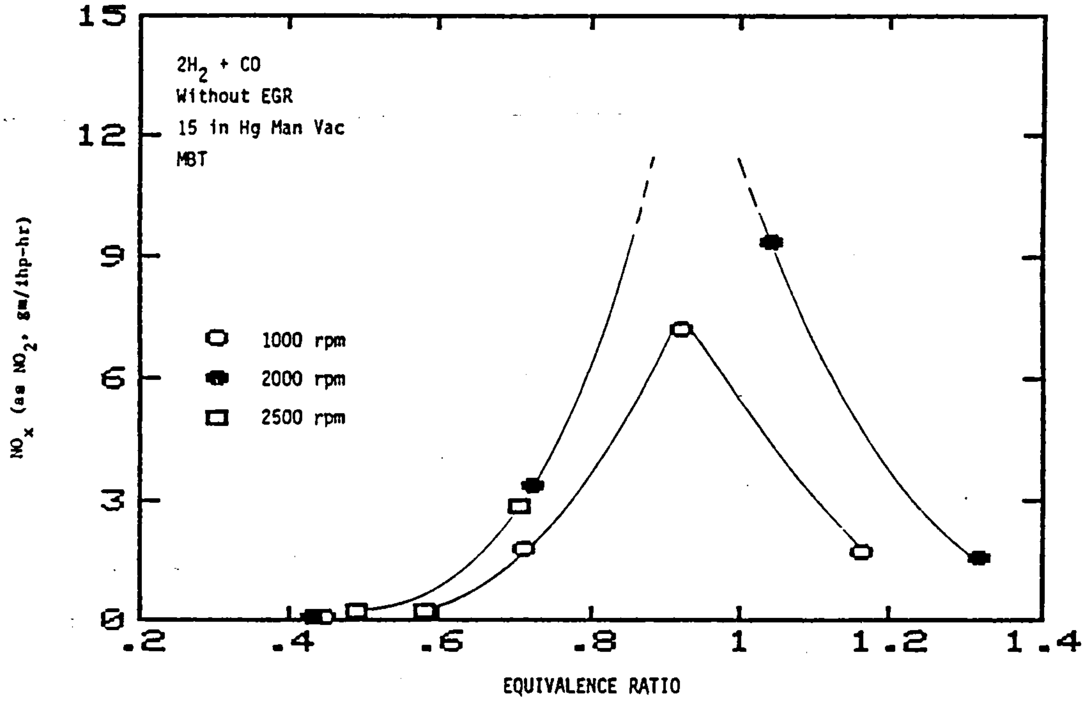

Figure 22: Effect of Engine Speed on Oxides of Nitrogen Emissions from Dissociated Methonal, without EGR.

Decreasing the engine speed, at constant brake power output and equivalence ratio, increased the brake thermal efficiency operating on reformed methanol (or any other fuel) as shown in Figures 23 and 24. Lines of constant manifold pressure (vacuum) for $\phi = 0.4$ have been included for dissociated methanol in Figure 23. At about 7 Bhp operating on dissociated methanol at $\phi = 0.4$ (Figure 23), the engine was operating at a low manifold pressure at 2000 rpm, while at 1000 rpm the engine was operating near WOT. By decreasing the engine speed at this operating condition, the brake thermal efficiency was increased by about 37%. This large increase in efficiency resulted primarily from opening the throttle more which lowered pumping losses, and secondarily from the decrease in engine speed which reduced friction losses.

As can be seen in Figure 23, the lines of constant manifold pressure for $\phi = 0.4$ slope upward and to the right. This means that at constant manifold pressure and equivalence ratio, the brake thermal efficiency and

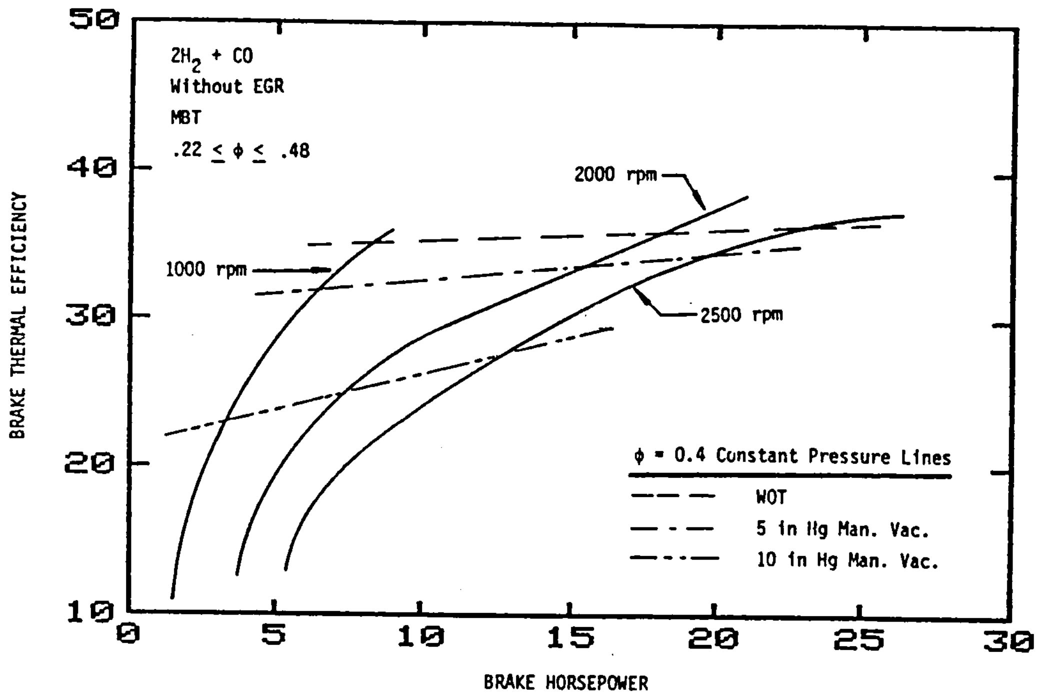

Figure 23: Effect of Engine Speed on the Variation of Brake Thermal Efficiency with Brake Horsepower for Dissociated Methanol.

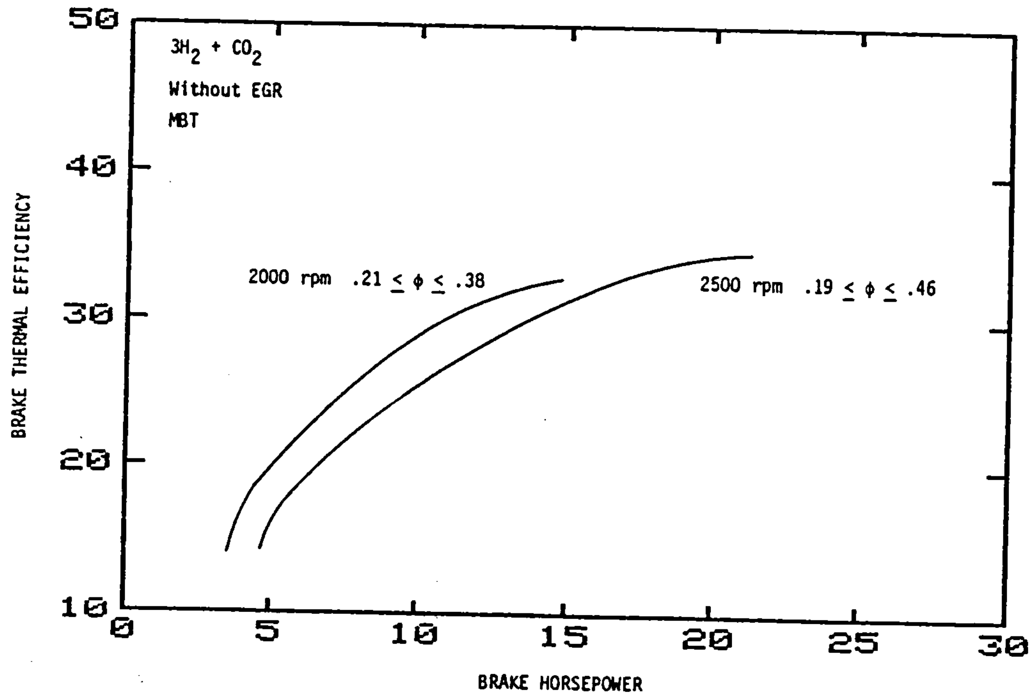

Figure 24: Effect of Engine Speed on the Variation of Brake Thermal Efficiency with Brake Horsepower for Steam Reformed Methanol.

brake power output increase as the engine speed is increased (within the range of conditions tested). As the engine speed is increased, overall heat losses increase, but the percent heat loss decreases resulting in an increase in efficiency (ref. 30). Also, friction losses increase with increased engine speed causing a decrease in efficiency. Eventually, the reduction in efficiency from friction losses will be greater than the increase in efficiency from lower percent heat losses, and the efficiency at constant manifold pressure will reach a maximum value and start to decrease with increased engine speed.

4.3.3 Effect of Throttling on Brake Thermal Efficiency. Because of hydrogen's wide flammability limits, the engine can be operated at WOT (unthrottled) and the power output controlled by varying the equivalence ratio. With dissociated methanol at 2000 rpm, the WOT power output can be varied from about 5 Bhp up to about 20 Bhp by varying the equivalence ratio from about 0.23 to 0.46 as can be seen in Figure 7. Also note from the figure that the same power outputs can be obtained by throttling the engine and increasing the equivalence ratio.

At low power outputs (i.e., 5 Bhp at 2000 rpm), some throttling from WOT to increase the equivalence ratio was beneficial in increasing the brake thermal efficiency at constant power output as shown in Figure 25. In this figure, the variation of brake thermal efficiency with equivalence ratio is shown for dissociated methanol at 2000 rpm with lines of constant brake horsepower added. At about 5 Bhp, throttling the engine from WOT to a manifold vacuum of 5 in. Hg and increasing the equivalence ratio resulted in an increase in efficiency of about 2 percentage points.

The increase in efficiency from throttling the engine from WOT and increasing the equivalence ratio while holding engine speed and power

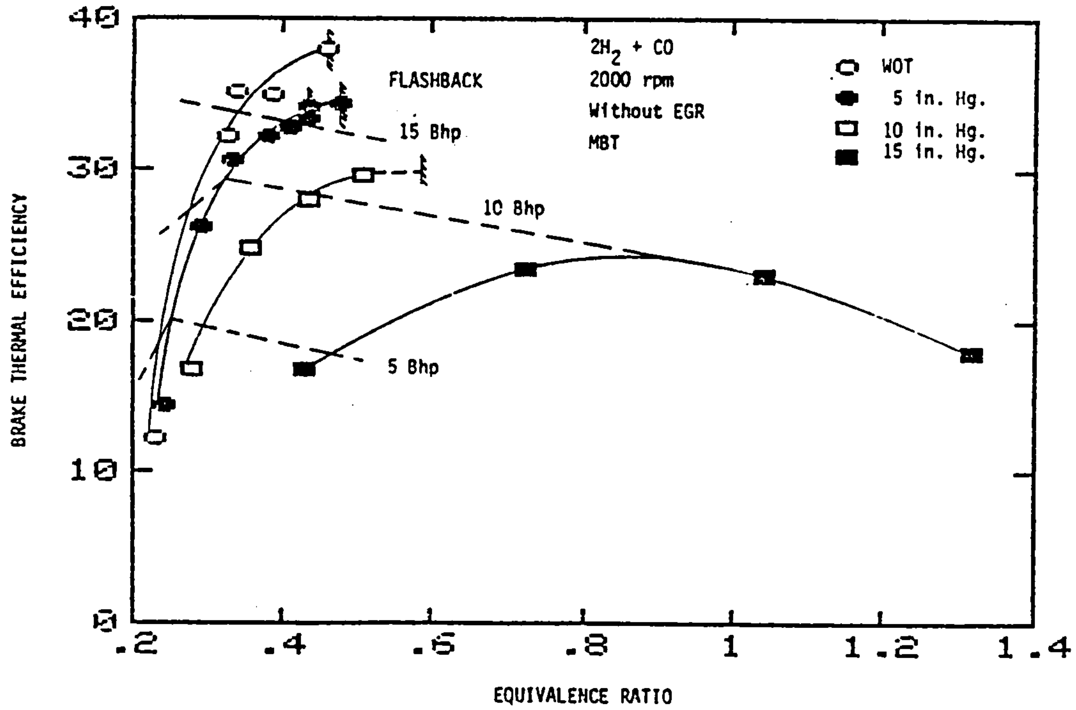

Figure 25: Effect of Throttle Setting on Brake Thermal Efficiency at Various Constant Brake Horsepower Outputs for Dissociated Methanol at 2000 rpm.

output constant decreased in value as the power output increased. Eventually, no increases in efficiency were obtained from throttling, and operation at WOT resulted in the highest efficiency. With dissociated methanol, operation at an equivalence ratio greater than 0.3 and WOT resulted in the highest efficiency, and with steam reformed methanol at equivalence ratios greater than 0.25 and WOT resulted in the highest efficiency. That is, to maintain the highest efficiency operating on dissociated methanol at constant engine speed, the engine should be operated unthrottled and the power output controlled with the equivalence ratio down to $\phi = 0.3$. To operate at powers below this point, the equivalence ratio should be enriched, and the power output lowered by throttling the engine. The same holds true for steam reformed methanol with the minimum equivalence ratio for WOT operation being 0.25.

The increase in efficiency from throttling the engine resulting from an increase in the combustion efficiency associated with enriching the fuel-air mixture. At the very lean equivalence ratios near the lean flammability limit of the fuel, there were large amounts of unburned fuel in the exhaust (as was evident from the CO emissions from dissociated methanol) (ref. 2). Increasing the equivalence ratio improved the combustion efficiency and reduced the amount of unburned fuel in the exhaust resulting in an increase in thermal efficiency greater than the reduction in efficiency from throttling. Eventually though, the increase in efficiency from improved combustion was offset by the throttling losses, and operation at WOT resulted in the highest efficiency.

4.4 Effect of Engine Operating Variables on Flashback

Operation on reformed methanol was limited to lean equivalence ratios (and low powers) by flashback, which is the violent propagation of a flame into the intake manifold. It could have been caused by a hot spot (exhaust valve, spark plug, residual gas) in the cylinder or by contact with hot exhaust gases in the intake manifold during valve overlap. Flashback is not unique to hydrogen-rich fuels but is more of a problem with them because of the low ignition energy required to initiate combustion and the high flame speed of hydrogen.

The objective of the investigation was to map the performance of an unmodified SI engine, so no modifications were made to extend the flashback limit. The engine was operated wih stock (or one heat range colder) spark plugs, without stock EGR, and with whatever cylinder deposits formed during baseline testing on indolene and liquid methanol. The effects of cylinder deposits, equivalence ratio, throttle, engine speed, and spark advance on flashback are presented and discussed below.

Several researchers (ref. 2 and 34) have investigated the possible causes of flashback operating on pure hydrogen. They concluded that the dominant cause of flashback was hot spots inside the cylinder. Based upon their results, it was assumed that flashback for reformed methanol was also caused by hot spots in the cylinder, and the following discussion of flashback reflects this assumption.

To determine if cylinder deposits were the cause of flashback, the engine was dismantled and cleaned of deposits which merely consisted of a light rust residue on the cylinder head and pistons. Some of the flashback operating conditions for dissociated methanol were retested. Little difference was found in the flashback points obtained with and without cylinder deposits in the engine. From this it was concluded that cylinder deposits were not the cause of flashback and that the lean flashback points measured were valid for the stock engine configuration. Hot spots such as the spark plug electrode or exhaust valve possibly caused flashback.

Flashback occurred at richer equivalence ratios as the throttle was closed (decreasing manifold pressure) and as the engine speed was increased. The increase in the value of the equivalence ratio at which flashback occurred with a decreased manifold pressure (closing the throttle) was the same as observed for operation on pure H_2 (ref. 2). Decreasing the manifold pressure at constant equivalence ratio lowered the flame speed (ref. 33) and cylinder wall temperatures (ref. 30). thus reducing the tendency of the fuel-air mixture to flash back. Increasing the equivalence ratio at constant manifold pressure had the opposite effect on flame speeds and cylinder temperatures resulting in an increased probability of flashback occurring. Therefore, flashback would occur at a richer equivalence ratio as the manifold pressure was decreased.

The reasons for the increase in the value of the equivalence ratio at which flashback occurred with increasing engine speed are unclear. It could possibly result from reduced residence times and faster flame speeds. The equivalence ratio at which flashback occurred for WOT increased from 0.42 to 0.48 for dissociated methanol and from 0.25 to 0.46 for steam reformed methanol when the engine speed was increased from 1000 to 2500 rpm. Increasing the engine speed increased cylinder and combustion chamber wall temperatures (ref. 30) and the flame speed (ref. 33). Since both of these factors tend to increase the probability of flashback occurring, the equivalence ratio at which flashback occurred should have decreased with increasing engine speed (at constant manifold pressure). This trend was observed for dissociated methanol at a manifold vacuum of 15 in. Hg.

About the only engine parameter that changed to cause the richer equivalence ratio where flashback occurred with increasing engine speed was the incoming air-fuel mixture velocity which increased proportionally to the increase in engine speed. If the incoming mixture velocity were greater than the flame speed of the mixture, flashback could not occur since the flame cannot propagate into the intake manifold. However, if the fuel-air mixture did preignite without the flame entering the intake manifold, power output should have declined since there would be compression of a burning gas mixture, and the peak pressure would not occur at the optimum crank angle.

Flashback occurred at leaner equivalence ratios with steam reformed methanol that with dissociated methanol. It was initially thought that the CO_2 in the steam reformed methanol would act like a built in "EGR" system and allow operation to richer equivalence ratios than dissociated methanol before flashback. It is not apparent from the values of equivalence ratio where flashback occurred that the CO_2 in steam reformed methanol did help in extending the flashback equivalence ratio. However, if the flashback points are compared on a mass percent H_2 in the fuel-air mixture, the CO_2 in steam reformed methanol did allow operation to mixtures with higher mass fraction of H_2 (before flashback occurred) than with dissociated methanol. Operation to higher concentrations of hydrogen in the fuel-air mixture with steam reformed methanol than with dissociated methanol was possible because the inert CO_2 in steam reformed methanol lowered combustion temperatures, thus reducing hot spot temperatures. Steam reformed methanol has about 5 to 10% less H_2 per lbm of fuel-air mixture than pure H_2 at lean equivalence ratios.

At some of the flashback points the spark timing was advanced and retarded by as much as 20° from MBT without affecting the equivalence ratio at which flashback occurred. Spark timing did not affect flashback because flashback was probably caused by a hot spot which preignited the fuel-air mixture early in the cycle well before the spark occurred.

4.5 Analysis of Exhaust Gas Energy

One of the possible methods in which to supply the energy necessary to reform methanol is to use waste energy in the engine exhaust. Several investigators (refs. 14-18, 20, 25) have designed and used reformers which operate on the energy in the exhaust gas. For proper reformer operation, the energy must be supplied above the temperature required for operation of the reformer. Most state-of-the-art reformers operate best in the temperature range from 480° to 570°F (250° to 300°C) and generally cannot tolerate temperatures in excess of 1200°F (650°C) without damaging the catalyst (refs. 15, 17). The total energy (per lbm of reformed fuel) required to convert the liquid feedstock fuel at 77°F (25°C) to the gaseous reformed products at 570°F (300°C) is 2045 Btu/lbm for dissociated methanol and 1430 Btu for steam reformed methanol. Heating of the fuel to 200°F (100°C) can be accomplished with engine coolant. The balance of the energy required to reform the fuel must then be supplied by the exhaust gas or an external source (1500 Btu/lbm for dissociated methanol and 990 Btu/lbm for steam reformed methanol). The minimum operating temperature of the reformer places a restriction on the "quality" (temperature) of the energy necessary to reform methanol, while the thermodynamics of the reforming process places a restriction on the minimum amount of energy required to reform methanol.

The purpose of this section is to determine whether enough energy exists in the exhaust gas at a high enough temperature to reform methanol for the operating conditions investigated. A first order analysis was performed using the exhaust gas temperature data obtained in the tests to determine the amount of energy available from the engine exhaust to reform methanol. A description of the measurements, calculations, and assumptions made in performing the analysis and a discussion of the results of the analysis are presented below.

Exhaust gas temperatures were measured with a thermocouple placed in the exhaust flow a short distance from the exhaust manifold runners. Since operation on reformed methanol was limited to lean equivalence ratios, exhaust gas temperatures were low compared to those from operation on liquid methanol. The temperature of the exhaust gas increased linearly with equivalence ratio at constant engine speed, and also increased with increasing engine speed. Exhaust gas temperatures were about equal for both reformed fuels at the same equivalence ratio.

The first assumption made in peforming the analysis was that the reformer operating temperature was 570°F (300°C). Thus, for energy to be available from the exhaust, the exhaust gas temperature had to be above this temperature. Exhaust gas temperatures with steam reformed methanol at 1000 rpm were never above this temperature in the range of equivalence ratios where operation was possible. For dissociated methanol at 1000, 2000, and 2500 rpm and steam reformed methanol at the latter two engine speeds, exhaust gas temperatures were above 570°F (300°C) for operation above an equivalence ratio of 0.3. Thus exhaust gas temperatures are too low at very lean equivalence ratios and low engine speeds to operate a reformer with the stock exhaust system.

Assumptions made in calculating the energy available from the exhaust were (1) that the exhaust gases could be cooled to the reformer operating temperature (this assumes 100% heat exchanger effectiveness and is the most optimistic case possible), and (2) that the specific heat of the exhaust gases was constant and equal to 0.26 Btu/lbm-°F. The energy available from the exhaust was then calculated by multiplying the specific heat of the gas by the temperature difference between the exhaust gas and the reformer. This value was then multiplied by the ratio of the total mass flow rate of the exhaust to the mass flow rate of the fuel into the engine to determine the amount of energy available per unit mass of fuel. The amount of energy available from the exhaust was then compared to the amount of energy required to completely reform methanol. It was assumed that the required amount of exhaust energy was that necessary to reform methanol or methanol-water mixture above 200°F, the temperature which was assumed could be reached by extracting energy from the engine cooling water.

It should be emphasized that this analysis represents the best possible case (100% heat exchanger effectiveness) whereby the maximum possible amount of methanol would be dissociated or reformed, the actual amount would be less. The analysis also does not consider the transient case, that is the question of whether the heat transfer rate would be great enough to effect the required vaporization and reformation rapidly enough.

Results of the analysis for reformed methanol at 2500 rpm are presented in Tables 2 and 3. Exhaust gas temperatures and energy available from the exhaust to reform methanol are listed as functions of equivalence ratio.

Table 2 : Energy Available from the Exhaust Gas for Dissociated Methanol at 2500 rpm

ϕ	T °F (C)	Energy[+] in Exh.	Deficit[+]	% of Required in Exh.	% Decrease in Eff.
0.25	610 (320)	244	1256	16.3	13.5
0.30	680 (360)	640	860	42.7	9.6
0.35	750 (400)	893	607	59.5	7.0
0.40	810 (430)	1057	443	70.5	5.2
0.45	860 (460)	1160	340	77.3	4.1
0.50	910 (490)	1221	279	81.4	3.3
0.60	980 (525)	1254	246	83.6	3.0
0.65	1010 (545)	1240	260	82.7	3.1
0.70	1030 (555)	1211	289	80.7	3.5

[+] Units of Btu/lbm of dissociated methanol.

Table 3: Energy Available from the Exhaust Gas for Steam Reformed Methanol at 2500 rpm

ϕ	T °F (C)	Energy[+] in Exhaust	Deficit[+]	% of Required in Exhaust
0.25	590 (310)	92	898	9.3
0.30	640 (340)	271	719	27.4
0.35	690 (365)	402	588	40.6
0.40	740 (390)	504	486	50.9
0.45	790 (420)	586	404	59.2
0.50	840 (445)	654	336	66.1
0.55	890 (475)	713	277	72.0
0.60	940 (505)	763	227	77.1

The values in the column labelled "Deficit" indicate the amoung of energy (above 200°F) still needed to completely reform the fuel assuming that the energy to heat the fuel to 200°F was available from the cooling water. Deficit energies were high at the lean equivalence ratios and were low at the rich equivalence ratios. There would never be enough energy available from the exhaust to completely reform methanol even though exhaust gas temperatures were high enough. For both fuels, the highest percentage of the energy that could be supplied from the exhaust to reform the fuel was about 80%.

For complete reformation of the liquid fuel, the deficit amount of energy would have to be supplied from another source. One such source could be the liquid fuel itself. If the deficit energy for dissociated methanol were supplied by burning some of the liquid methanol at stoichiometry and cooling the products to 570°F (300°C) the overall system thermal efficiency would decrease by the amounts listed in the last column of Table 6. These values are precent decreases in efficiences (based upon the LHV of liquid methanol). At the richer equivalence ratios (and higher exhaust gas temperatures), the thermal efficiency would be reduced by about 3 to 4% if methanol were burned to supply the deficit energy needs. If all of the energy above 200°F necessary to dissociate methanol were supplied by burning some methanol, the thermal efficiency would be decreased by about 16% negating all of the increase in efficiency resulting from the increase in LHV of the fuel. That is, operating on reformed methanol in this manner would result in only a very small increase in thermal efficiency over that of liquid methanol (Figure 10), and the advantage of using reformed methanol rather than liquid methanol in terms of higher engine efficiency would no longer exist.

This analysis was obviously very ideal in nature. If temperature gradients necessary for suitable heat transfer rates are considered, there would be even less energy available from the exhaust to reform methanol than is presented in the tables. The minimum temperature to which the exhaust gas could be cooled to operate a reformer on a vehicle is probably around 750°F (400°C). With this as the minimum temperature, there would be only about 50% of the energy necessary to reform methanol available in the exhaust. For exhaust gases to be greater than this temperature, the engine would have to be operated at equivalence ratios greater than about 0.4 and engine speeds above 1000 rpm with reformed methanol.

The energy available from the exhaust to reform methanol could be increased by increasing the exhaust gas temperature, resulting in higher reformer conversion efficiencies. The temperature could be increased by insulating the exhaust system to reduce heat losses.

It may be more advantageous to operate an engine on the products of incomplete reformation than from complete reformation. Since the incompletely reformed fuel would contain some unreacted methanol, the flame speed of the fuel mixture would be lower than for the completely reformed fuel. This may allow engine operation to richer equivalence ratios and higher powers without flashback, thus making partially reformed methanol a more attractive fuel.

Another means of taking advantage of reformed fuel at low power levels while still being able to attain higher power without flashback might be to use a liquid/gas combination fuel system such that the engine could be supplied with gaseous reformed fuel or liquid methanol. With such a system liquid fuel could be dissociated or reformed to fuel the engine at low power levels, and operation would switch to liquid fuel for high power.

4.6 Comparison of Reformed Methanol Results to Those of Others

Several researchers have investigated dissociated methanol and pure hydrogen as automotive engine fuels (refs. 1, 2, 15, 16, 21, 25, 35). No engine research with steam reformed methanol has been found. The purpose of this section is to determine if the dissociated methanol performance results obtained in this study are similar to those obtained by others, and if the performance of an engine operating on reformed methanol is similar to that of an engine operating on pure hydrogen. A comparison of dissociated methanol results to those of references 15 and 25 and a comparison of reformed methanol results to the pure hydrogen results of reference 2 are presented in Figures 26 and 27 and discussed below.

In figure 26, brake thermal efficiency is shown as a function of brake mean effective pressure (Bmep) for dissociated methanol with the results from this study and references 15 and 25 plotted. The brake thermal efficiency is plotted as a function of Bmep because a different size and type engine was used for each study. The basis for computing the efficiency for these two studies was the same as for this investigation, that is the LHV of liquid methanol. The results from references 15 and 25 are from engines with 14.0:1 compression ratio operating on the products from an actual reformer. Below about 40 Bmep, there was good agreement in Bte obtained in this study with the results of references 15 and 25 with the maximum difference in Bte between studies being about 2 percentage points. Above 40 Bmep, the Bte results from reference 25 dropped while the efficiency from this study and reference 15 continued to rise. At about 60

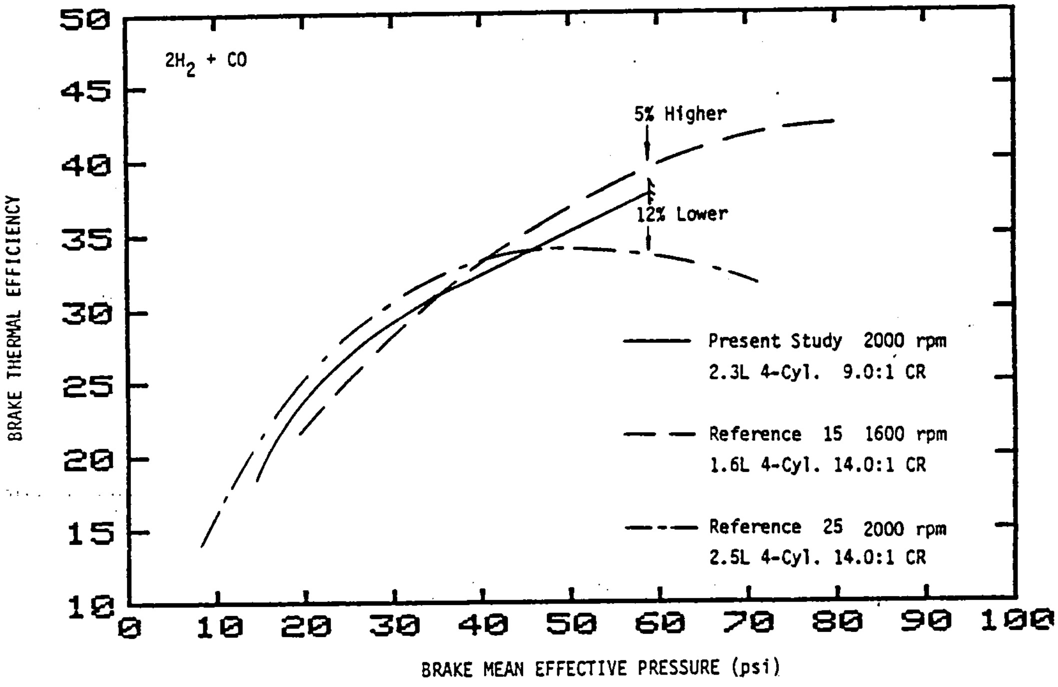

Figure 26: Comparison of the Variation of Brake Thermal Efficiency with Brake Mean Effective Pressure for Dissociated Methanol to Those of References [15,25].

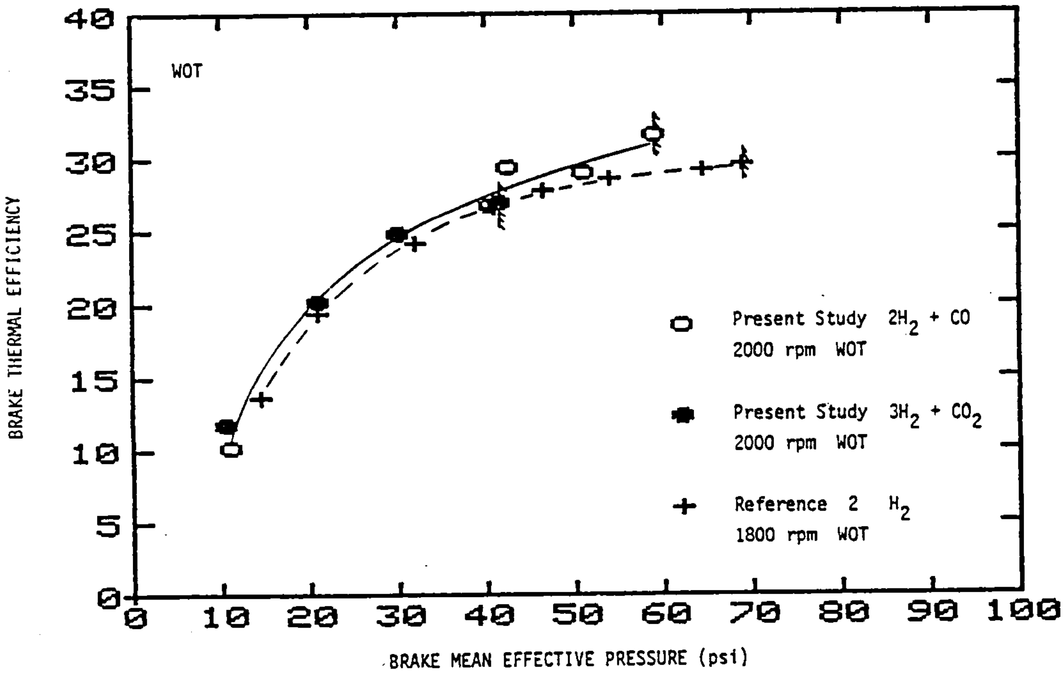

Figure 27: Comparison of the Variation of Brake Thermal Efficiency with Brake Mean Effective Pressure for Reformed Methanol and Hydrogen Results of Reference [2] (Efficiency Based upon the LHV of the Gaseous Fuel).

Bmep (where flashback occurred in this study), the Bte results from reference 15 were about 5% higher than those from this study, and the Bte from reference 25 were about 12% lower. The efficiency from both of those studies should be higher because of the higher compression ratio engines used. It is not known why the efficiency results of reference 25 decreased at the higher loads. Based upon differences in compression ratio of the engines used in this study and reference 15, there should be a larger difference in Bte between studies. The reason for the smaller than expected difference in Bte could be because the results from reference 15 are from operation at a slower engine speed, and the products from the actual reformer were at a higher temperature than the gaseous fuel of this study.

The WOT performance on both dissociated and steam reformed methanol was very similar to that reported in reference 2 for engine operation on pure H_2 as shown in Figure 27. In the figure, the WOT Bte is plotted as a function of Bmep with the reformed methanol results from this study (based upon the LHV of the gaseous fuel) and the pure hydrogen results from the reference 2 plotted. The brake thermal efficiency results from both studies were about the same, but operation was possible to higher loads with hydrogen than with reformed methanol before flashback occurred. Differences in engine performance with reformed methanol and pure hydrogen were small as expected since reformed methanol is primarily hydrogen.

Although comparisons are not exact because of differences in engine types, sizes, compression ratios, test engine speeds, and fuel composition used in each study, the comparisons do show that the reformed methanol results from this study are in general agreement with the results obtained by others.

5. SUMMARY

The most important findings of this investigation are summarized below:

- Reformed methanol, either dissociated or steam reformed, can be used as a fuel for an unmodified spark ignition engine, but operation is limited to low power outputs by the occurrence of flashback into the intake manifold.
- Over the power range where engine operation was possible it was found that under the ideal conditions investigated, up to 25% increase in thermal efficiency with reformed methanol is possible. However, these results were obtained using gaseous fuels simulating 100% conversion of liquid methanol to dissociated or steam reformed methanol, and assuming that there would be sufficient energy in the exhaust to accomplish complete conversion. When the efficiency was computed based upon the LHV of the gaseous fuel, there was little or no difference in the efficiencies between reformed and liquid methanol.
- Under the conditions where engine operation was possible, calculations indicated that there would not be sufficient energy in the exhaust to completely reform methanol.
- Dissociated methanol was found to be a better fuel than steam reformed methanol for the engine tested. This was based upon power output, emissions levels, and flashback resistance.
- In general the exhaust emissions were lower from the reformed gaseous fuels than from liquid methanol.

6. CONCLUSIONS

- An automobile could not be operated over the required power range exclusively with reformed methanol - a supplementary fuel (energy) supply would be required to reach the higher powers.

- The use of reformed methanol (compared to liquid methanol) may result in a small improvement in thermal efficiency in the low power range, with the increase in efficiency resulting from the greater heating value of the reformed fuel.

- Dissociated methanol is a better fuel for a spark ignition engine than steam reformed methanol.

- Use of reformed methanol (compared to liquid methanol) in a spark ignition engine may result in lower exhaust emissions.

7. RECOMMENDATIONS

Based upon the results of the investigation, the following recommendations are made:

- Since complete conversion of the liquid methanol may be unattainable, experimentation should be conducted to determine if there exists an optimum conversion efficiency. Optimum would clearly involve a tradeoff between performance and emissions. The use of fuel mixtures (mixtures of reacted and unreacted methanol) may also extend or eliminate flashback limits and allow operation at higher power.

- Since power outputs are low with completely reformed methanol, experimentation should be conducted with a two-phase fuel scheme whereby dissociated methanol is supplemented by liquid or two-phase (liquid and vapor) methanol at high powers.

- Any further research should be conducted with dissociated methanol rather than steam reformed methanol.

APPENDIX A

EPA DRIVING CYCLE PROJECTIONS

To compare the results of this investigation with the results of others, a three-point, mini-driving cycle method was used to approximate operation of a 3000 pound vehicle over the EPA urban driving cycle. The purpose of these tests was to predict the energy consumption and emissions from various fuels over the EPA driving cycle. The procedures followed were those outlined in reference 26 and by a panel on Hydrogen Automotive Engine Performance, Fuel Ecology and Emissions Potential at the World Hydrogen Energy Conference IV, June 16, 1982, Dr. Carl Kukkonen of Ford Motor Co., Chairman.

A.1. Experimental Procedures

To simulate the EPA urban driving cycle, the engine was operated at the three steady-state engine speeds and load conditions listed in Table A1, and the results multiplied by the weighting factors shown for each condition. The total work required to propel the vehicle over the 7.5 mile driving cycle was 3 hp-hr (sum of weighting factors).

Table A1: EPA Driving Cycle Simulation Test Conditions (ref. 26)

Engine Speed (rpm)	Torque (ft-lbf)	Weighting Factor (hp-hr)
1000	13.4	0.49
1250	32.9	1.35
1700	66.7	1.16

Tests were conducted at the above operating conditions with liquid methanol, methane, and natural gas at equivalence ratios of 0.9 and 1.0. Spark timing was set at MBT at each operating condition, and the conventional EGR system was disconnected.

Data were taken at each of the three conditions and two equivalence ratios with all three fuels to obtain the brake specific fuel consumption and the specific emissions of hydrocarbons, NO_x, and CO at each point. These values were multiplied by the weighting factor for each point and summed for each fuel and equivalence ratio, to obtain the projected totals over the cycle. These sums were then divided by the simulated total distance travelled (7.5 miles) to obtain fuel consumption and mass emissions per mile. The fuel consumption values were then multiplied by the LHV of the fuel to obtain the energy consumption per mile projections over the driving cycle.

A.2 Results

The results of the driving cycle simulation are presented in Table A2. The equivalence ratios shown in the table are nominal values with the actual equivalence ratios varying by ± 0.03 from the nominal value. As can be seen at $\phi = 0.9$, the energy consumption, hydrocarbon emissions, and NO_x emissions were approximately the same for all three fuels, but CO emission predictions differed by a large amount. As expected, at the higher equivalence ratio was increased, the energy consumption, hydrocarbon emissions, and CO emissions increased while the NO_x emissions decreased.

Table A2: EPA Driving Cycle Projections

Fuel	ϕ	HC[1]	NO_x	CO	Energy Consumption[2]
Methanol	0.9	0.5	1.2	12.0	4400
Methane	0.9	0.7	1.3	4.2	4200
Natural Gas	0.9	0.5	0.8	0.6	4200
Methanol	1.0	0.6	0.6	39.0	4700
Methane	1.0	1.2	0.4	44.0	5000
Natural Gas	1.0	0.5	0.4	8.9	4600

[1] Emissions are presented as gm/mi.

HC reported as carbon, and NO_x reported as NO_2.

[2] Btu/mi.

Differences in performance and emissions projections for the fuels were larger at ϕ=1.0 than at ϕ=0.9. This larger scatter resulted from the nominal variation of equivalence ratio at each test condition. The knee of the CO versus ϕ curve is near stoichiometry, and a small perturbation in ϕ from ϕ =1.0 results in a large change in the magnitude of the CO emissions.

As mentioned earlier, these tests were conducted to obtain results that could be compared to those of others. Shown in Figure A1 are the results of other investigations for the same driving cycle simulation. These results were presented at the Panel meeting cited above. As can be seen by comparing values, the energy consumption projections and NO_x projections obtained in this study are very close to the Methanol results presented at that meeting.

Fuel	Engine	BTU/mile	NO_x gm/mi	HP/lb.Inertia Wt.	Source
Gasoline	Escort 2.2ℓ	4880	0.7	~ .028	Ford Motor Co.
IDI Diesel (Prechamber)	2.2ℓ	4180	0.8	~ .022	Ford Motor Co.
DI Diesel (Open Chamber)	Ford 2.2ℓ	3630	0.8	~ .022	Ford Motor Co.
Methanol	Escort 2.2ℓ	4440	0.9	.030	Ford Motor Co.
Hydrogen	Pre-Intake Valve Closure	4042	0.6	.023	U. of Miami
Hydrogen	Post-Intake Valve Closure	5822	2.3	-	U. of Miami
Hydrogen	Mitsubishi Turbocharged aftercooled	7370	1.4	-	Hydrogen Consultants, Inc.
Hydrogen	Caterpillar Turbocharged aftercooled	4074	0.7	-	Hydrogen Consultants, Inc.
Hydrogen	BMW 520	5200	-	-	DFVLR (Germany)
Gasoline	BMW 520	5200	-	-	DFVLR (Germany)
Dissociated Methanol	Citation 2.5ℓ	4200	0.6	.030	SERI
Dissociated Methanol	Citation 1.6ℓ	3560	-	.022	SERI

Figure A1: EPA Driving Cycle Projections Determined by Others Using Three-Point Mini-driving Cycle.

(Presented by Panel on Hydrogen Automotive Engine Performance, Fuel Economy and Emissions Potential at the World Hydrogen Energy Conference IV, June 16, 1982).

APPENDIX B

PRESENTATION OF RESULTS

The complete performance and emission results for reformed methanol, liquid methanol, indolene, methane, and natural gas are presented and discussed in this appendix.

The performance results are presented as plots of power, brake thermal efficiency (Bte), and brake specific fuel consumption as functions of equivalence ratio. Brake horsepower values were not corrected to standard conditions because the correction factors were small (less than 2%). The thermal efficiency results presented are based upon the constant pressure lower heating value (LHV) of the fuel. The LHV of the fuels used in the thermal efficiency calculations are listed in Table B1 along with the stoichiometric air-fuel ratio of the fuels. The Bte results for the reformed fuels are based upon the LHV of the liquid fuel mixture that would be stored in the tank and later reformed. Presented in this manner, the Bte results are for the engine-reformer system, assuming 100% conversion of liquid to dissociated or steam reformed methanol, and therefore, include the increase in efficiency resulting from the endothermic reforming reaction. The brake specific fuel consumption (which is inversely proportional to the brake thermal efficiency) results are also presented because this would be a more useful parameter to use when sizing fuel storage and delivery systems.

The emissions results are presented as plots of hydrocarbons, oxides of nitrogen, aldehydes, CO, CO_2, and O_2 as functions of equivalence ratio. The results presented were measured from the composite exhaust and include the effects of maldistribution on emissions. The hydrocarbons, NO_x, and aldehyde emissions are presented as mass emissions with units of grams per

Table B1: Stoichiometric Air-Fuel Ratios and Lower Heating Values

Fuel	Air-Fuel Ratio	LHV (Btu/lbm)
$2H_2 + CO$	6.45	10290
$3H_2 + CO_2$	4.13	6230
CH_3OH	6.45	8570
$CH_3OH + H_2O$[1]	4.13	5120
CH_4	17.19	21210
Natural Gas[2]	16.71	20990
Indolene	14.60	18870

[1] Used in calculating the efficiency for steam reformed methanol when the efficiency was based upon the LHV of the liquid fuel mixture.

[2] See Table B5 for composition.

indicated horsepower-hour (gm/ihp-hr) to be consistent with other published data (refs. 10, 11 & 29). Presented in this manner, the emissions results are independent of engine size or type. The hydrocarbons mass emissions are based upon the molecular weight of carbon, and the NO_x mass emissions are based upon the molecular weight of NO_2. It is commonly accepted that formaldehyde comprises 70% or more of the aldehydes present; therefore, the aldehyde mass emissions are reported as formaldehyde (ref. 10). CO, CO_2, and O_2 emissions are presented as volume percentages of the exhaust gas on a dry basis.

The results are grouped according to fuel type - reformed methanol, liquid methanol/indolene, and methane/natural gas - and are further grouped according to engine test speed. For example, in the following reformed methanol results section, the 2000 rpm performance and emissions results are presented together. The reformed methanol results obtained at 1000, 2000, and 2500 rpm are presented and discussed first, followed by the liquid methanol and indolene results obtained at 2000 rpm without EGR. In the last part, the methane and natural gas results obtained at 1000 and 2000 rpm are presented and discussed.

Results were obtained for indolene for two reasons. First, indolene baseline results were needed to determine that the test engine's baseline performance was the same as that obtained by others with similar engines. Second, they were needed in order to show that differences in engine performance between liquid methanol and indolene which were obtained were the same as observed by other investigators (i.e., validate the liquid methanol baseline engine performance results).

B.1. Reformed Methanol Results

Results from operation on reformed methanol at 1000, 2000, and 2500 rpm without EGR are presented in this section. The ambient conditions and range of intake manifold and coolant temperatures for the tests are listed in Table B2.

Plots of brake horsepower, brake specific fuel consumption, and brake thermal efficiency as functions of equivalence ratio for reformed methanol at 2000 rpm are shown in Figures B1 through B6. As can be seen in Figures B1 and B2, the power output increased rapidly as the equivalence ratio was increased until flashback occurred. The maximum power output with dissociated methanol and steam reformed methanol were 20.8 and 14.7 Bhp, respectively, and occurred at equivalence ratios of 0.46 and 0.38 (flashback ϕ).

Table B2: Reformed Methonal Test Conditions

Fuel	Speed rpm	$T_{ambient}$ F	$P_{ambient}$ in. Hg	T_{intake} F	$T_{coolant}$ F
$2H_2+CO$	2000	83	29.78	95-100	195-200
$2H_2+CO$	2500	81	29.77	95-100	190-200
$2H_2+CO$	1000	84	29.75	95-115	190-200
$3H_2+CO_2$	2000	89	29.75	95-105	185-200
$3H_2+CO_2$	2500	85	29.80	90-100	190-200
$3H_2+CO_2$	1000	85	29.80	95-110	185-195
$2H_2+CO$ +	2000	77	30.15	80- 85	185-195

+ Retest Flashback points after cleaning engine.

No constant manifold vacuum curves greater than 10 in. Hg are shown for steam reformed methanol because operation at very low manifold pressures and 2000 rpm was not possible. Either not enough power was produced to drive the dynamometer at the test speed or flashback occurred before the test speed was reached.

Flashback occurred at richer equivalence ratios as the throttle was closed (higher manifold vacuum). This was also observed by the University of Miami (ref. 2) operating on pure hydrogen. Flashback did not occur at a manifold vacuum of 15 in. Hg operating on dissociated methanol at 2000 rpm (and 1000 rpm), but did occur at 2500 rpm (fig. B33). Two flashback points are shown in Figure B1 for dissociated methanol at WOT and 5 in. Hg manifold

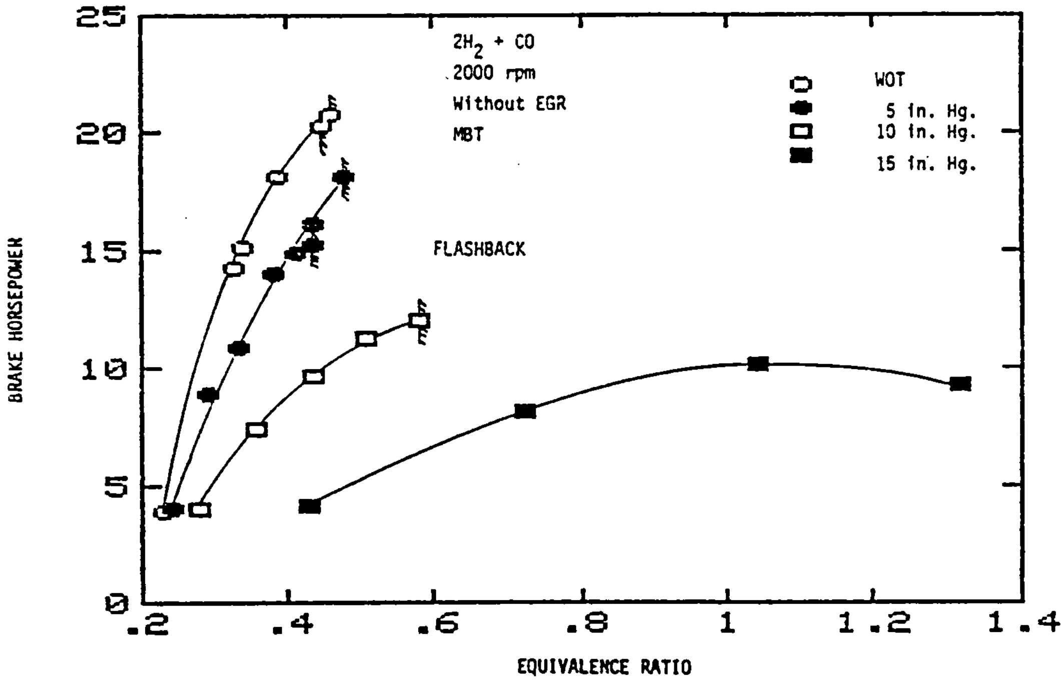

Figure B1: Effect of Equivalence Ratio on Brake Horsepower at Various Throttle Settings for Dissociated Methanol at 2000 rpm, without EGR.

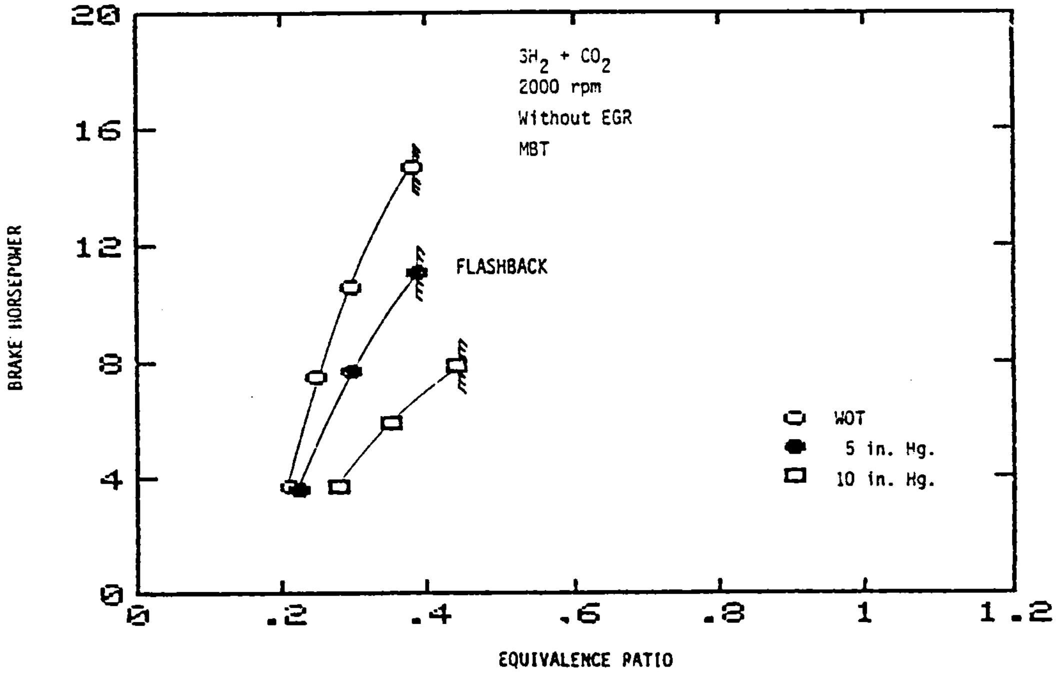

Figure B2: Effect of Equivalence Ratio on Brake Horsepower at Various Throttle Settings for Steam Reformed Methanol at 2000 rpm, without EGR.

vacuum because these two flashback points were retested after the engine was cleaned of deposits, and as can be seen, there were not significant changes in the flashback points.

The brake specific consumption, Figures B3 and B4, decreased as the equivalence ratio was increased. At the lowest throttle tested with dissociated methanol (15 in. Hg) the Bsfc decreased to a minimum value, after which, it increased with increasing equivalence ratio. The dashed portion of the 10 in. Hg manifold vacuum Bsfc curve in Figure B3 indicates that operation at that flashback point was not long enough to obtain a complete set of data and that the Bsfc (or the Bte) was not calculated at that point.

Shown in Figures B5 and B6 are plots of brake thermal efficiency as a function of equivalence ratio for reformed methanol. The Bte is inversely proportional to the Bsfc, and as can be seen, the Bte increased rapidly with increasing equivalence ratio. The maximum Bte measured from operation with dissociated and steam reformed methanol was 38% and 33%, respectively, and occurred at each fuel's WOT flashback point. To convert the efficiency based upon the LHV of the liquid fuel mixture to the efficiency based upon the LHV of the gaseous fuel, divide the efficiencies shown in the figures by 1.20 for dissociated methanol and 1.22 for steam reformed methanol. That is, based upon the LHV of the gaseous fuels, the maximum efficiencies were 31.7% for dissociated methanol and 27% for steam reformed methanol.

Hydrocarbon, NO_x, aldehyde, CO, CO_2, and O_2 emissions are plotted as functions of equivalence ratio in Figures B7 through B17. Hydrocarbon emissions were very low from reformed methanol as shown in Figures B7 and B8, and were not affected by equivalence ratio or manifold pressure. NO_x emissions from reformed methanol were low at the very lean equivalence

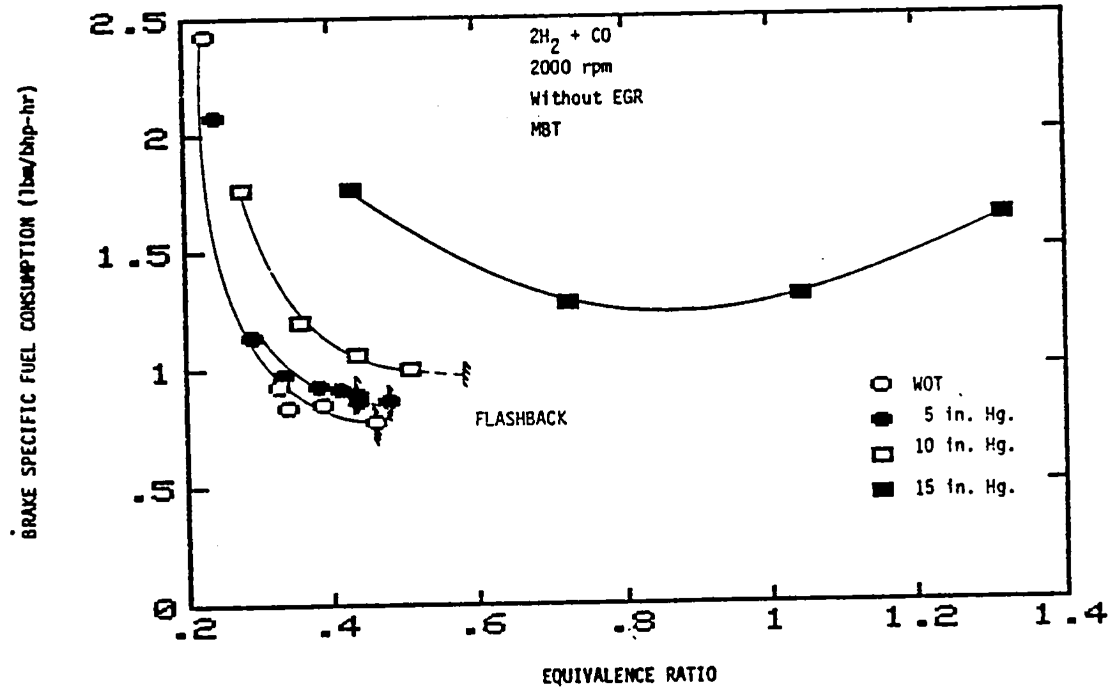

Figure B3: Effect of Equivalence Ratio on Brake Specific Fuel Consumption at Various Throttle Settings for Dissociated Methanol at 2000 rpm, without EGR.

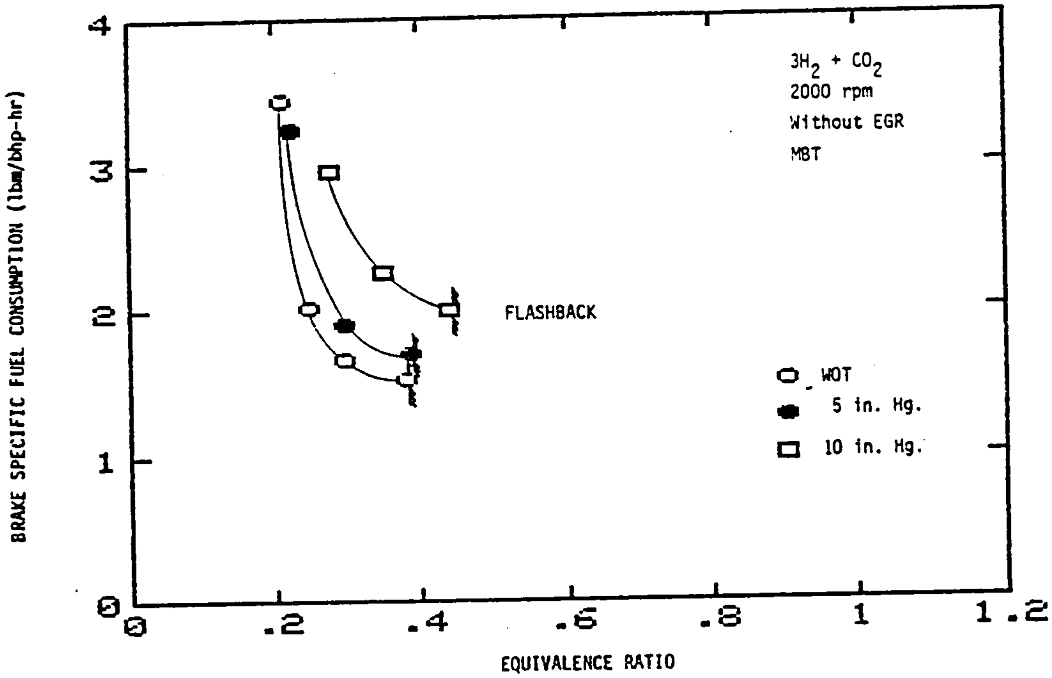

Figure B4: Effect of Equivalence Ratio on Brake Specific Fuel Consumption at Various Throttle Settings for Steam Reformed Methanol at 2000 rpm, without EGR.

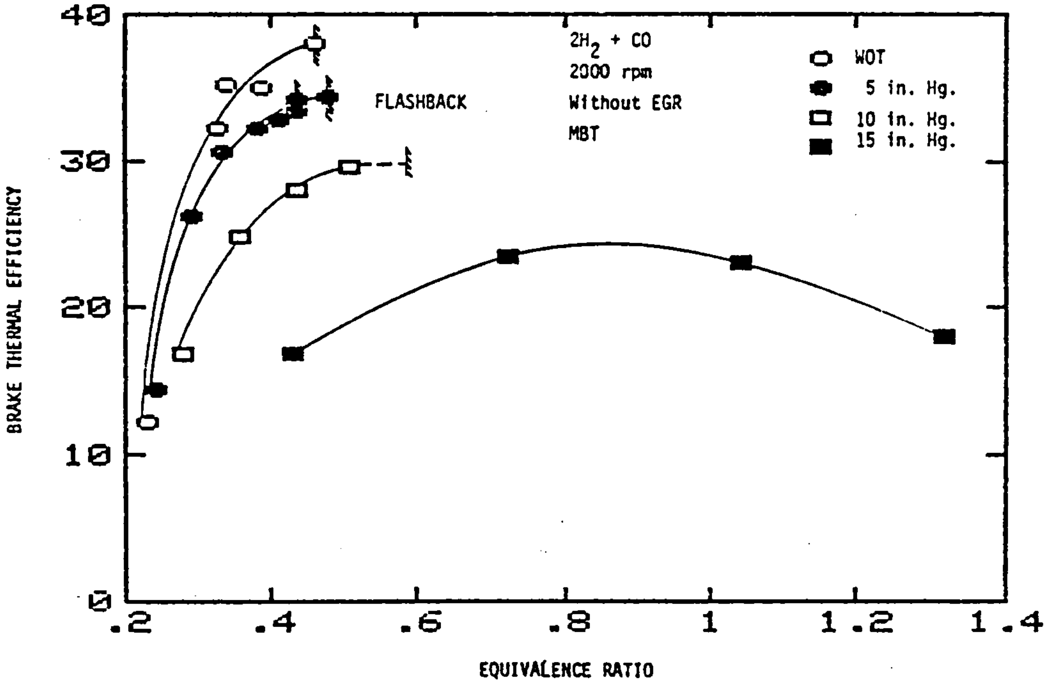

Figure B5: Effect of Equivalence Ratio on Brake Thermal Efficiency at Various Throttle Settings for Dissociated Methanol at 2000 rpm, without EGR (Efficiency Based upon the LHV of Liquid Methanol).

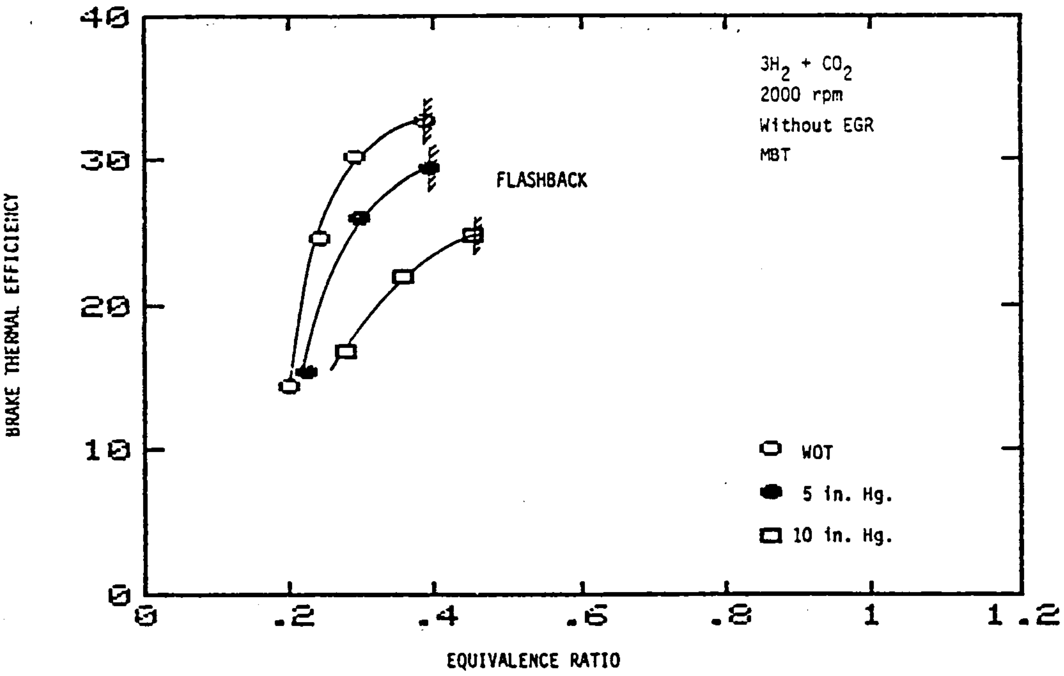

Figure B6: Effect of Equivalence Ratio on Brake Thermal Efficiency at Various Throttle Settings for Steam Reformed Methanol at 2000 rpm, without EGR (Efficiency Based upon the LHV of an Equimolar Methanol-Water Mixture).

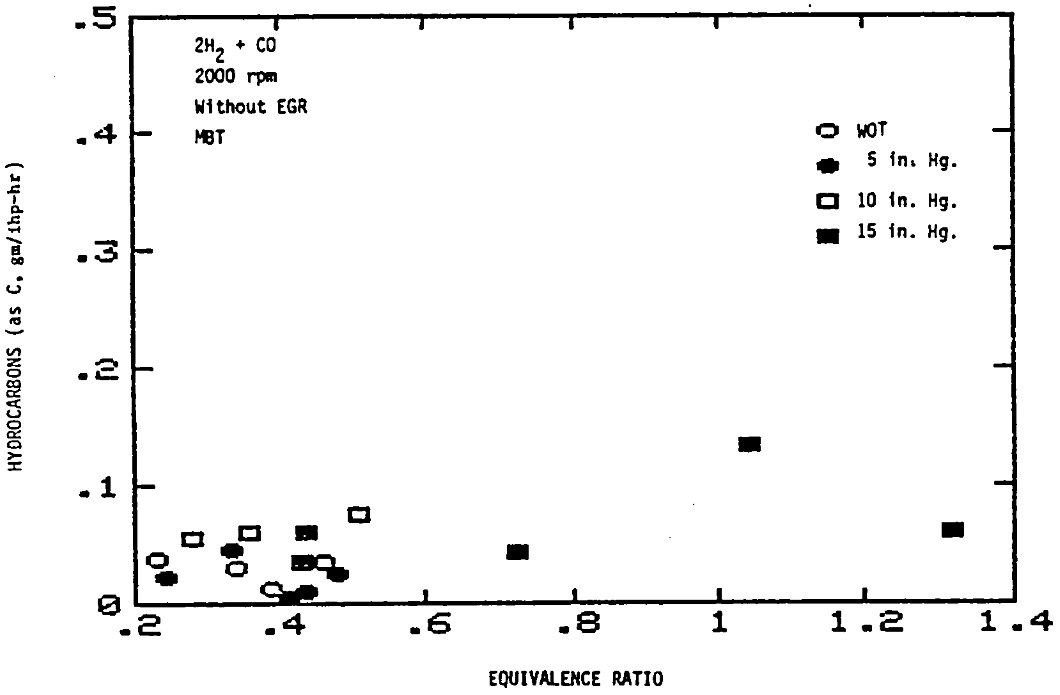

Figure B7: Effect of Equivalence Ratio on Hydrocarbon Emissions at Various Throttle Settings for Dissociated Methanol at 2000 rpm, without EGR.

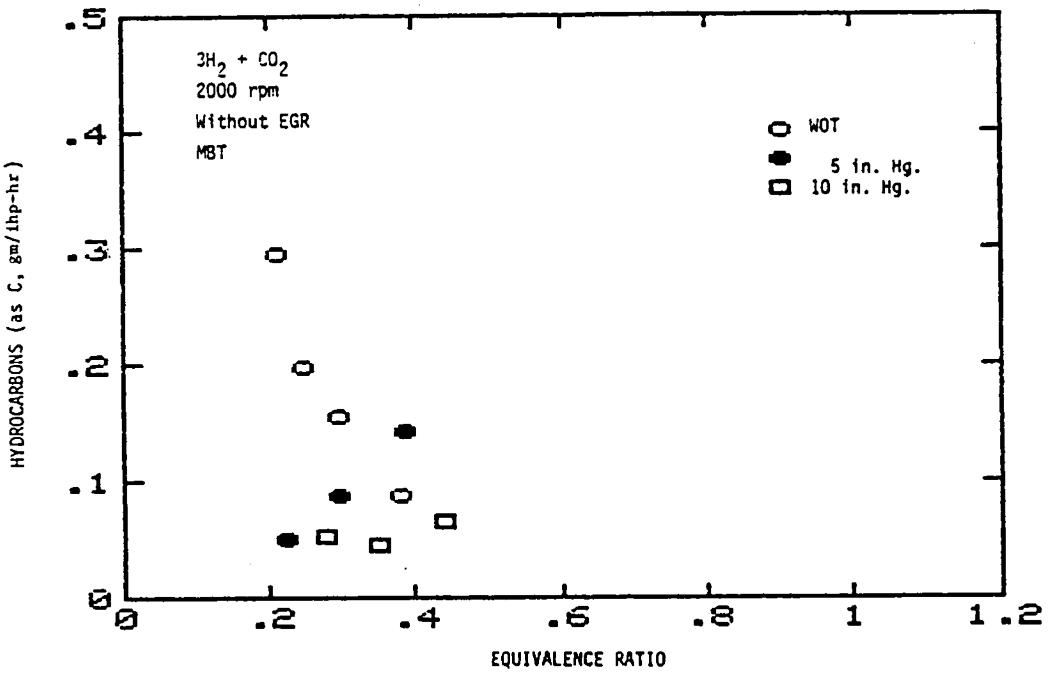

Figure B8: Effect of Equivalence Ratio on Hydrocarbon Emissions at Various Throttle Settings for Steam Reformed Methanol at 2000 rpm, without EGR.

ratios as shown in Figures B9 and B10 but were high from dissociated methanol at equivalence ratios near stoichiometry. Aldehyde emissions were also low from reformed methanol as shown in Figures B11 and B12 with the maximum aldehyde emission measured being about 0.06 gm/ihp-hr. Limited data were taken of aldehyde emissions because the wet chemistry method used to measure aldehydes was time consuming and tedious, and the aldehyde measurements were very low.

CO, CO_2, and O_2 were strong functions of equivalence ratio, and throttle setting had little effect on these emissions. There were no CO emissions from steam reformed methanol in the range of equivalence ratios where operation without flashback was possible. CO emissions from dissociated methanol, shown in Figure B13, increased rapidly as the equivalence ratio was decreased below $\phi = 0.35$, and were low for equivalence ratios greater than 0.35 up to near stoichiometry where they began to increase again.

CO_2 and O_2 emissions from reformed methanol as functions of equivalence ratio are shown in Figures B14 through B17 and were similar to those from other fuels. CO_2 emissions increased with increasing equivalence ratio and peaked near stoichiometry, after which, they decreased. O_2 emissions decreased with increasing ϕ and were very low at equivalence ratios greater than one (fuel rich mixture).

Similar performance and emissions results were obtained with reformed methanol at 1000 and 2500 rpm and are presented in Figures B18 through B47. The results are presented in the same order as the 2000 rpm results.

One thing that should be noted is that operation on steam reformed methanol at 1000 rpm was limited by flashback to extremely lean eqivalence ratios, resulting in a maximum power output of 3 Bhp. Also, the difference

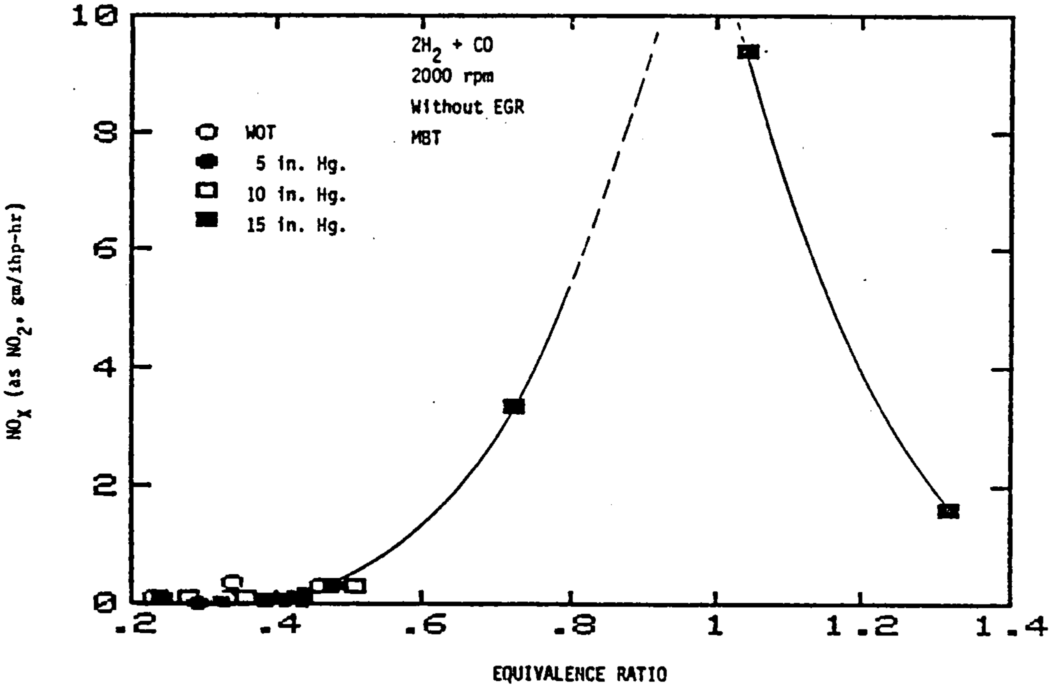

Figure B9: Effect of Equivalence Ratio on Oxides of Nitrogen Emissions at Various Throttle Settings for Dissociated Methanol at 2000 rpm, without EGR.

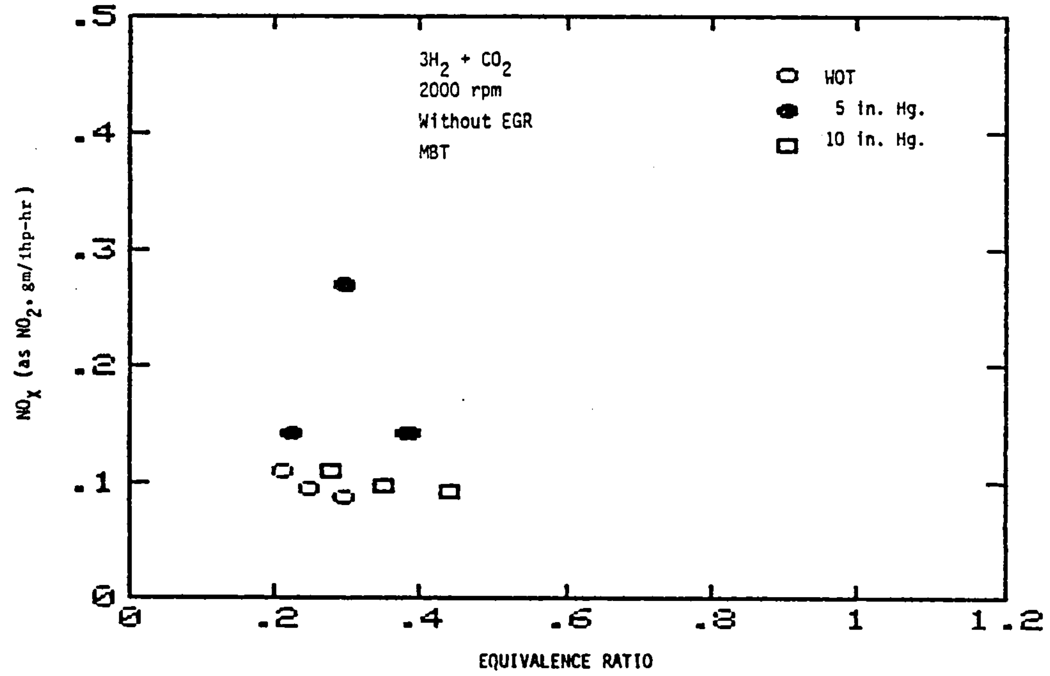

Figure B10: Effect of Equivalence Ratio on Oxides of Nitrogen Emissions at Various Throttle Settings for Steam Reformed Methanol at 2000 rpm, without EGR.

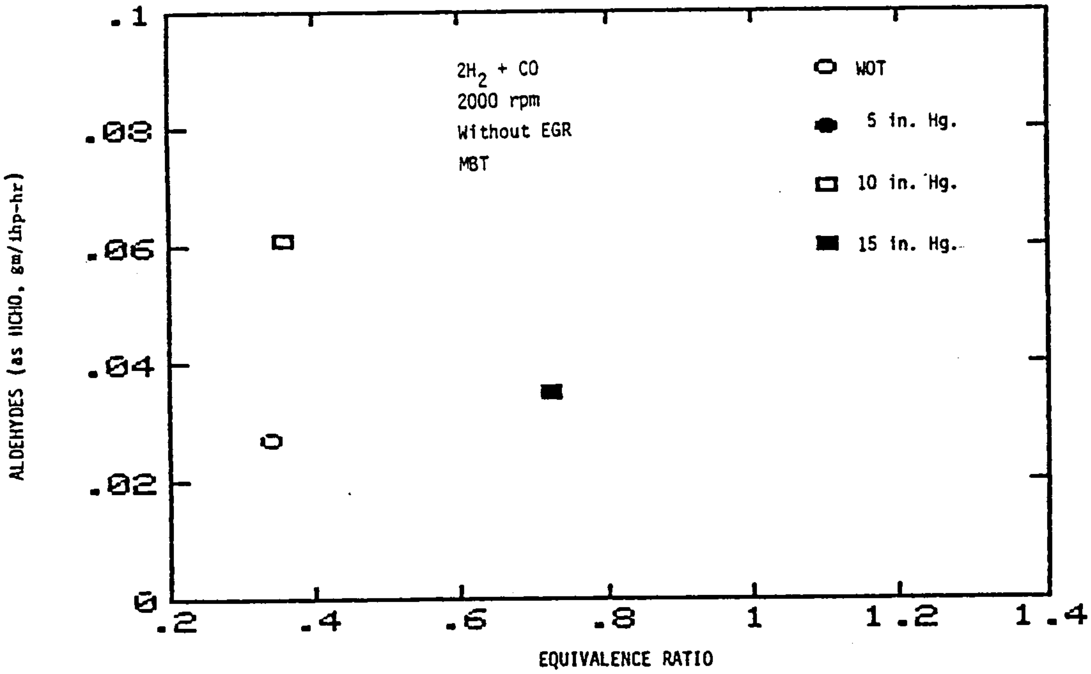

Figure B11: Effect of Equivalence Ratio on Aldehyde Emissions at Various Throttle Settings for Dissociated Methanol at 2000 rpm, without EGR.

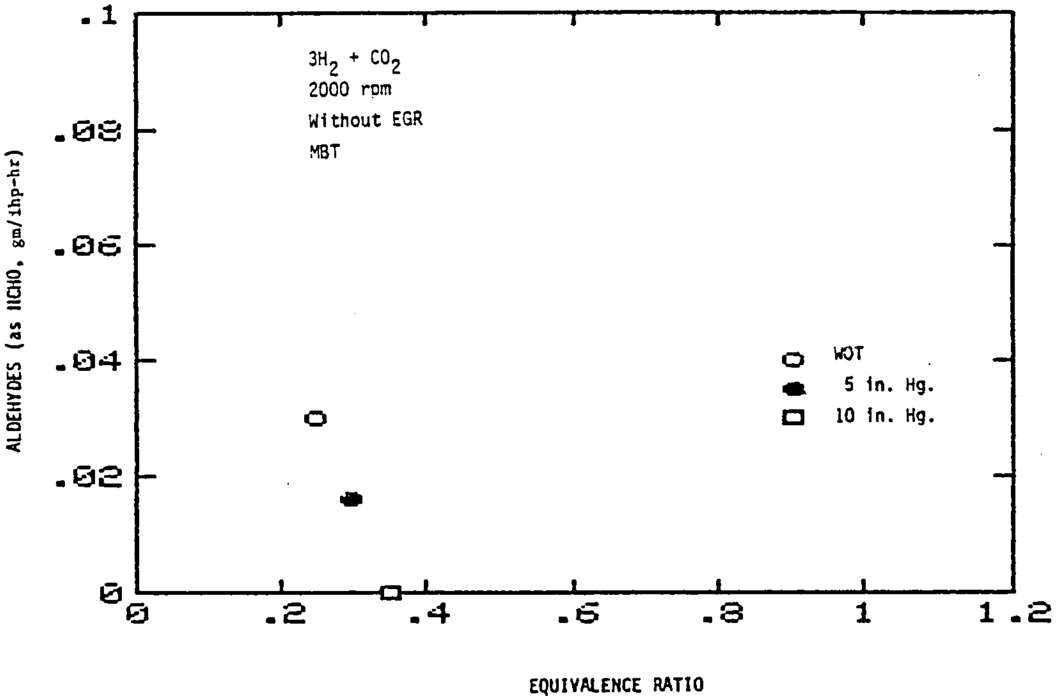

Figure B12: Effect of Equivalence Ratio on Aldehyde Emissions at Various Throttle Settings for Steam Reformed Methanol at 2000 rpm, without EGR.

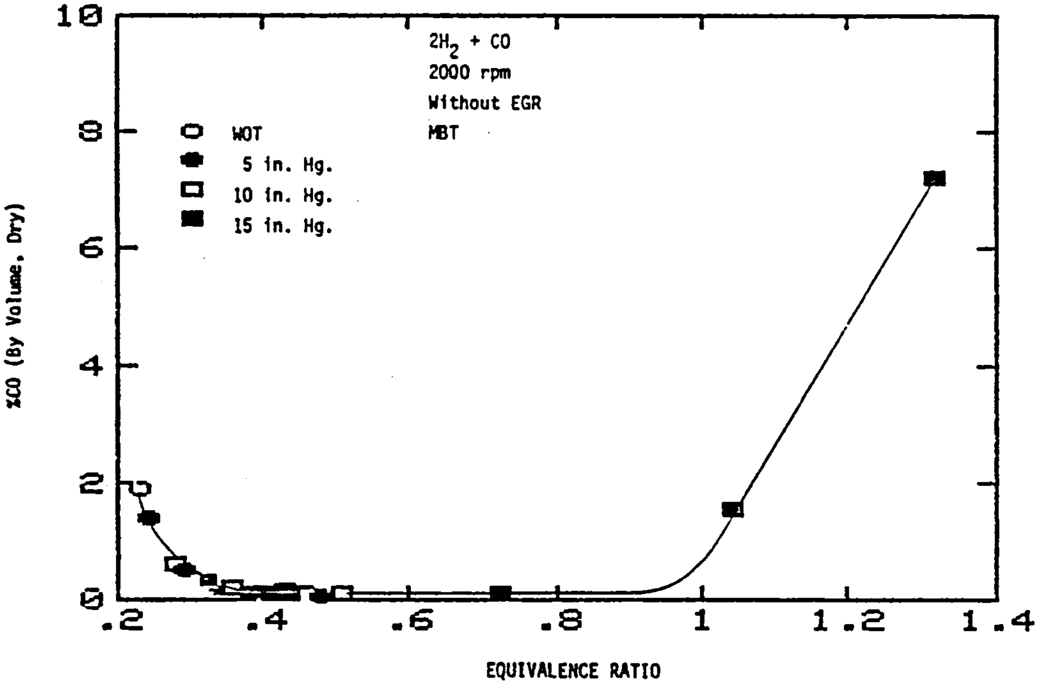

Figure B13: Effect of Equivalence Ratio on Carbon Monoxide Emissions at Various Throttle Settings for Dissociated Methanol at 2000 rpm, without EGR.

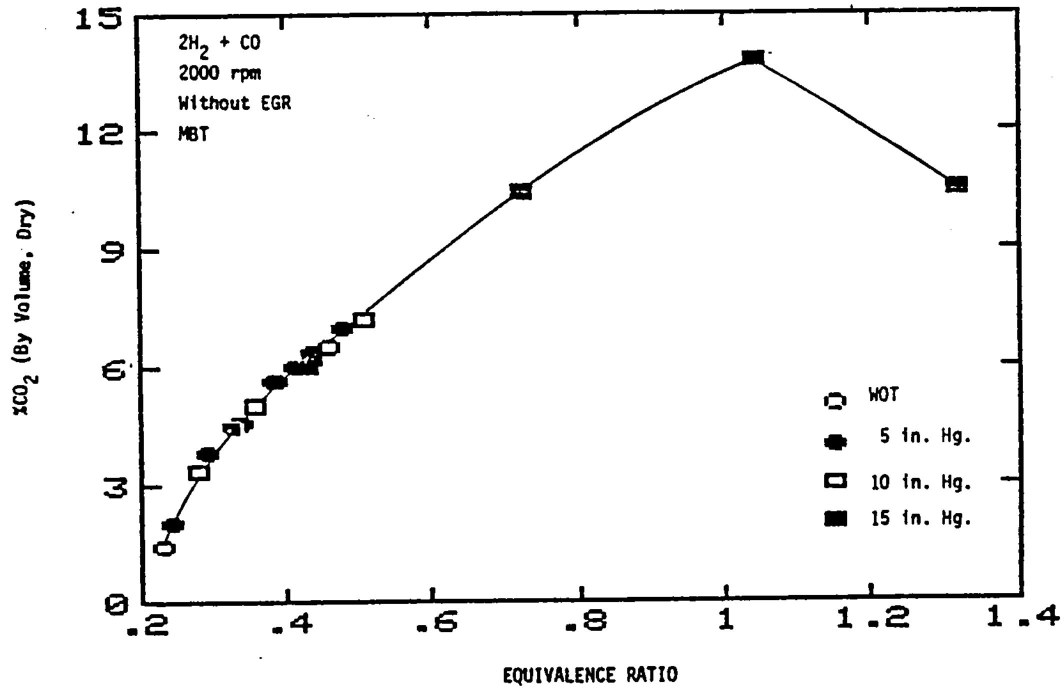

Figure B14: Effect of Equivalence Ratio on Carbon Dioxide Emissions at Various Throttle Settings for Dissociated Methanol at 2000 rpm, without EGR.

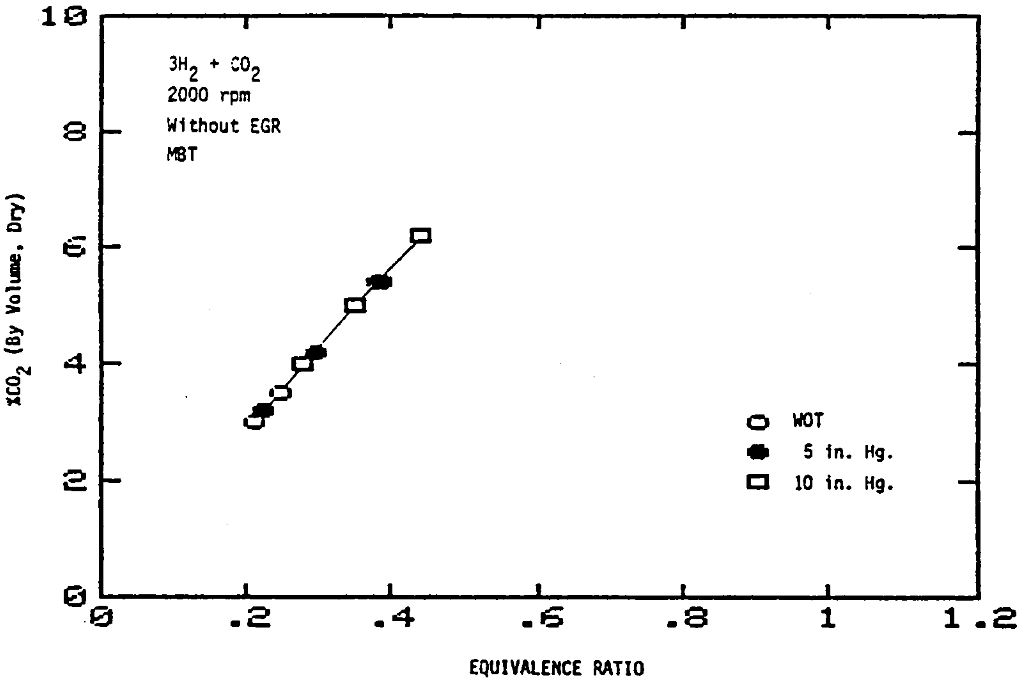

Figure B15: Effect of Equivalence Ratio on Carbon Dioxide Emissions at Various Throttle Settings for Steam Reformed Methanol at 2000 rpm, without EGR.

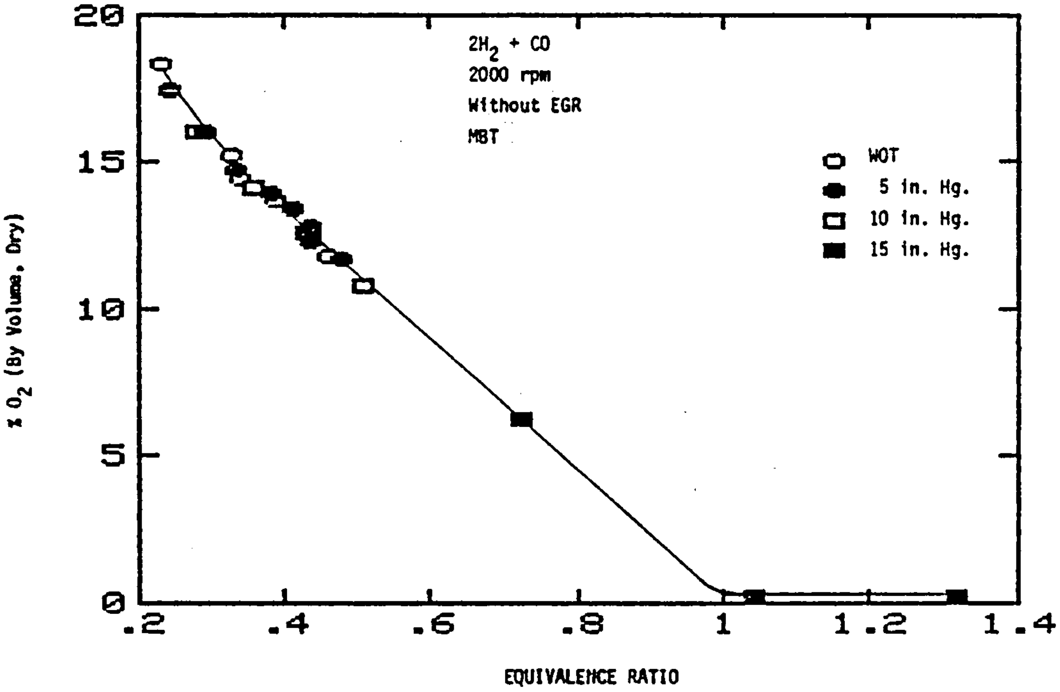

Figure B16: Effect of Equivalence Ratio on Oxygen Emissions at Various Throttle Settings for Dissociated Methanol at 2000 rpm, without EGR.

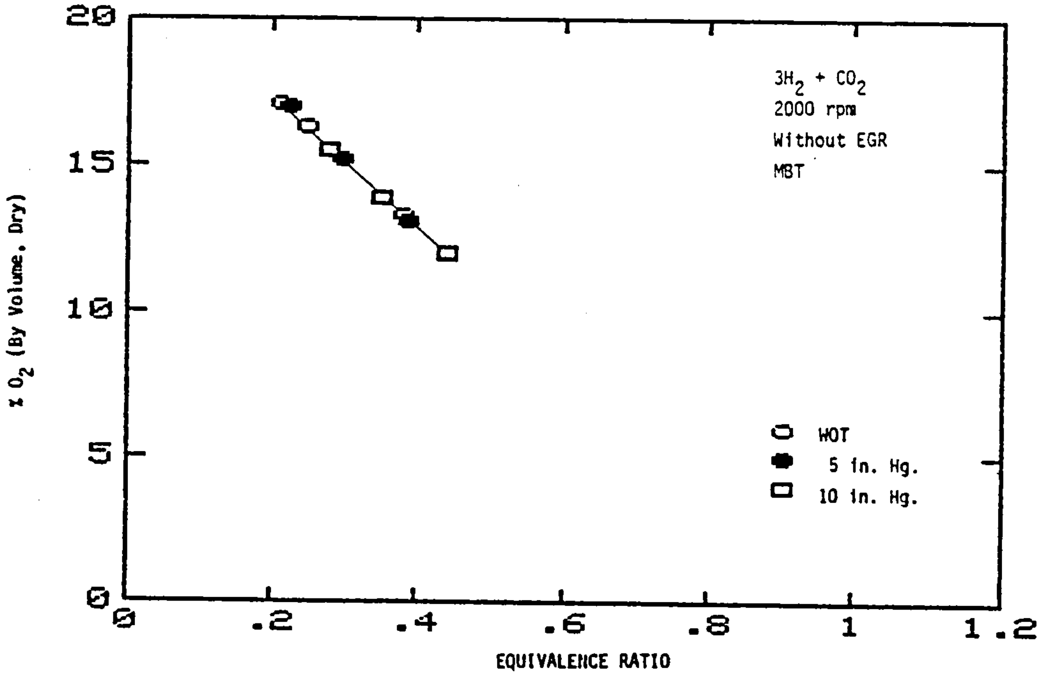

Figure B17: Effect of Equivalence Ratio on Oxygen Emissions at Various Throttle Settings for Steam Reformed Methanol at 2000 rpm, without EGR.

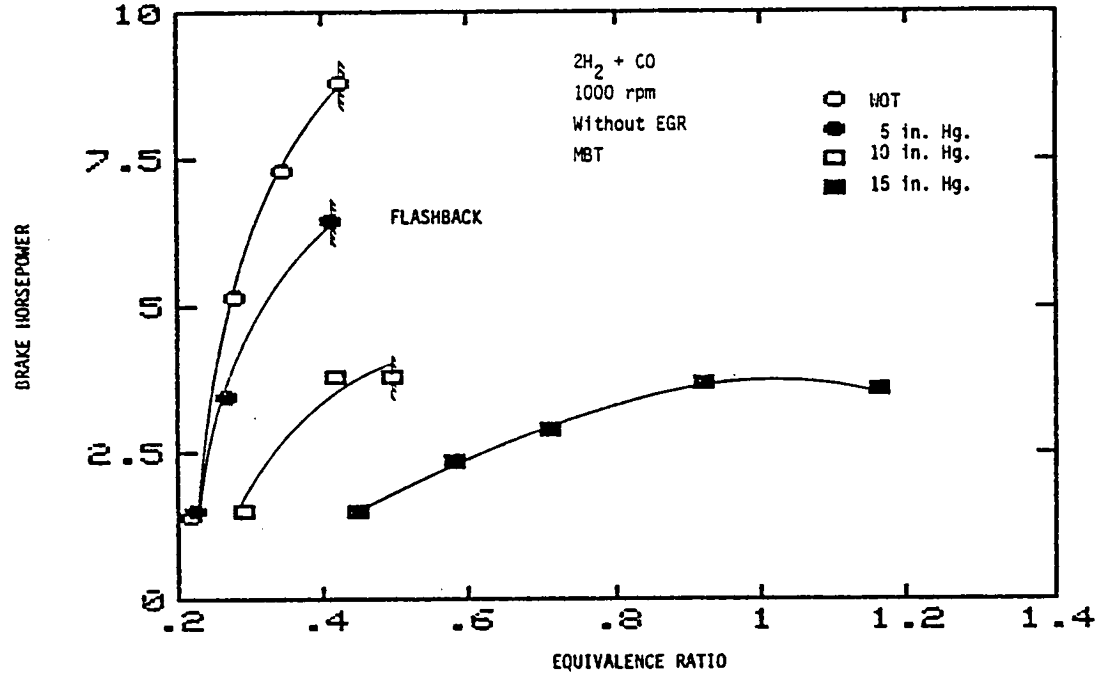

Figure B18: Effect of Equivalence Ratio on Brake Horsepower at Various Throttle Settings for Dissociated Methanol at 1000 rpm, without EGR.

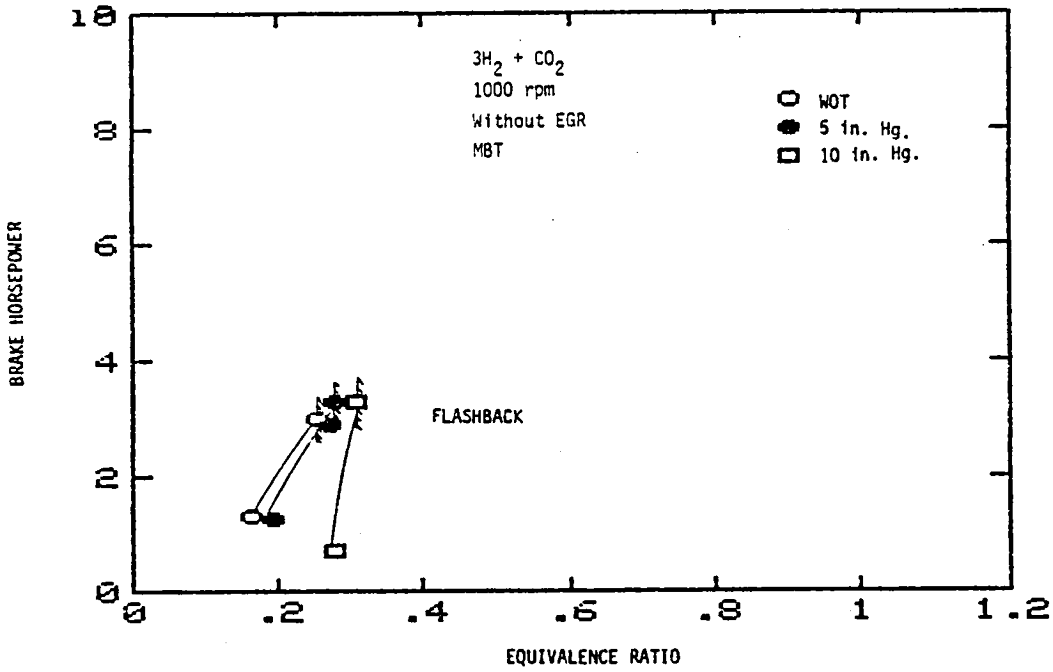

Figure B19: Effect of Equivalence Ratio on Brake Horsepower at Various Throttle Settings for Steam Reformed Methanol at 1000 rpm, without EGR.

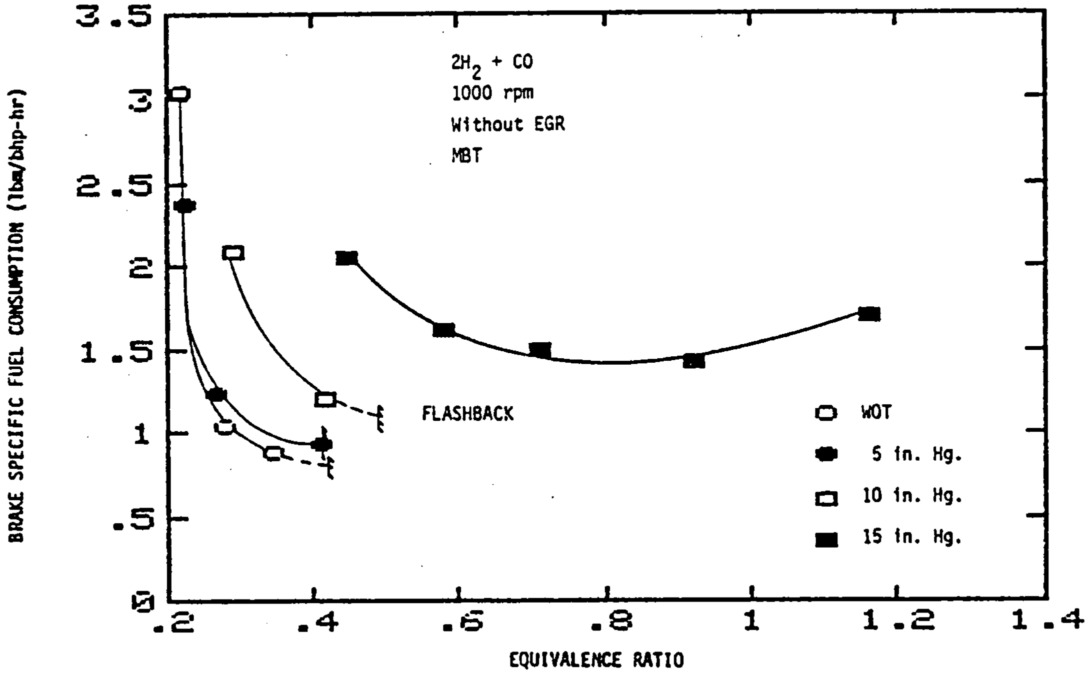

Figure B20: Effect of Equivalence Ratio on Brake Specific Fuel Consumption at Various Throttle Settings for Dissociated Methanol at 1000 rpm, without EGR.

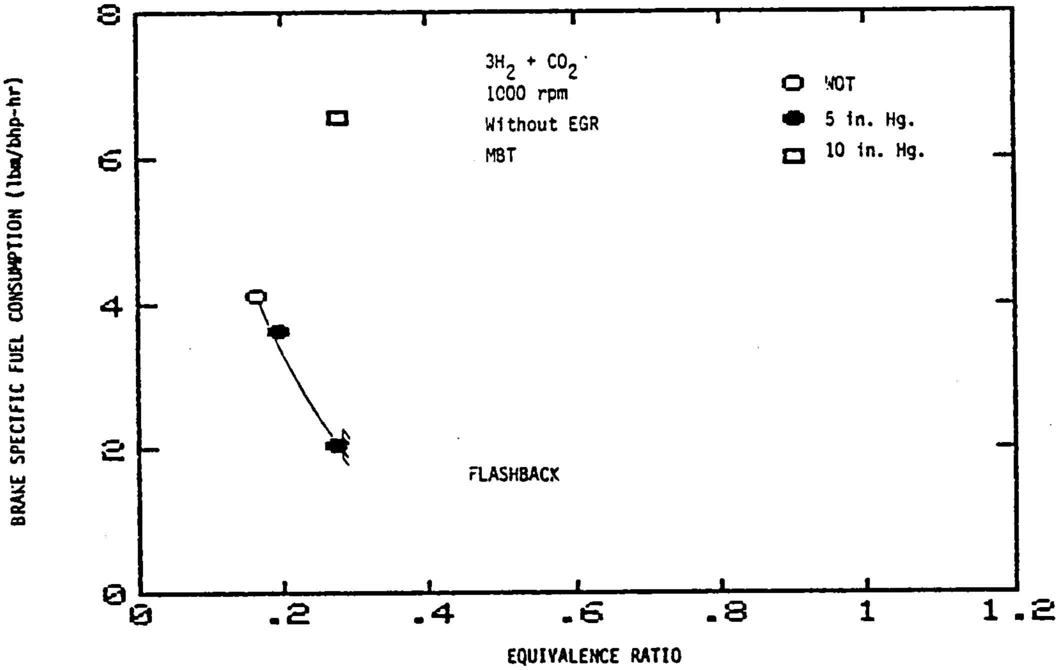

Figure B21: Effect of Equivalence Ratio on Brake Specific Fuel Consumption at Various Throttle Settings for Steam Reformed Methano at 1000 rpm, without EGR.

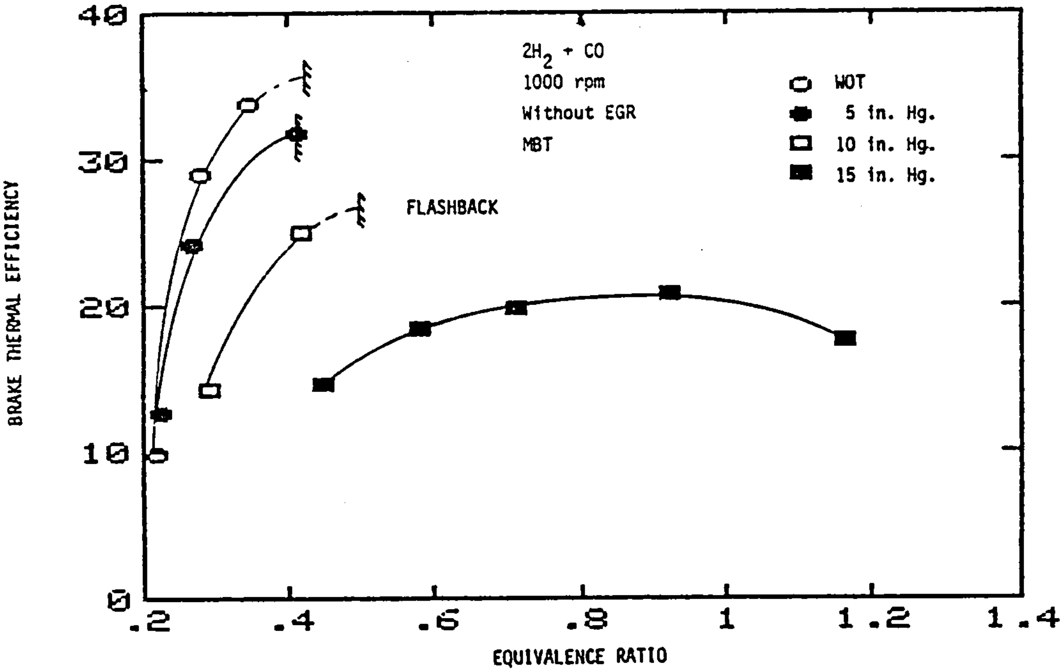

Figure B22: Effect of Equivalence Ratio on Brake Thermal Efficiency at Various Throttle Settings for Dissociated Methanol at 1000 rpm, without EGR (Efficiency Based upon the LHV of Liquid Methanol).

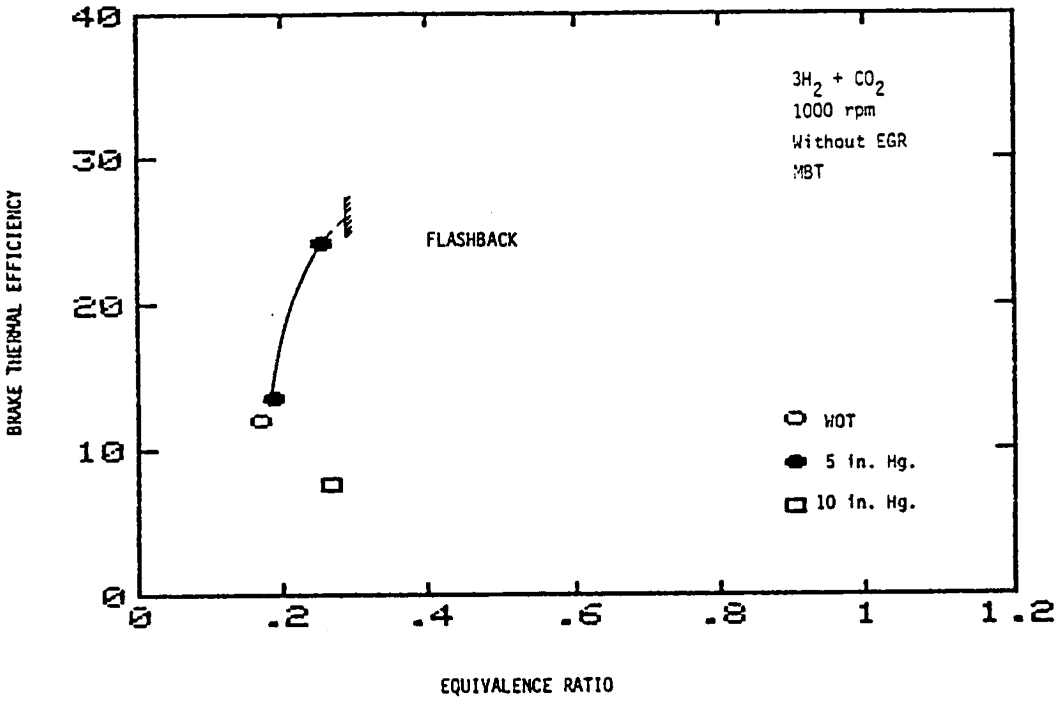

Figure B23: Effect of Equivalence Ratio on Brake Thermal Efficiency at Various Throttle Settings for Steam Reformed Methanol at 1000 rpm, without EGR (Efficiency Based upon the LHV of an Equimolar Methanol-Water Mixture).

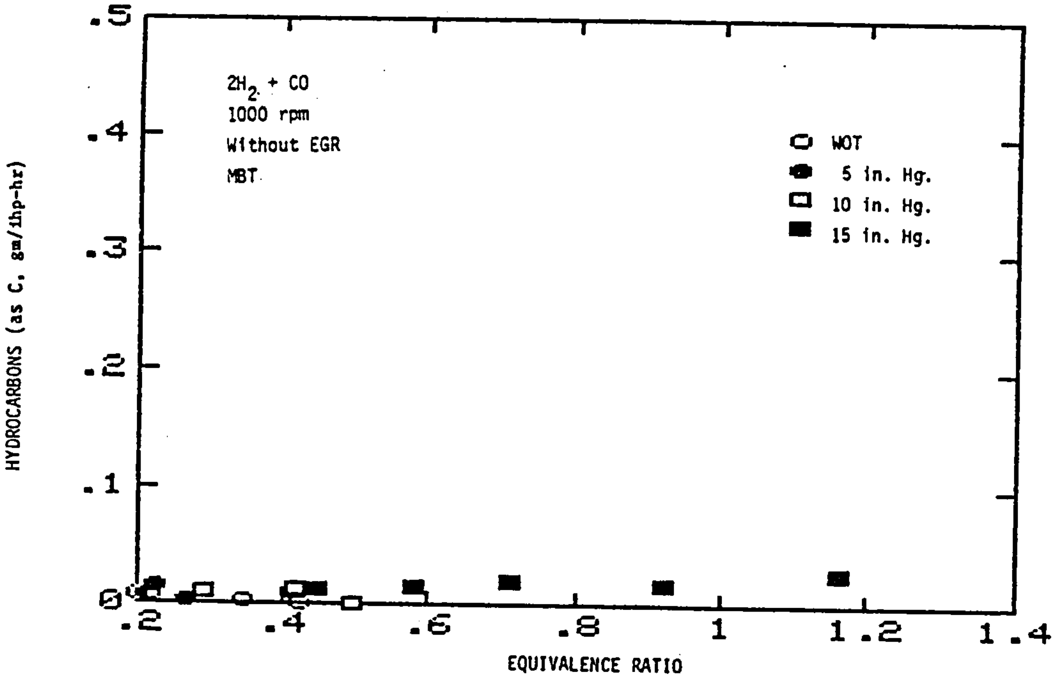

Figure B24: Effect of Equivalence Ratio on Hydrocarbon Emissions at Various Throttle Settings for Dissociated Methanol at 1000 rpm, without EGR.

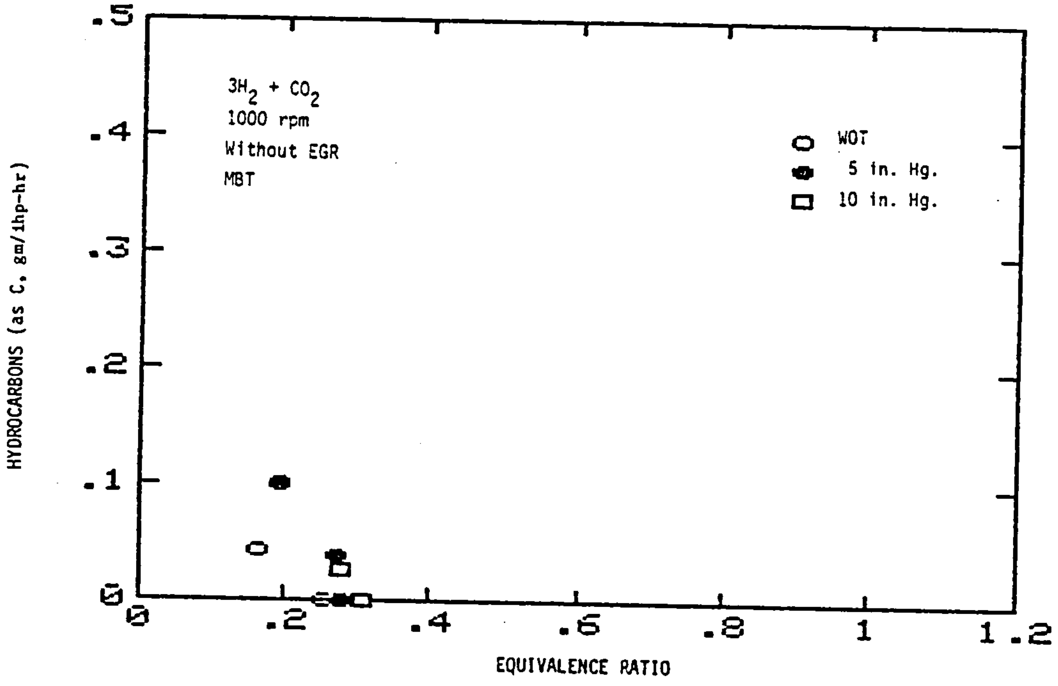

Figure B25: Effect of Equivalence Ratio on Hydrocarbon Emissions at Various Throttle Settings for Steam Reformed Methanol at 1000 rpm, without EGR.

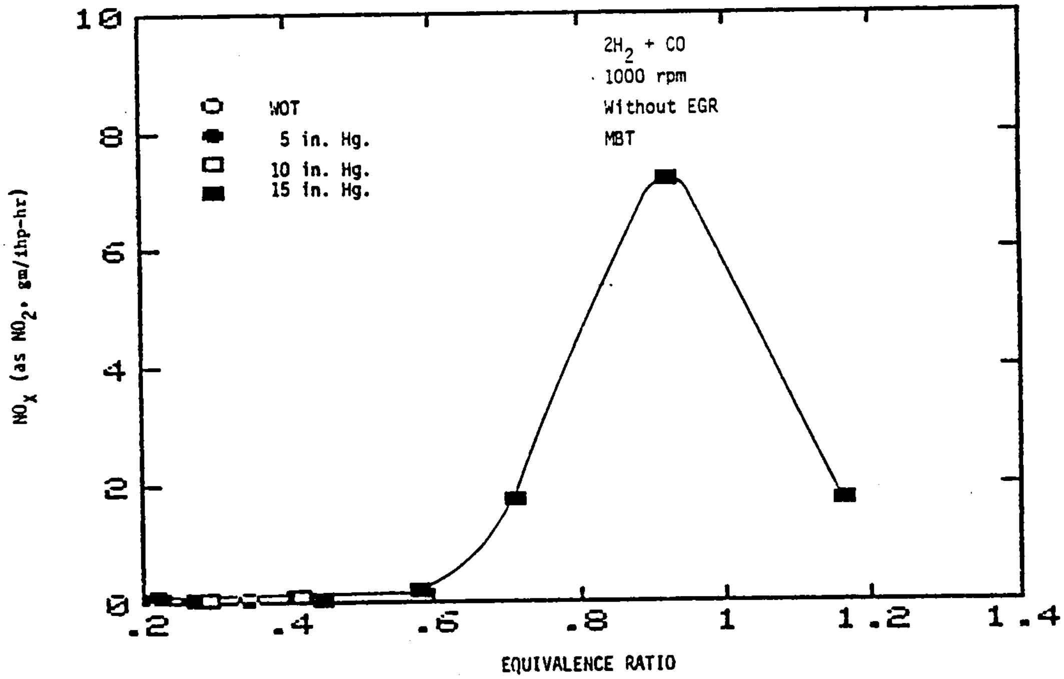

Figure B26: Effect of Equivalence Ratio on Oxides of Nitrogen Emissions at Various Throttle Settings for Dissociated Methanol at 1000 rpm, without EGR.

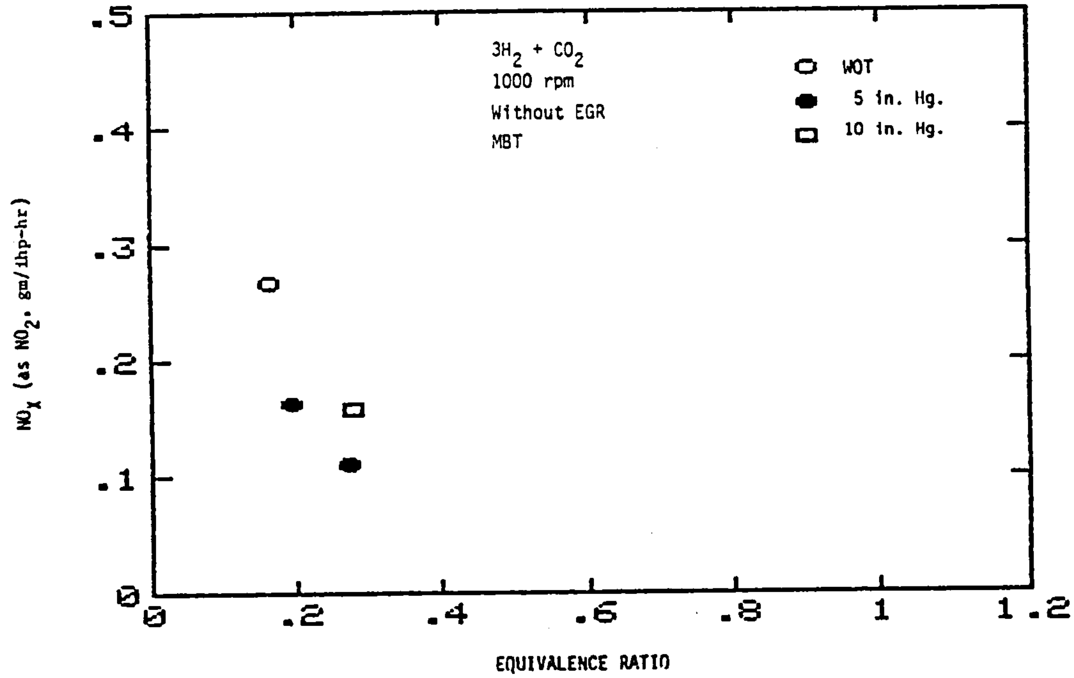

Figure B27: Effect of Equivalence Ratio on Oxides of Nitrogen Emissions at Various Throttle Settings for Steam Reformed Methanol at 1000 rpm, without EGR.

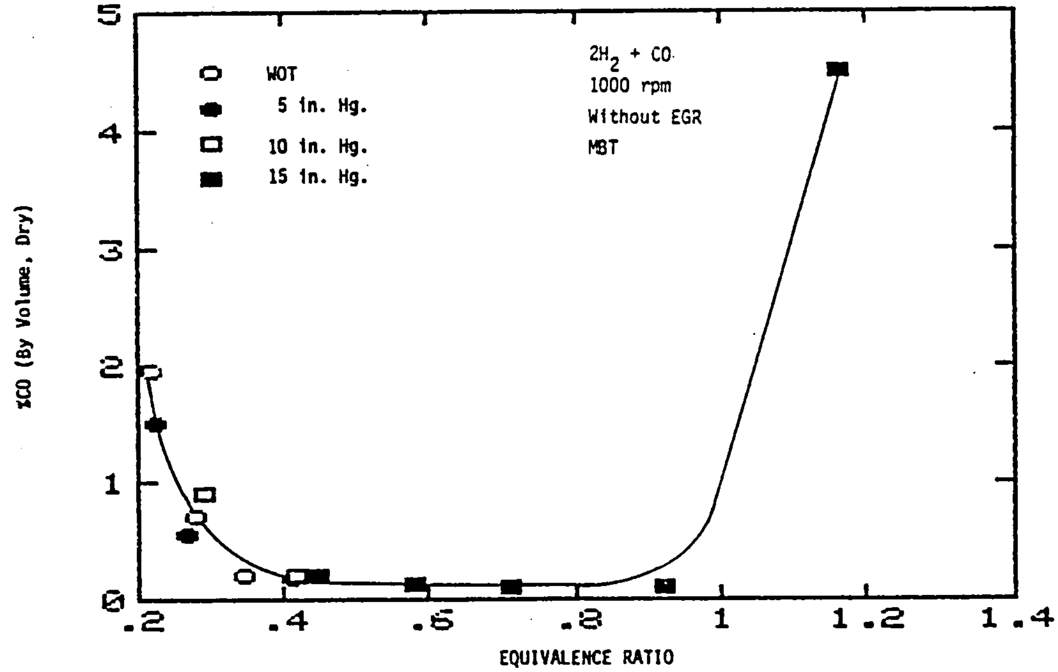

Figure B28: Effect of Equivalence Ratio on Carbon Monoxide Emissions at Various Throttle Settings for Dissociated Methanol at 1000 rpm, without EGR.

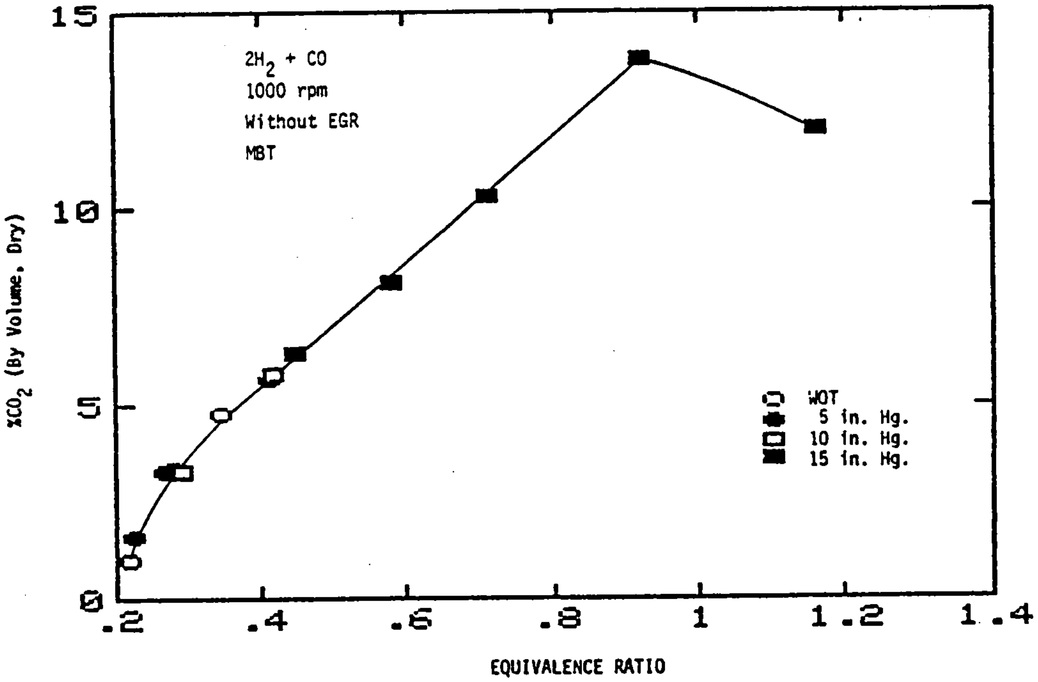

Figure B29: Effect of Equivalence Ratio on Carbon Dioxide Emissions at Various Throttle Settings for Dissociated Methanol at 1000 rpm, without EGR.

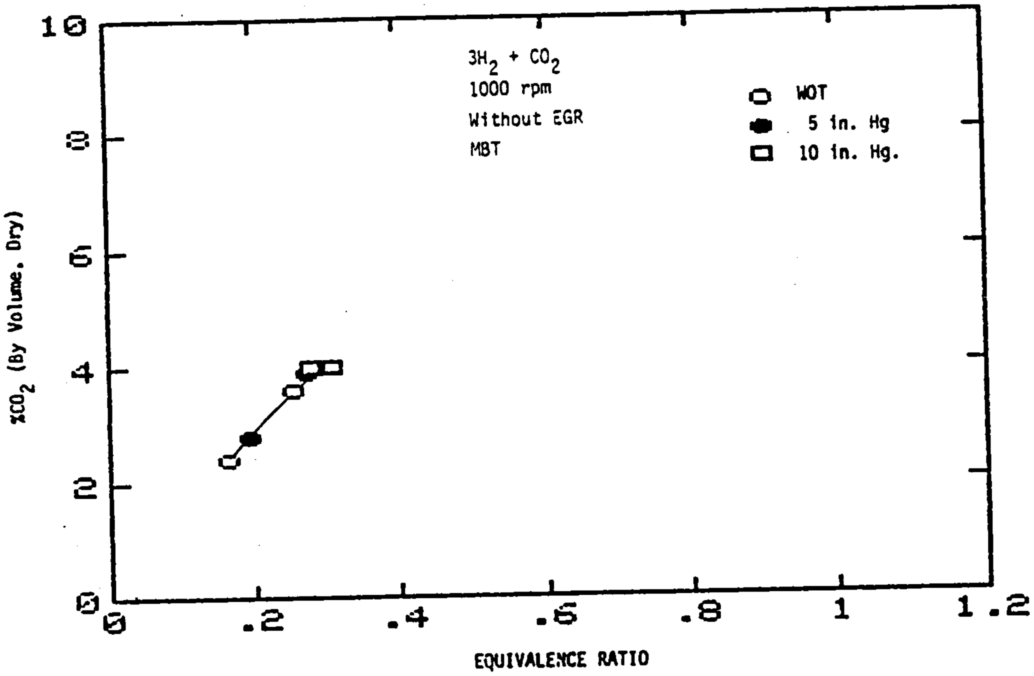

Figure B30: Effect of Equivalence Ratio on Carbon Dioxide Emissions at Various Throttle Settings for Steam Reformed Methanol at 1000 rpm, without EGR.

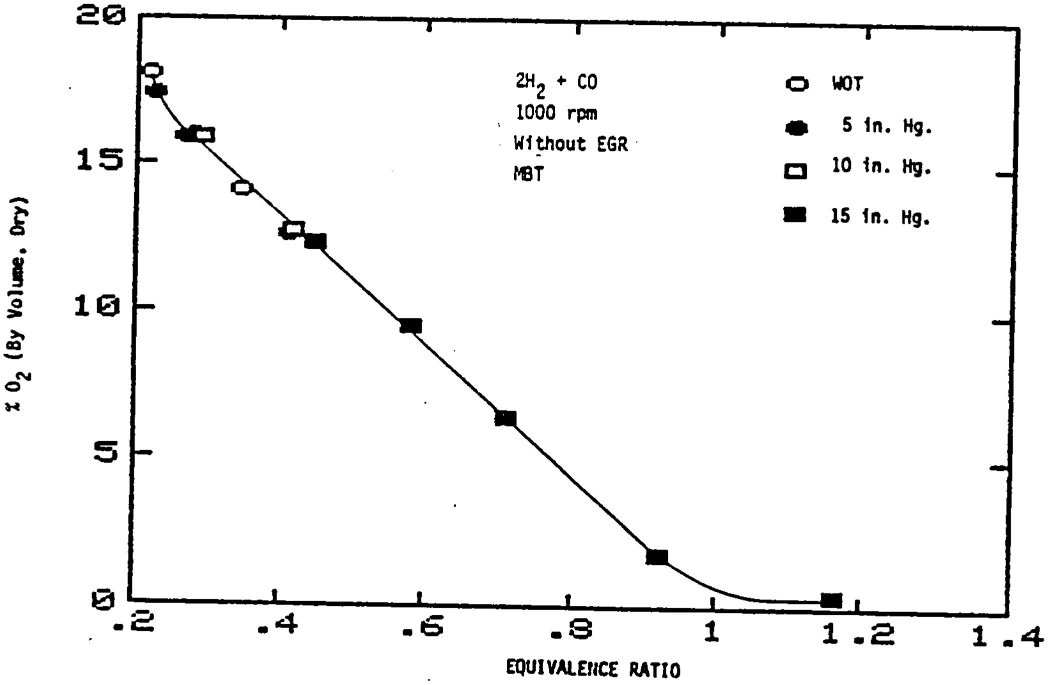

Figure B31: Effect of Equivalence Ratio on Oxygen Emissions at Various Throttle Settings for Dissociated Methanol at 1000 rpm, without EGR.

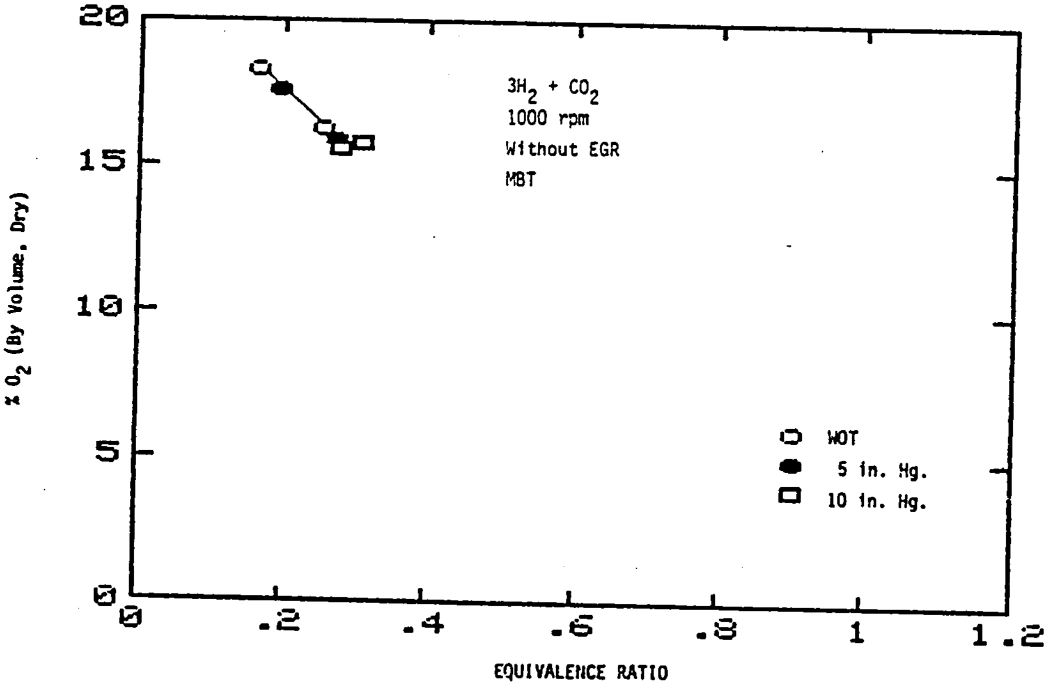

Figure B32: Effect of Equivalence Ratio on Oxygen Emissions at Various Throttle Settings for Steam Reformed Methanol at 1000 rpm, without EGR.

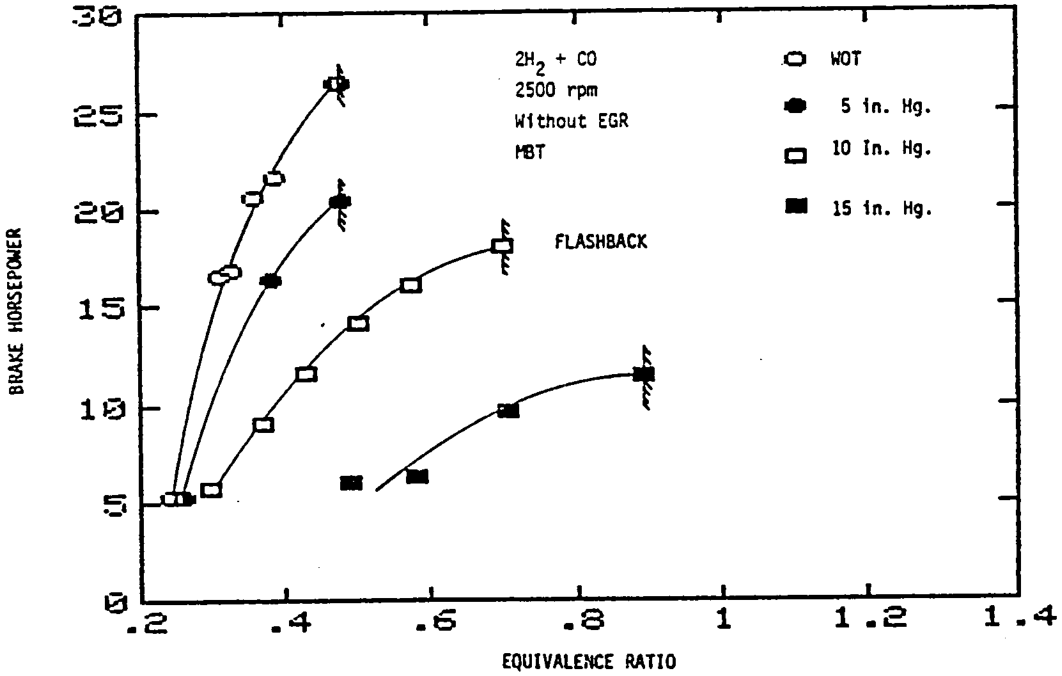

Figure B33: Effect of Equivalence Ratio on Brake Horsepower at Various Throttle Settings for Dissociated Methanol at 2500 rpm, without EGR.

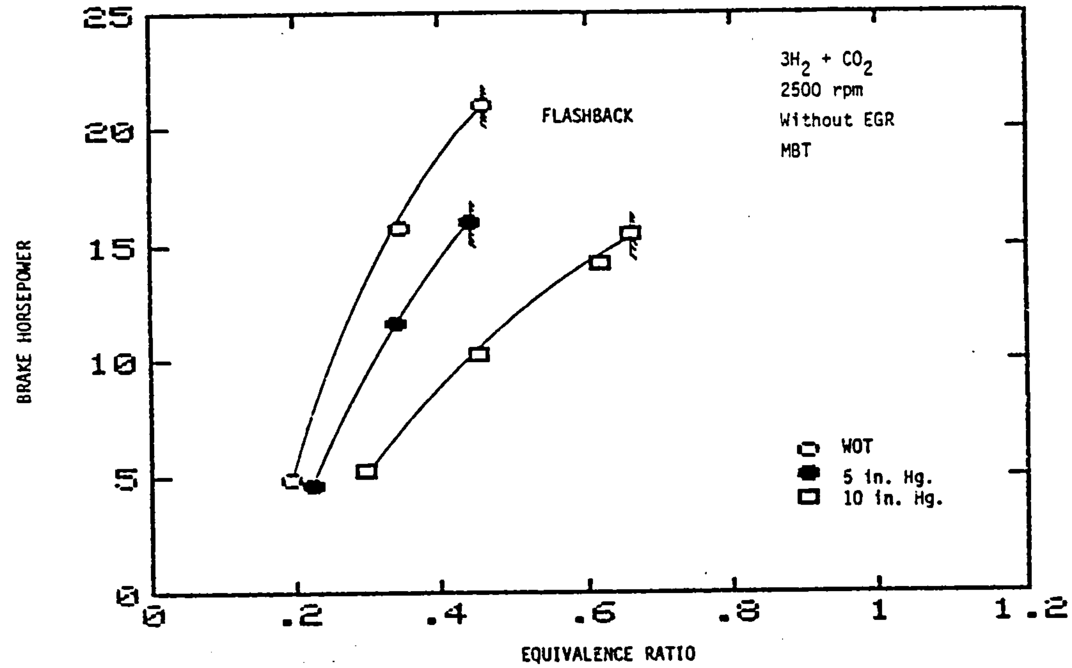

Figure B34: Effect of Equivalence Ratio on Brake Horsepower at Various Throttle Settings for Steam Reformed Methanol at 2500 rpm, without EGR.

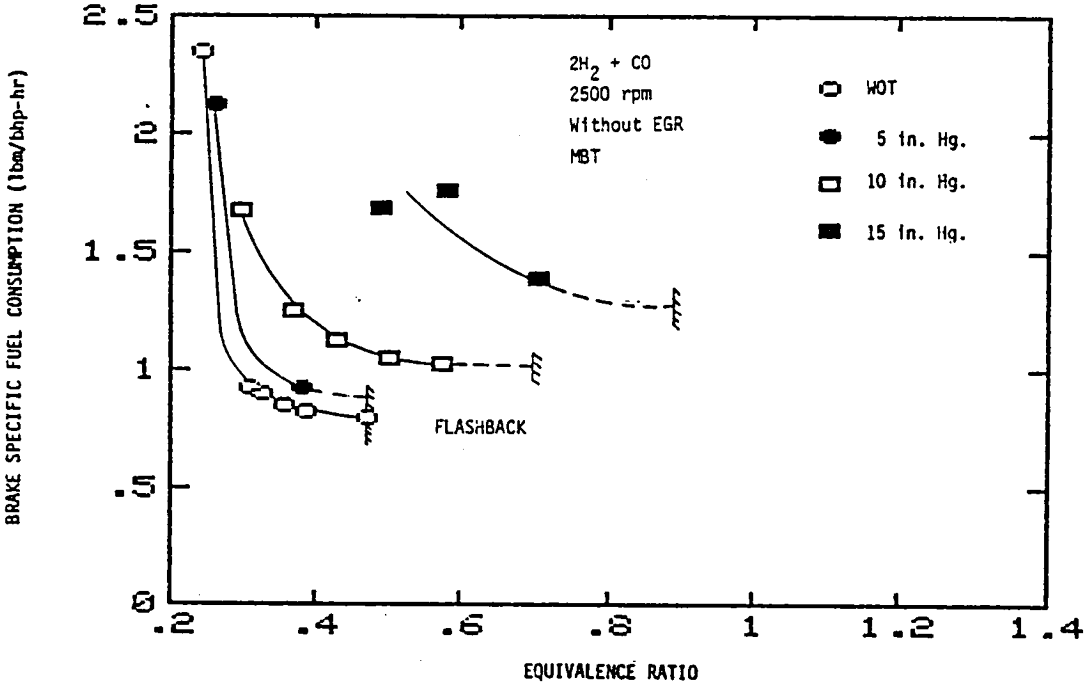

Figure B35: Effect of Equivalence Ratio on Brake Specific Fuel Consumption at Various Throttle Settings for Dissociated Methanol at 2500 rpm, without EGR.

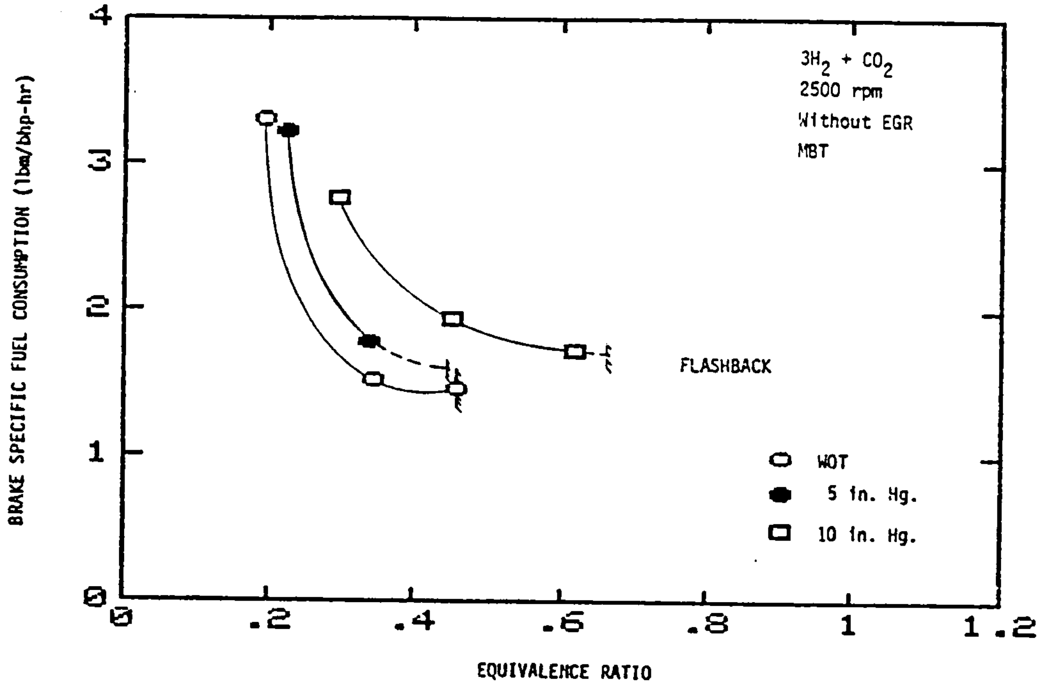

Figure B36: Effect of Equivalence Ratio on Brake Specific Fuel Consumption at Various Throttle Settings for Steam Reformed Methanol at 2500 rpm, without EGR.

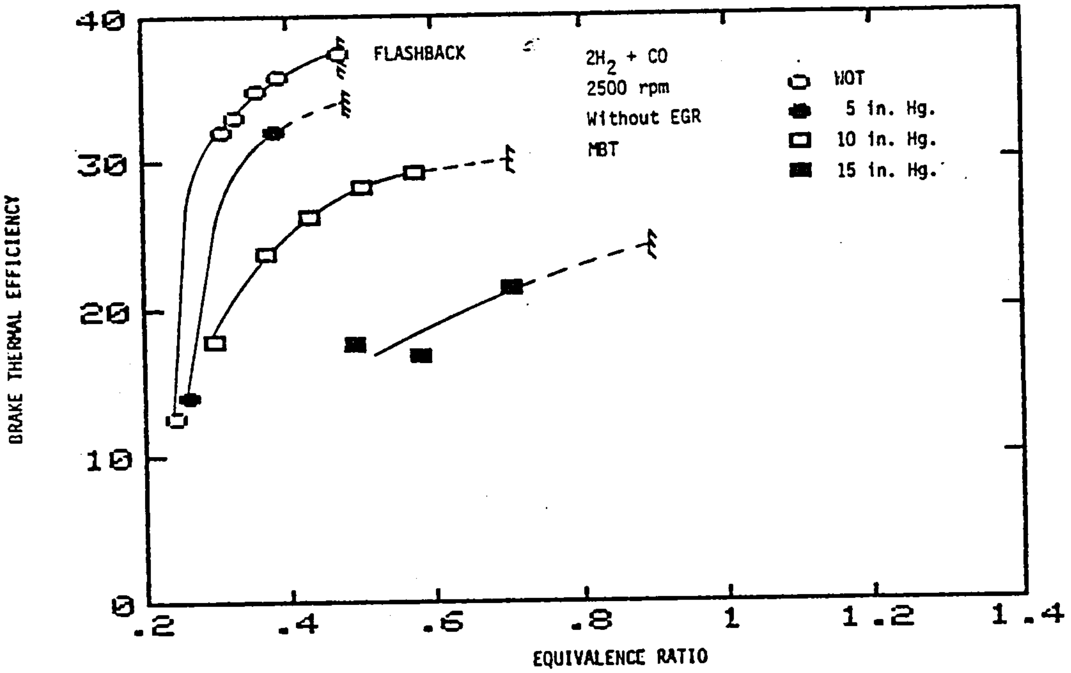

Figure B37: Effect of Equivalence Ratio on Brake Thermal Efficiency at Various Throttle Settings for Dissociated Methanol at 2500 rpm, without EGR (Efficiency Based upon the LHV of Liquid Methanol)

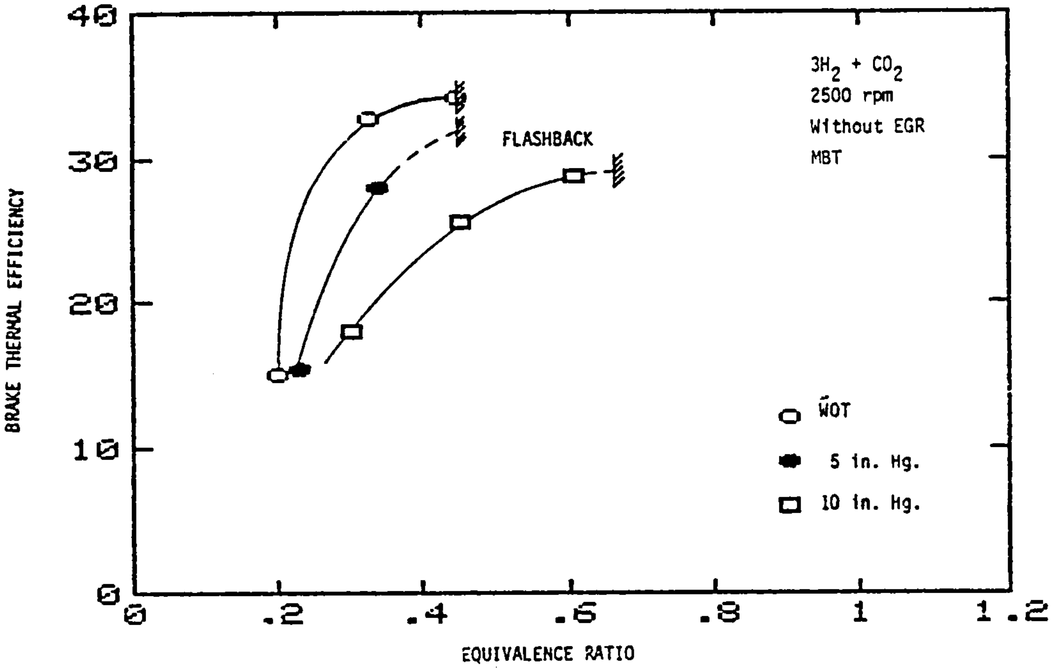

Figure B38: Effect of Equivalence Ratio on Brake Thermal Efficiency at Various Throttle Settings for Steam Reformed Methanol at 2500 rpm, without EGR (Efficiency Based upon the LHV of an Equimolar Methanol-Water Mixture).

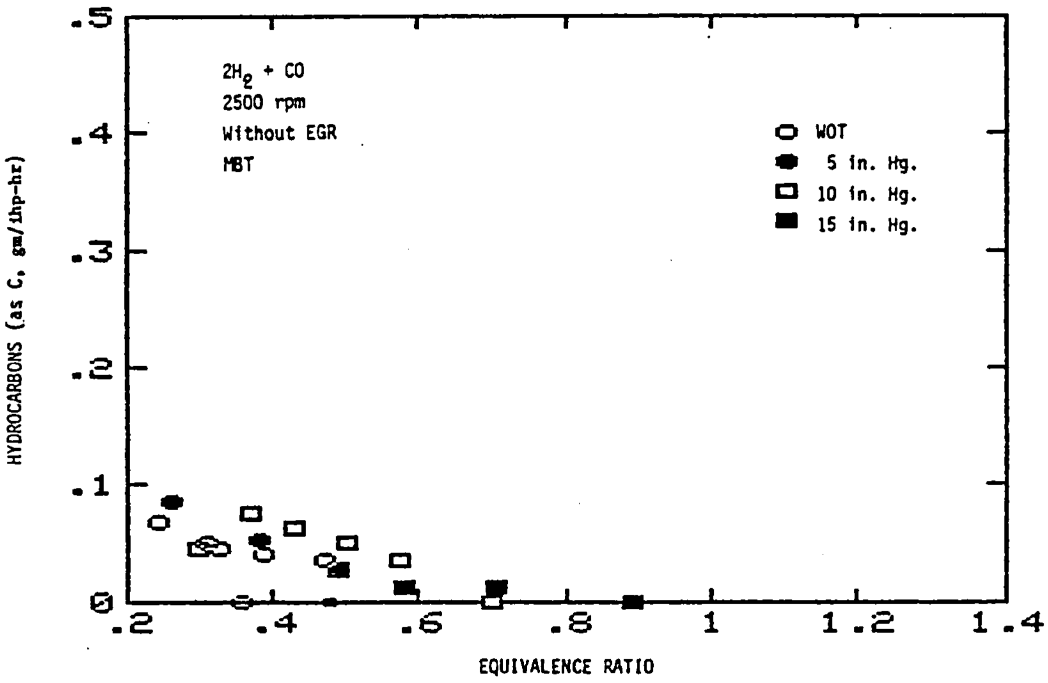

Figure B39: Effect of Equivalence Ratio on Hydrocarbon Emissions at Various Throttle Settings for Dissociated Methanol at 2500 rpm, without EGR.

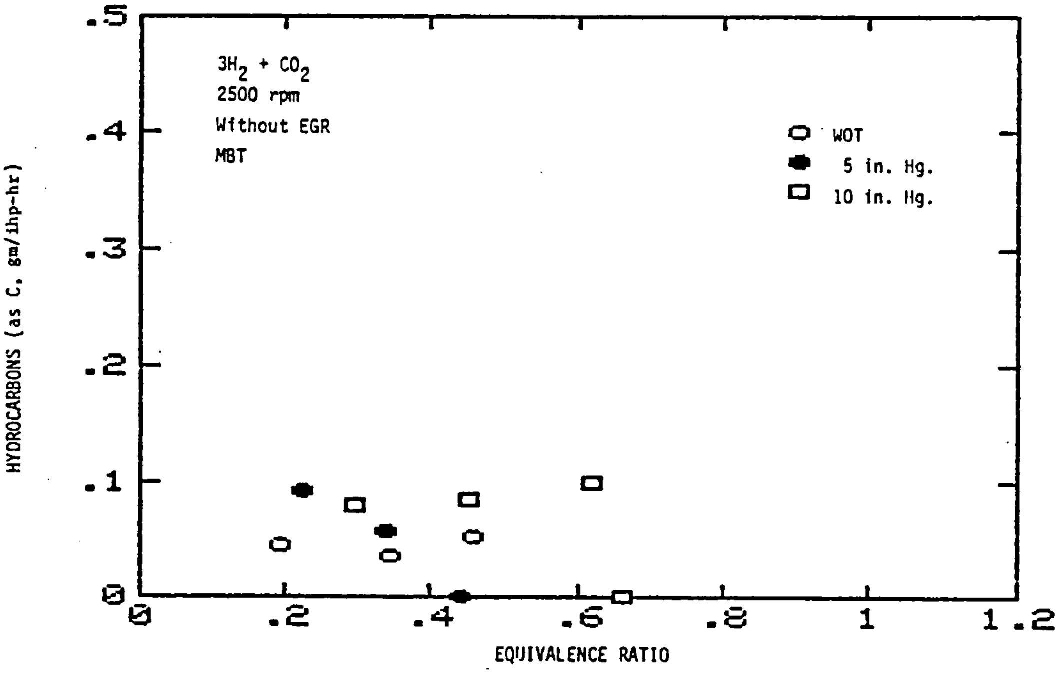

Figure B40: Effect of Equivalence Ratio on Hydrocarbon Emissions at Various Throttle Settings for Steam Reformed Methanol at 2500 rpm, without EGR.

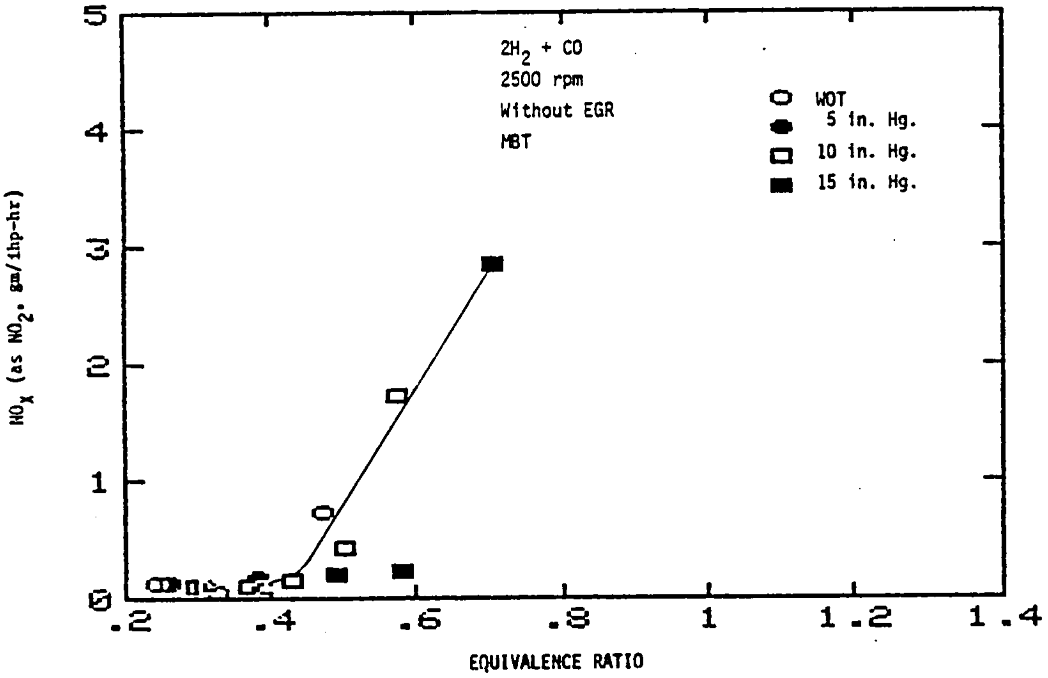

Figure B41: Effect of Equivalence Ratio on Oxides of Nitrogen Emissions at Various Throttle Settings for Dissociated Methanol at 2500 rpm, without EGR.

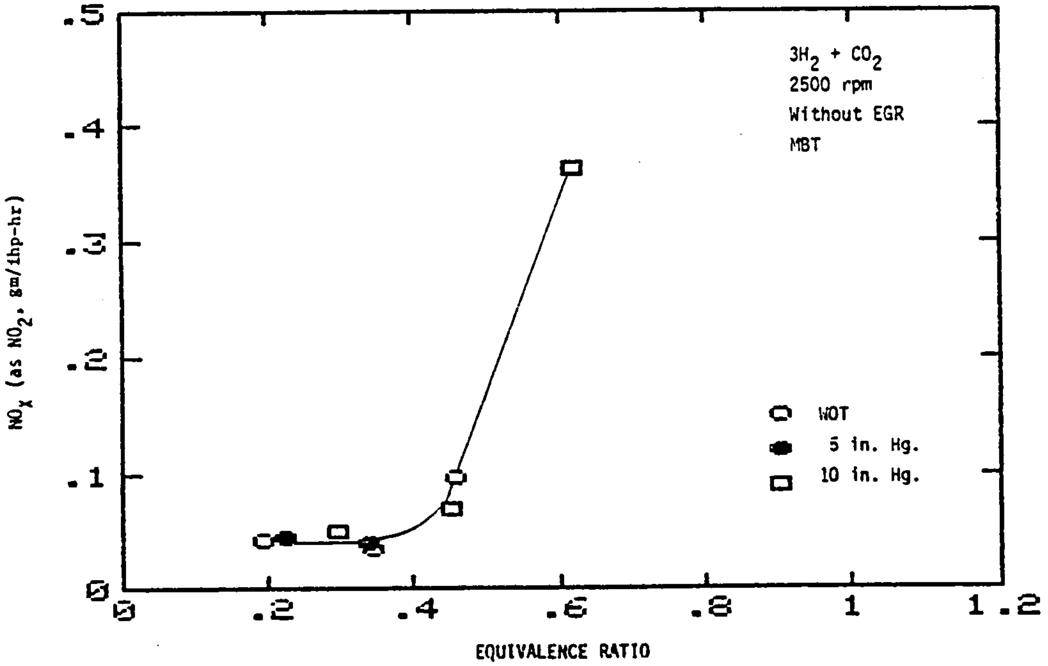

Figure B42: Effect of Equivalence Ratio on Oxides of Nitrogen Emissions at Various Throttle Settings for Steam Reformed Methanol at 2500 rpm, without EGR.

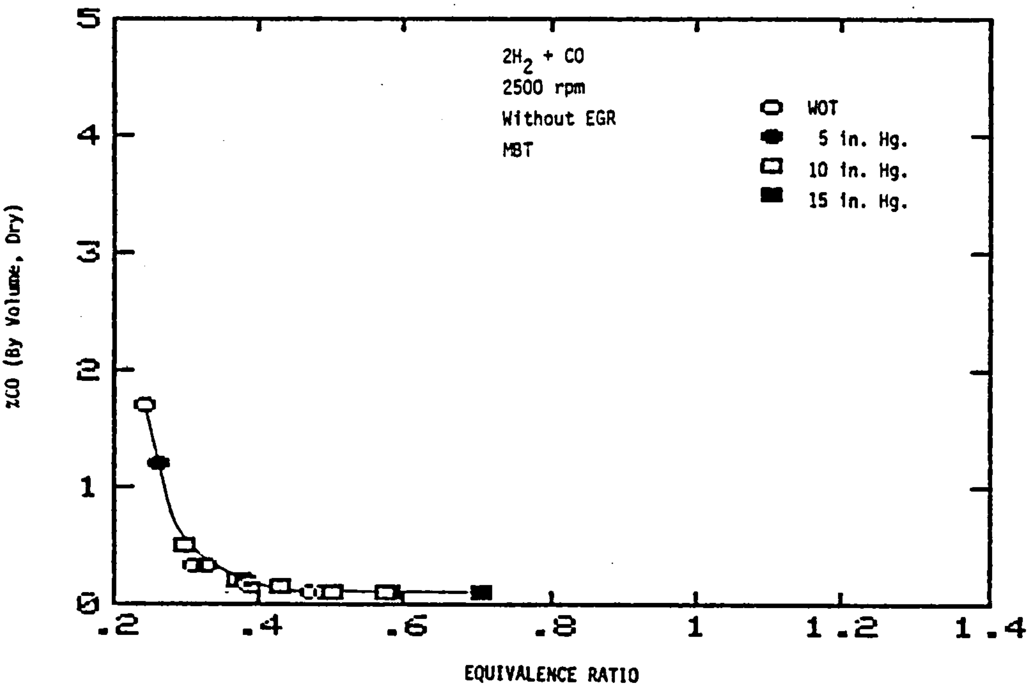

Figure B43: Effect of Equivalence Ratio on Carbon Monoxide Emissions at Various Throttle Settings for Dissociated Methanol at 2500 rpm, without EGR.

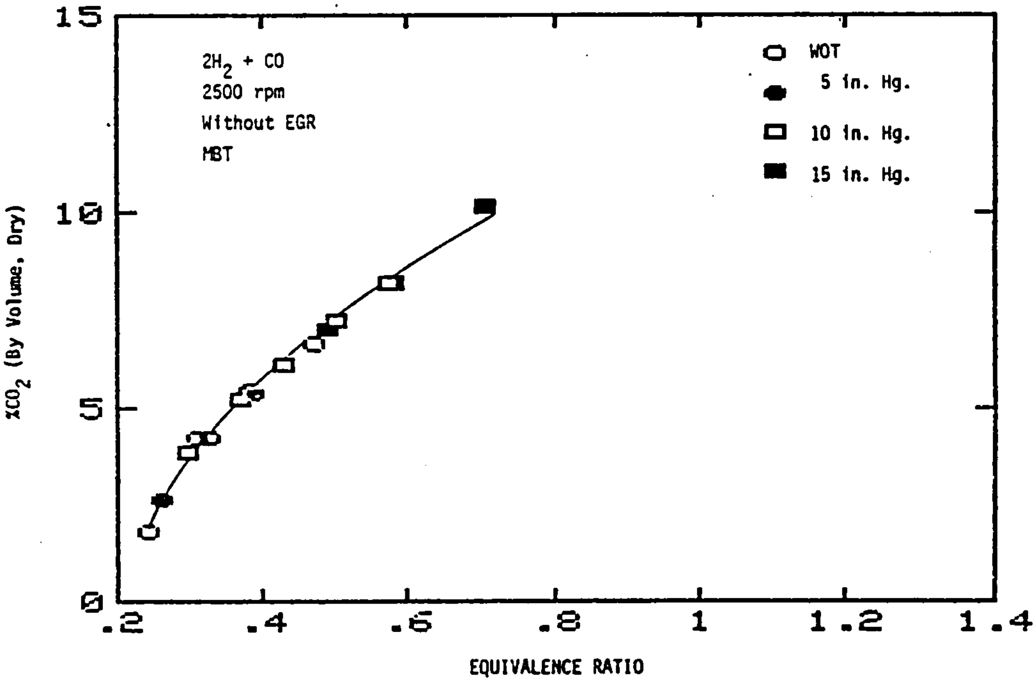

Figure B44: Effect of Equivalence Ratio on Carbon Dioxide Emissions at Various Throttle Settings for Dissociated Methanol at 2500 rpm, without EGR.

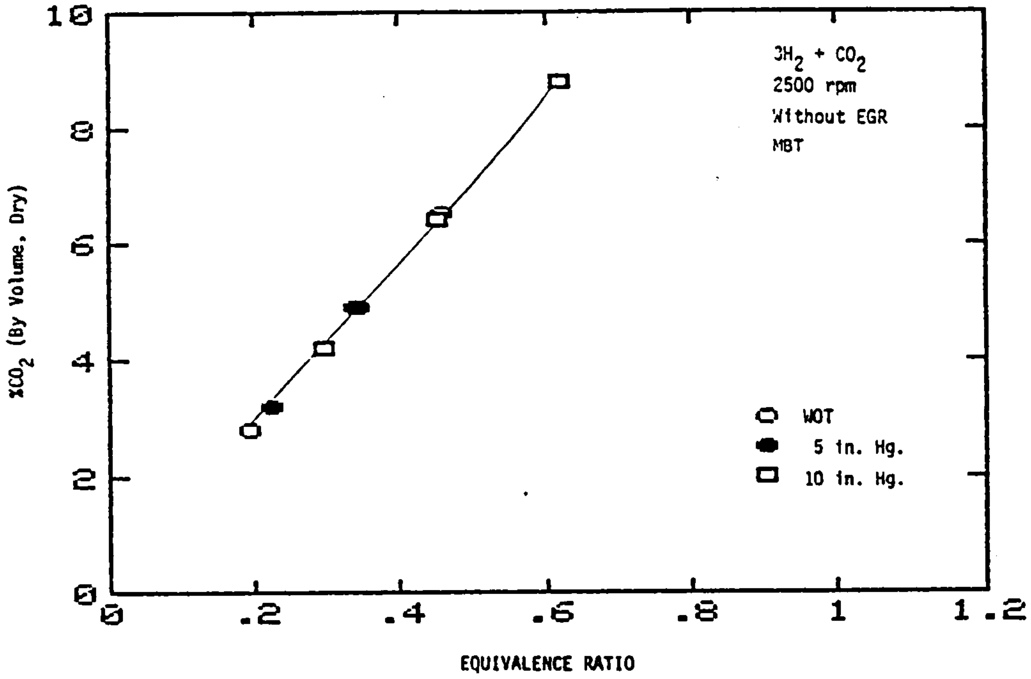

Figure B45: Effect of Equivalence Ratio on Carbon Dioxide Emissions at Various Throttle Settings for Steam Reformed Methanol at 2500 rpm, without EGR.

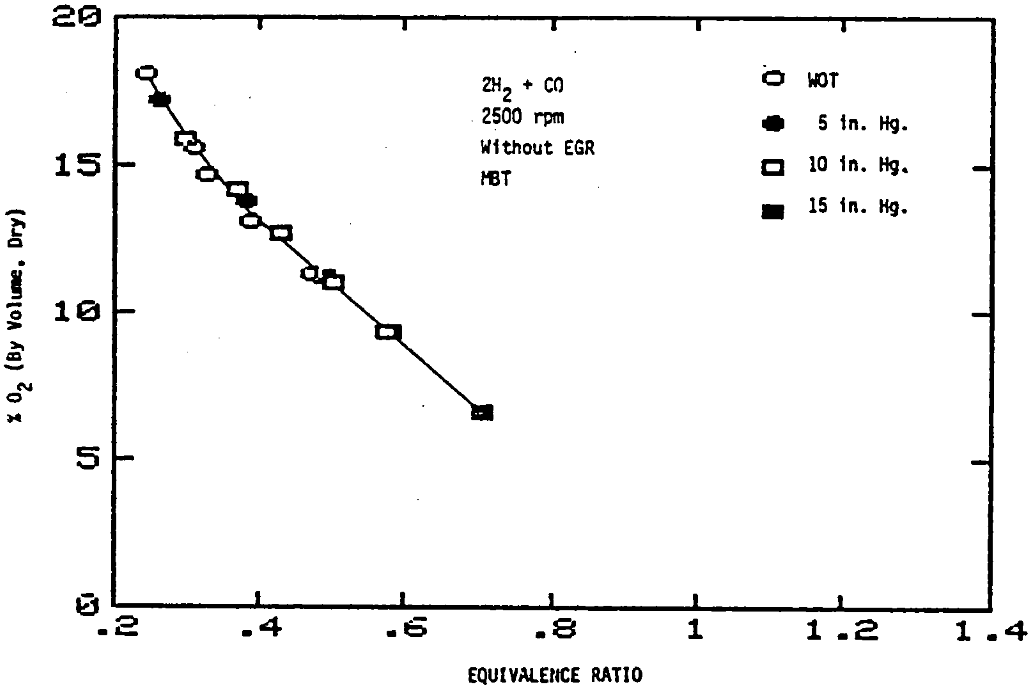

Figure B46: Effect of Equivalence Ratio on Oxygen Emissions at Various Throttle Settings for Dissociated Methanol at 2500 rpm, without EGR.

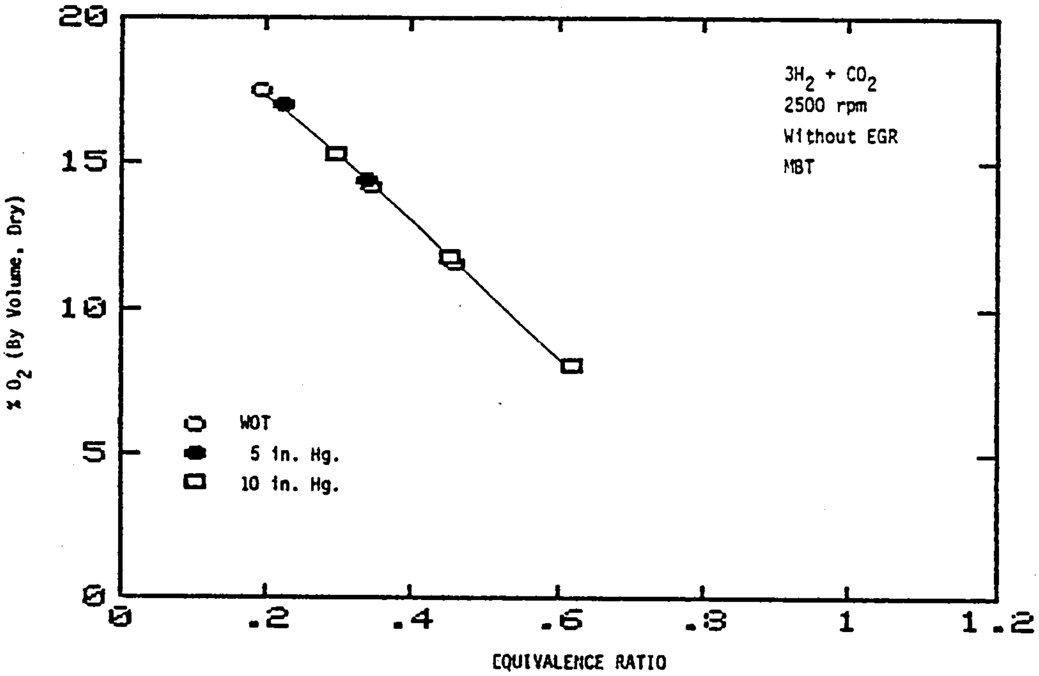

Figure B47: Effect of Equivalence Ratio on Oxygen Emissions at Various Throttle Settings for Steam Reformed Methanol at 2500 rpm, without EGR.

(in maximum power output obtained and equivalence ratio at which flashback occurred) between dissociated and steam reformed methanol was less at 2500 rpm, but higher maximum powers still resulted from operation on dissociated methanol.

A check was done to determine how much of the current EPA driving cycle could be completed using reformed methanol fuel. Based upon the maximum power output at 2500 rpm, enough power would be available using reformed methanol fuel to propel a midsize car through 98 to 99% of the driving cycle. There was an acceleration at the beginning of the cycle that could not be accomodated with either fuel, and an acceleration later in the cycle that could not be met with steam reformed methanol.

B.2 Liquid Methanol and Indolene Baseline Results

The baseline performance and emissions results from operation on liquid methanol and indolene are presented in this section. The baseline fuels and conditions tested are listed in Table B3. Of these tests, only the methanol and indolene baseline results from operation at 2000 rpm and without EGR are included. The remainder of the baseline results are presented in reference 27 along with a comparison of the baseline results from this study to those obtained by others. The baseline results were, in general, in good agreement (within 5%) with those of others showing similar trends and magnitudes. However, the magnitude of the efficiency was slightly higher for both fuels than that reported by others for a similar engine. The performance results are presented first, followed by the emissions results.

As indicated in Table B3, three separate methanol baseline tests at 2000 rpm without EGR were conducted with almost nine months elapsing between the first and third test. The third test was conducted after the engine was

Table B3: Baseline Fuels and Conditions Tested

Fuel	Speed	EGR	$T_{ambient}$	$P_{ambient}$	T_{intake}	$T_{coolant}$	Remarks
Methanol	2000	Yes	68°F	29.78"Hg	32-55°F	195-200°F	First baseline after breaking engine in.
Methanol	2000	No	77°F	29.65"Hg	33-50°F	195-200°F	Resumed testing after repairing broken shaft in driveline.
Indolene	2000	Yes	72°F	29.57"Hg	50-135°F	195-199°F	
Indolene	2000	No	64-61°F	29.86-29.98"Hg	44-64°F	198°F	
Methanol	2000	No	75°F	30.09"Hg	38-43°F	198°F	Checked power and efficiency measurements.
Methanol	2000	No	78°F	28.89"Hg	28-40°F	190-200°F	Cleaned engine.
Methanol	1000	No	77°F	29.75"Hg	38-43°F	185-200°F	

cleaned and cylinder deposits consisting of light rust residue on the cylinder head and piston were removed. Removing cylinder deposits resulted in significant effect on engine performance with methanol. Agreement between the three sets of data was within the calculated experimental uncertainties, indicating good repeatability. Plots of the methanol results at 2000 rpm wtihout EGR presented in this section include the data from all three tests.

Figures B48 and B49 are plots of brake horsepower (Bhp) as a function of the equivalence ratio for methanol and indolene. Comparing the methanol and indolene results, the power was about the same for both fuels over most of the equivalence ratios and throttle settings tested. Power output with methanol was about 2% higher than indolene at $\phi = 1.1$ and WOT, and the highest power measured at 2000 rpm was the same for both fuels (47 Bhp), but occurred at $\phi = 1.2$ for indolene compared to $\phi = 1.1$ for methanol. At the lean equivalence ratio ($\phi \leq 0.8$), the power output obtained with methanol was 10% higher than indolene as the power dropped off more sharply with indolene in this region.

Figures B50 and B51 are plots of brake thermal efficiency (Bte) versus for the two fuels. As expected, efficiencies with methanol ranged from 10% to 20% higher than indolene at the lower equivalence ratios ($\phi \leq 1.0$). The thermal efficiencies of the two fuels tended to differ most at leaner equivalence ratios and lower throttle settings, as did the power. Efficiencies for methanol peaked around $\phi = 0.8$ at WOT and peaked at increasingly higher equivalence ratios as the manifold vacuum increased (throttle closed). Maximum efficiency obtained with indolene was 32% at $\phi = 0.9$ and WOT compared to 37% at $\phi = 0.8$ and WOT for methanol.

As expected, leaner operation was obtained with the methanol. The lowest effective lean limit occurred near $\phi = 0.7$ at wide open and 3/4 throttle for methanol. The lowest effective lean limit for indolene occurred at $\phi = 0.75$ at

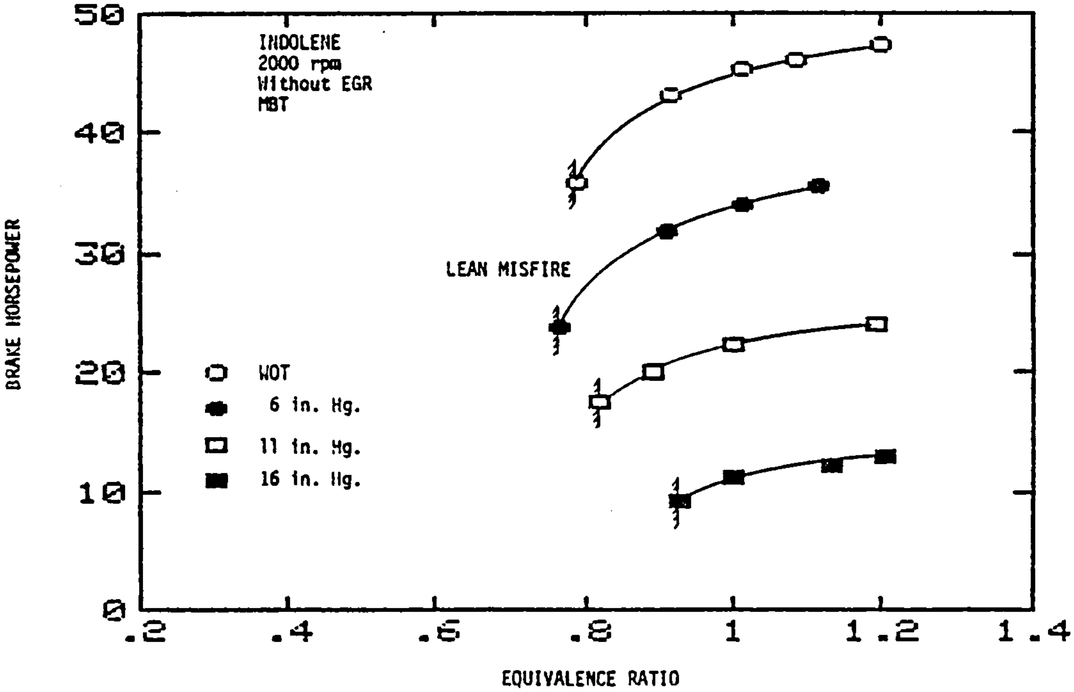

Figure B48: Effect of Equivalence Ratio on Brake Horsepower at Various Throttle Settings for Indolene at 2000 rpm, without EGR.

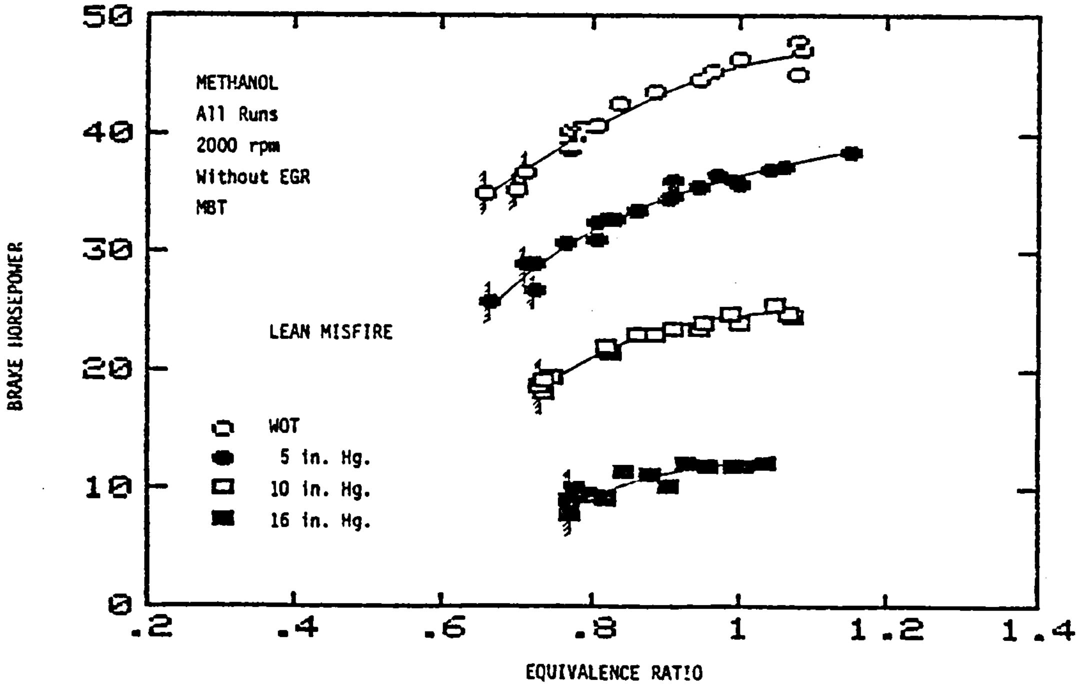

Figure B49: Effect of Equivalence Ratio on Brake Horsepower at Various Throttle Settings for Methanol at 2000 rpm, without EGR.

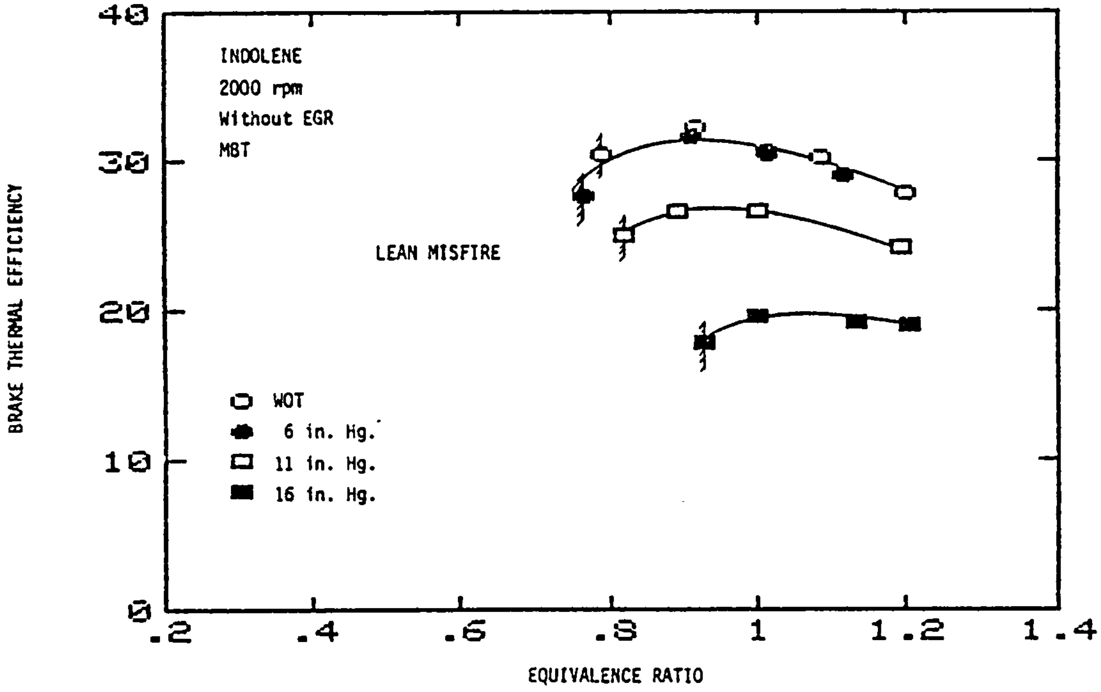

Figure B50: Effect of Equivalence Ratio on Brake Thermal Efficiency at Various Throttle Settings for Indolene at 2000 rpm, without EGR.

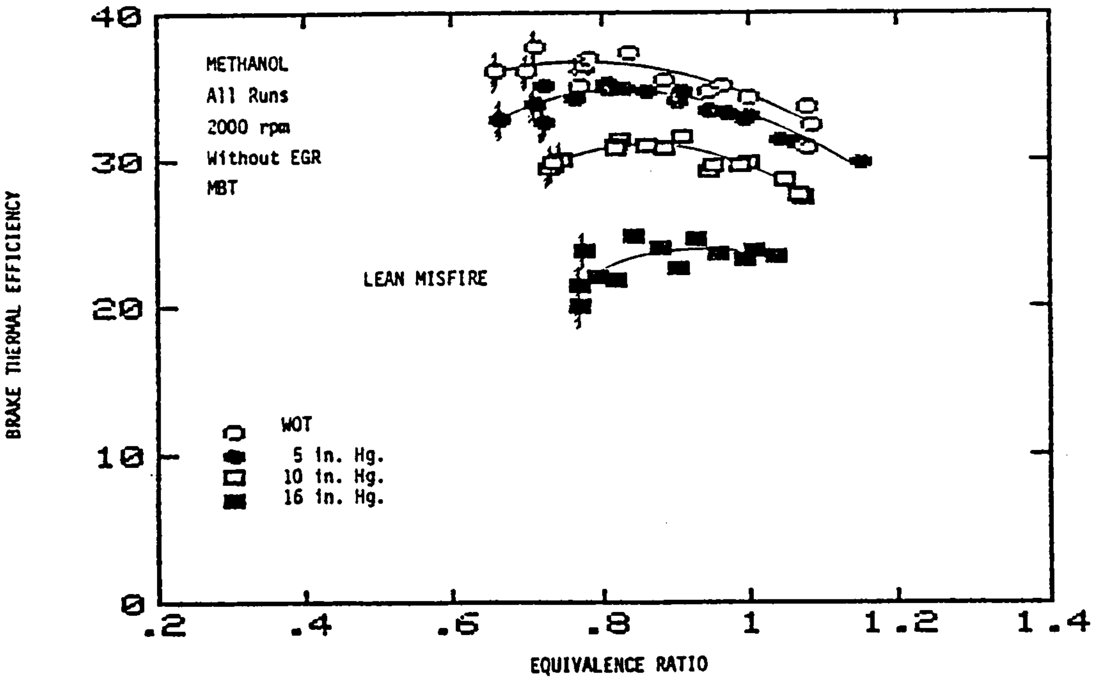

Figure B51: Effect of Equivalence Ratio on Brake Thermal Efficiency at Various Throttle Settings for Methanol at 2000 rpm, without EGR.

3/4 throttle. The term "effective" lean limit refers to the fact that the true lean limit obtained in a single cyclinder is masked by maldistribution in a multicylinder engine. The effective lean limit in a multicylinder engine occurs when the cylinder operating at the leanest air-fuel ratio begins to misfire. Since the air-fuel mixtures in the other cylinders are normally richer than the leanest cylinder, the lean misfire limit for the engine occurs at an overall richer air-fuel ratio than it would for a single cylinder engine.

Maldistribution was checked by measuring the emission levels from the individual cylinder exhaust ports at regular intervals during the initial tests with indolene and methanol. An extensive mapping of maldistribution was not undertaken as this was not of primary concern in this work. However, data obtained for methanol without EGR and indolene with EGR indicated a variation in ϕ among cylinders from 20% to 30% for both fuels. Maldistribution tended to decrease at leaner operation, and throttle setting appeared to have no effect. It should be pointed out that the conditions tested were to represent best case (indolene with EGR) and worst case (methanol without EGR) maldistribution, but despite the higher intake temperatures and the lower heat of vaporization of the indolene, the expected differences in maldistribution between the two fuels were not observed.

Hydrocarbons, NO_x, CO, CO_2, O_2, and aldehydes at each test point are plotted as a function of the equivalence ratio for indolene and methanol in Figures B52 through B63. In general, emissions levels of hydrocarbons and NO_x were lower for methanol than for indolene. The hydrocarbon emissions showed little variation with throttle setting for methanol. Hydrocarbons tended to be twice as high for indolene than for methanol at similar equivalence ratios and throttle settings when they are calculated on the basis of carbon. However, when methanol hydrocarbon emissions are calculated based on the molecular weight of the unburned fuel (CH_3OH), the differences are not so large. Expressed as unburned fuel, hydrocarbon emissions for methanol range from 10% lower than for indolene at WOT to 200% lower at 1/4 throttle. NO_x levels for methanol were lower than those with indolene at the lower throttle settings where the majority of NO_x emissions were below 5 gm/ihp-hr for methanol. This was not the case for indolene.

Aldehydes showed significant scatter due to the inherent uncertainties of the wet chemistry technique used to measure them, but, on an order of magnitude basis, aldehyde emissions were 2 to 4 times higher for methanol than for indolene. The overall magnitudes were quite low, however, (less than 0.5 gm/ihp-hr). CO, CO_2, and O_2 were similar for both fuels. Since CO, CO_2, and O_2 are strong functions of the equivalence ratio, throttle had little effect on the exhaust emissions. Scatter and insufficient indolene data make it impossible to identify differences in these emissions between the two fuels.

Figure B64 shows MBT spark advance as a function of ϕ for the methanol at 2000 rpm and wide open throttle. The line running through the points represents the best fit curve of the data. The wide degree of scatter in the data illustrates the difficulty in determining MBT spark timing which is due in part to two factors. The first is that the torque curve as a function of MBT spark advance is relatively flat at 2000 rpm making MBT difficult to pinpoint. Secondly, the maldistribution of the fuel-air charge among the cylinders causes each cylinder to be operating at a different air-fuel ratio, and for a multicylinder engine, MBT timing appears to be a compromise between the proper MBT settings for the individual cylinders. NO_x emissions have been

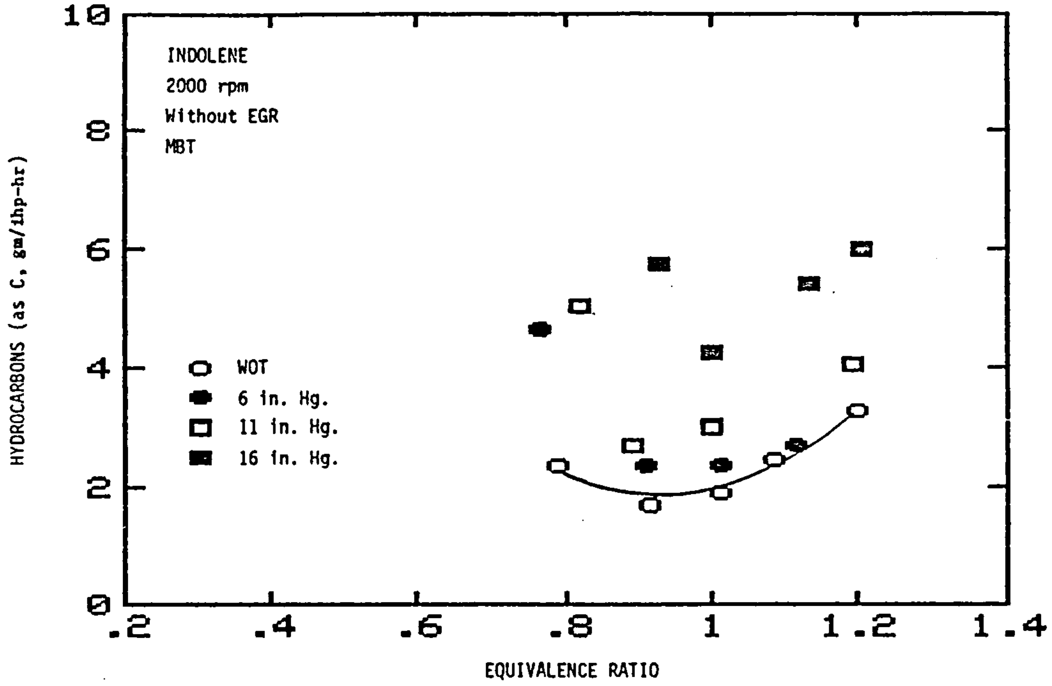

Figure B52: Effect of Equivalence Ratio on Hydrocarbon Emissions at Various Throttle Settings for Indolene at 2000 rpm, without EGR.

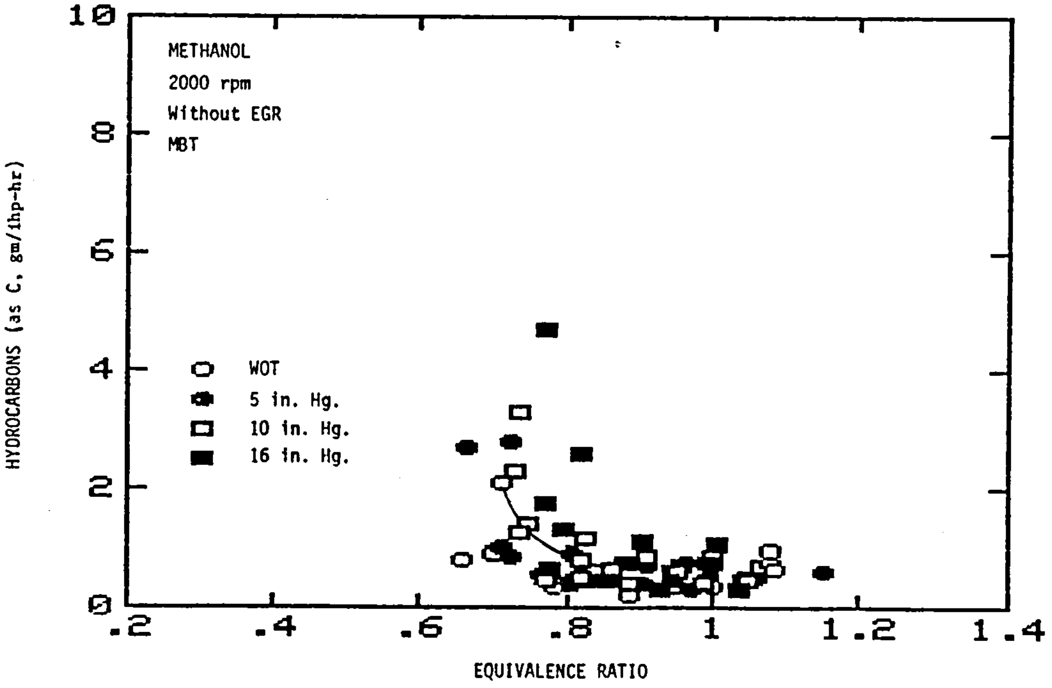

Figure B53: Effect of Equivalence Ratio on Hydrocarbon Emissions at Various Throttle Settings for Methanol at 2000 rpm, without EGR.

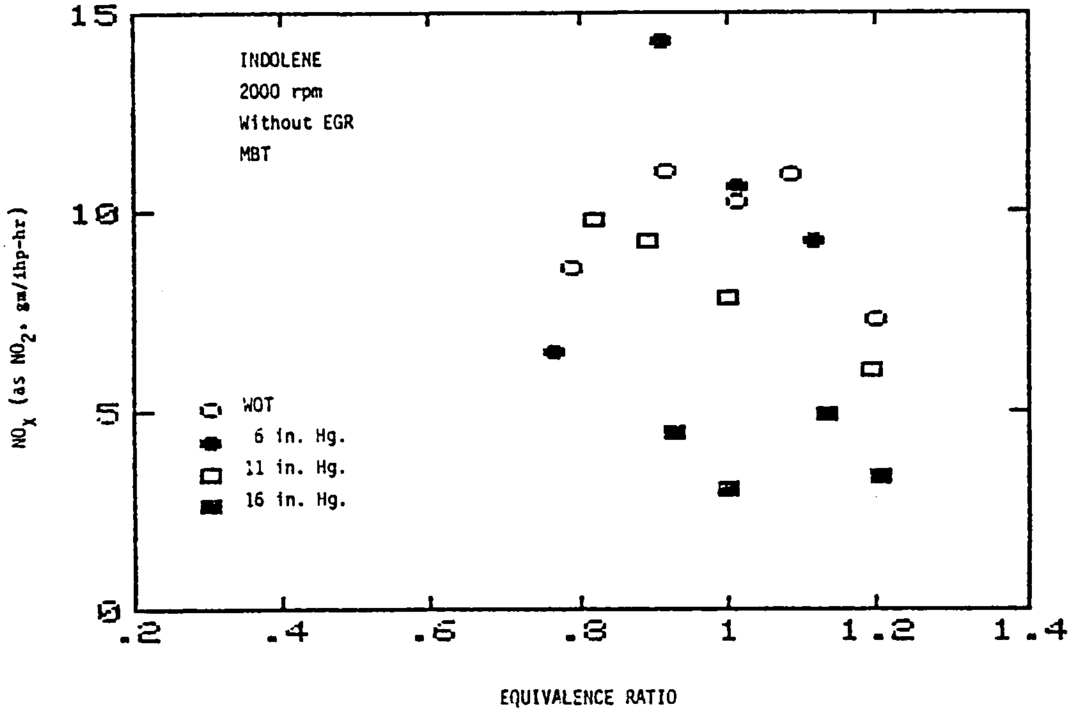

Figure B54: Effect of Equivalence Ratio on Oxides of Nitrogen Emissions at Various Throttle Settings for Indolene at 2000 rpm, without EGR.

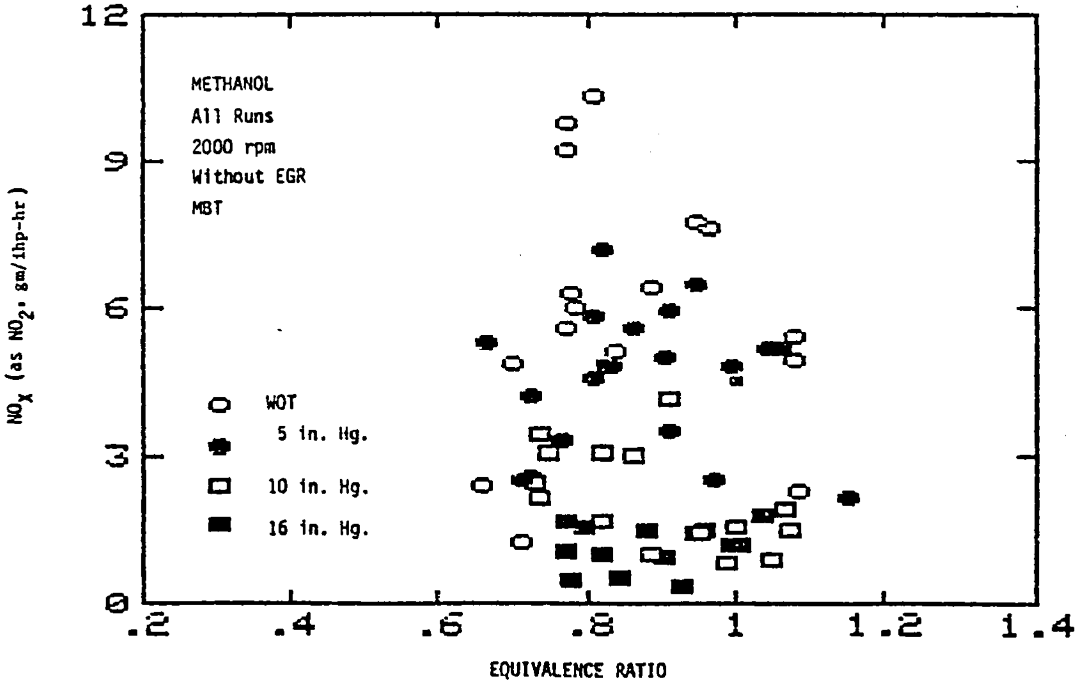

Figure B55: Effect of Equivalence Ratio on Oxides of Nitrogen Emissions at Various Throttle Settings for Methanol at 2000 rpm, without EGR.

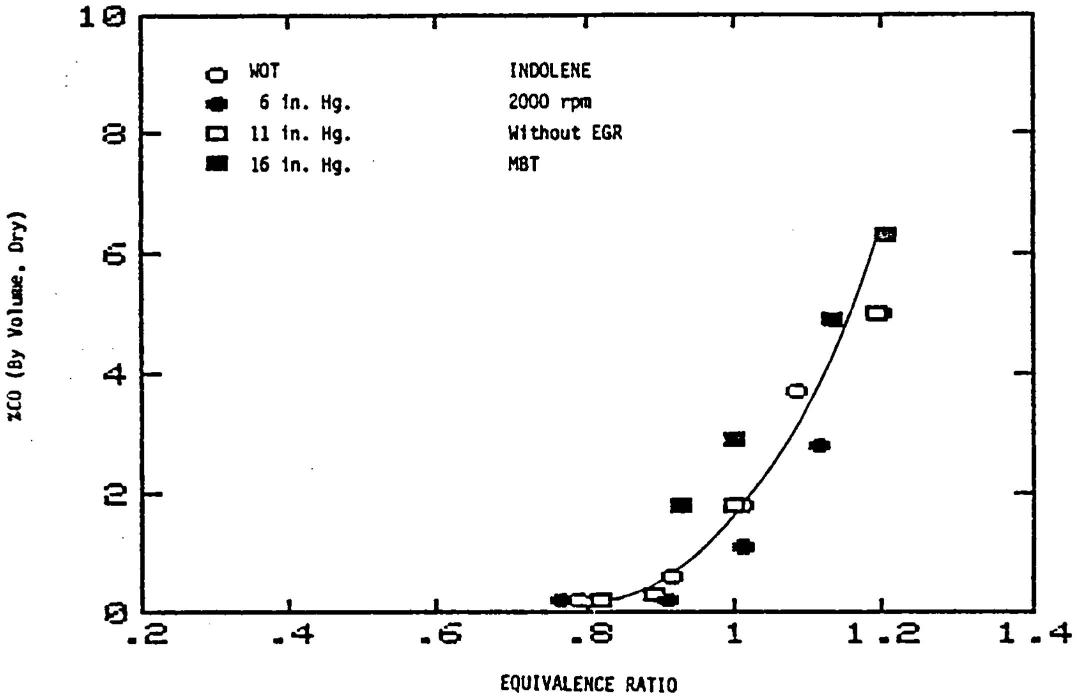

Figure B56: Effect of Equivalence Ratio on Carbon Monoxide Emissions at Various Throttle Settings for Indolene at 2000 rpm, without EGR.

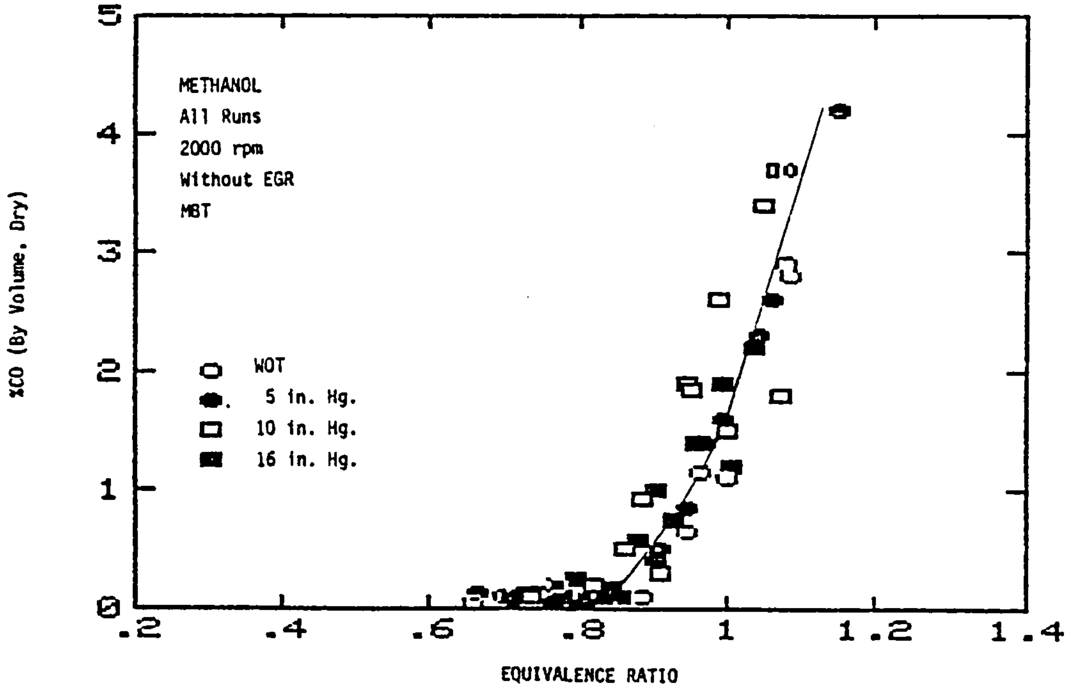

Figure B57: Effect of Equivalence Ratio on Carbon Monoxide Emissions at Various Throttle Settings for Methanol at 2000 rpm, without EGR.

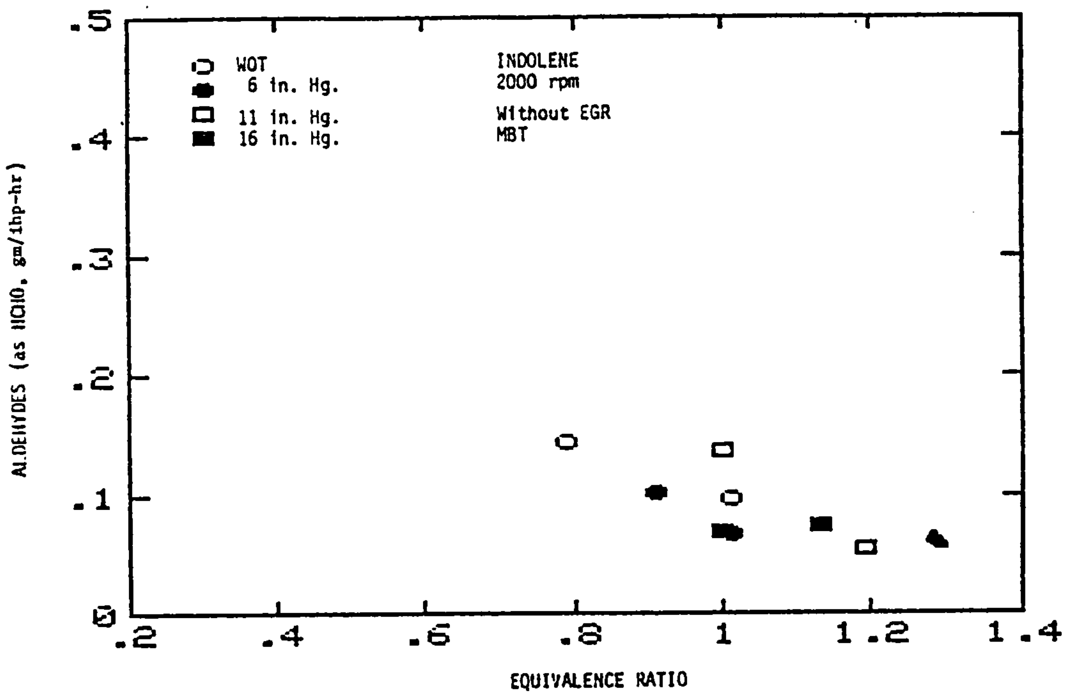

Figure B58: Effect of Equivalence Ratio on Aldehyde Emissions at Various Throttle Settings for Indolene at 2000 rpm, without EGR.

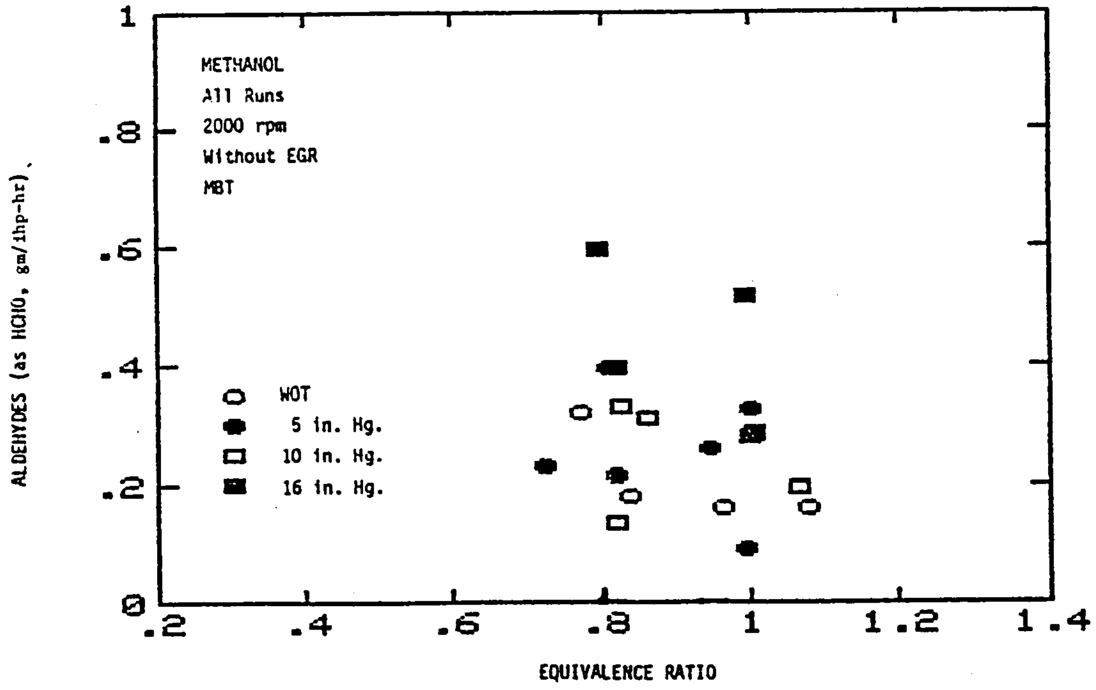

Figure B59: Effect of Equivalence Ratio on Aldehyde Emissions at Various Throttle Settings for Methanol at 2000 rpm, without EGR.

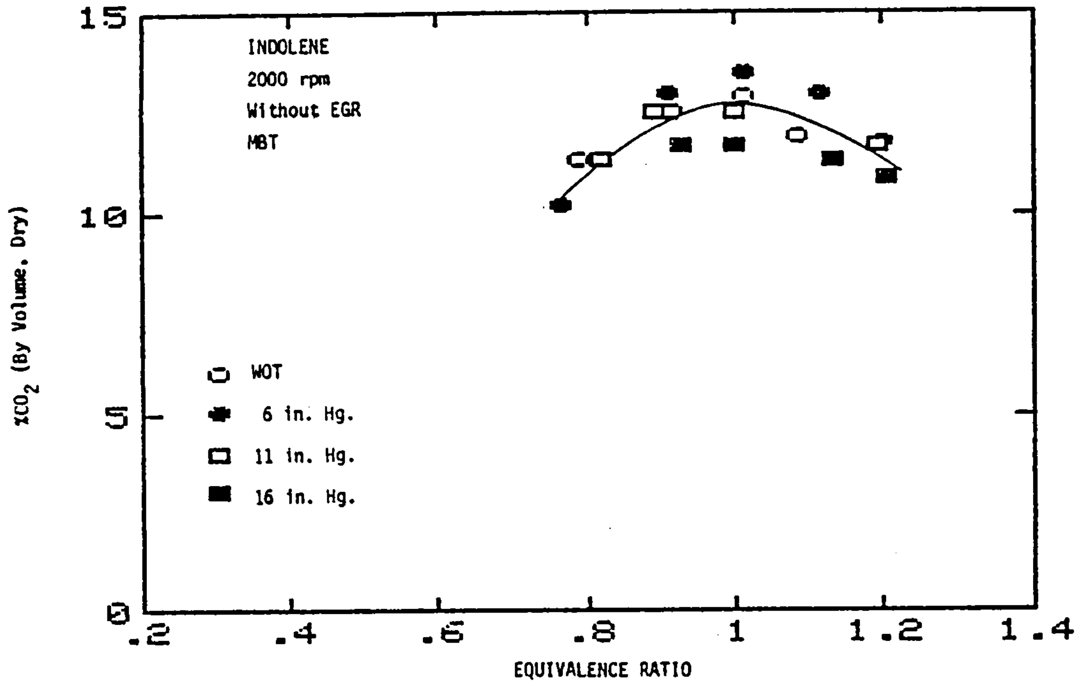

Figure B60: Effect of Equivalence Ratio on Carbon Dioxide Emissions at Various Throttle Settings for Indolene at 2000 rpm, without EGR.

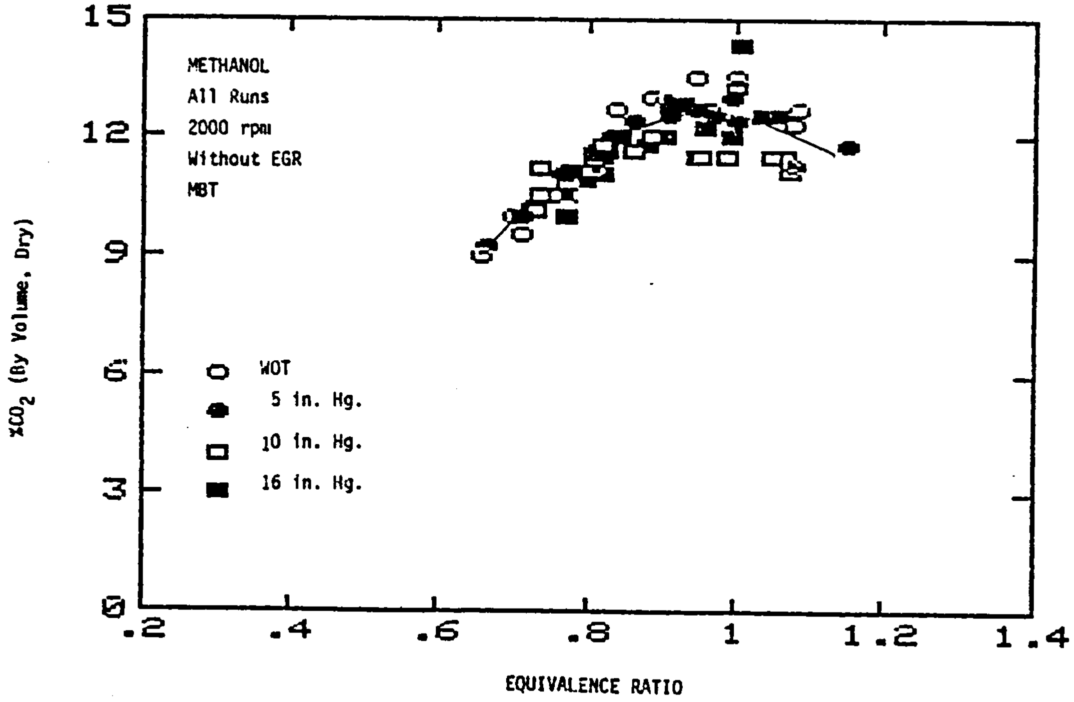

Figure B61: Effect of Equivalence Ratio on Carbon Dioxide Emissions at Various Throttle Settings for at 2000 rpm, without EGR.

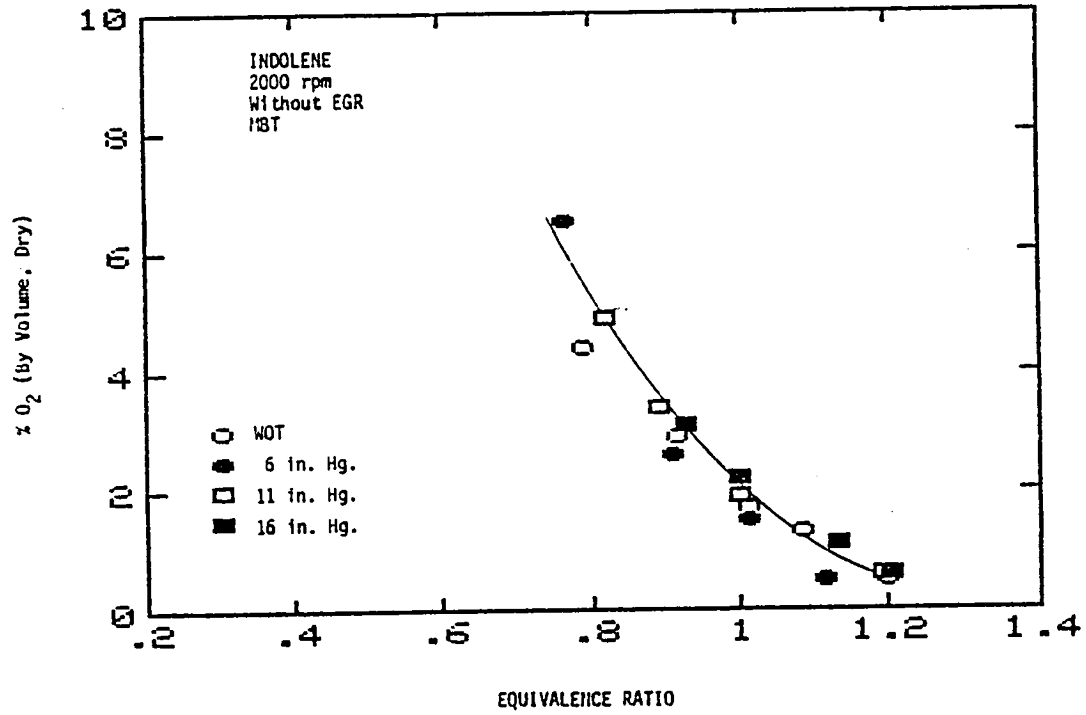

Figure B62: Effect of Equivalence Ratio on Oxygen Emissions at Various Throttle Settings for Indolene at 2000 rpm, without EGR.

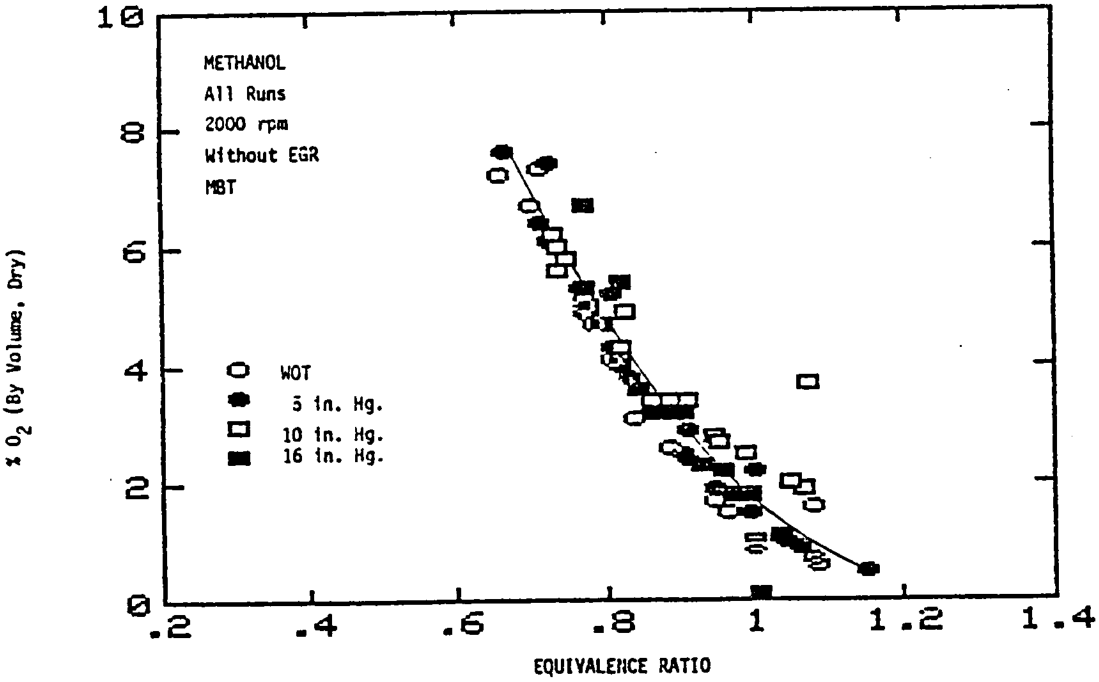

Figure B63: Effect of Equivalence Ratio on Oxygen Emissions at Various Throttle Settings for Methanol at 2000 rpm, without EGR.

shown to be sensitive to spark timing (refs. 5, 10, 11). Maldistribution, as well as error in setting MBT, may very wel have contributed to the scatter in the NO_x emissions data.

Figure B65 shows the brake thermal efficiency (Bte) as a function of power (Bhp) for indolene at 2000 rpm and methanol at 1000 and 2000 rpm. The indolene curve represents ϕ =0.8, 0.9 and 1.0, since these three curves were very close. The methanol curves are shown individually for ϕ =0.8, 0.9, and 1.0. These curves illustrate the gains in efficiency over indolene which are possible with methanol over the entire load range because of reduced compression work and higher flame speeds. Over the power range, the highest efficiencies obtained wtih indolene occurred at ϕ=0.9 while with methanol they occurred at ϕ =0.8 (i.e., at equivalence ratios less than these, the efficencies begin to drop again as shown in Figures B50 and B51). The methanol 1000 rpm curves are included to illustrate the strong effect of speed on the Bte versus Bhp curve. As the speed is decreased while maintaining the power output the throttle must be opened more resulting in a higher thermal efficiency at a particular power level.

In terms of power and efficiency, the methanol and indolene curves are similar except that owing to methanol's higher flame speed, the methanol lean misfire limits are shifted to the lean side. If, for comparison, the methanol curves were shifted right (richer) until the effective lean misfire limits of the two fuels matched (approximately 0.1 equivalence ratios), then the similarities in the performance characteristics of the two fuels would become apparent. Comparing the present results in this fashion, power output would be about the same for both fuels over the entire range of ϕ and throttle settings. The thermal efficiency curves would have the same slope and peak at

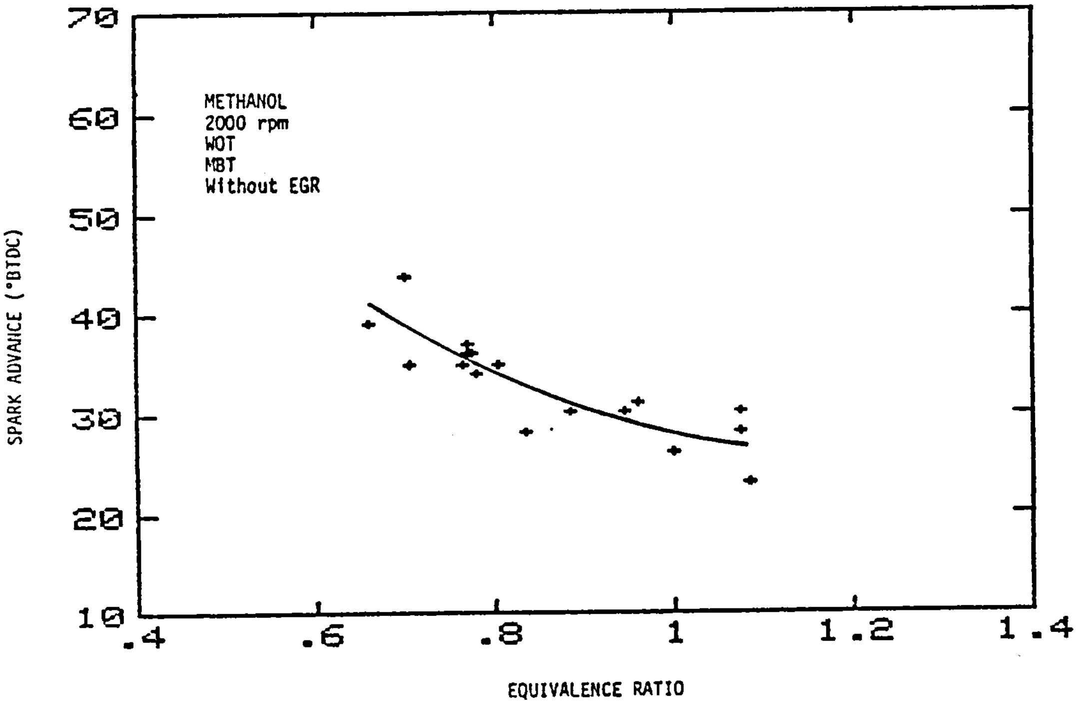

Figure B64: Effect of Equivalence Ratio on MBT Spark Advance for Methanol at 2000 rpm, WOT.

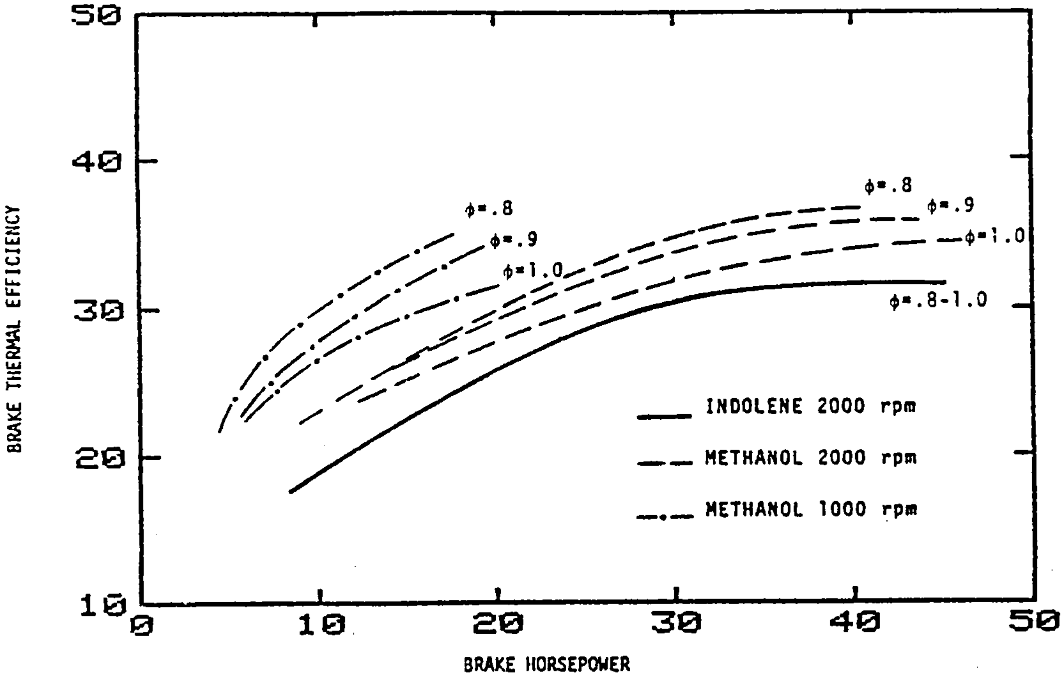

Figure B65: Brake Thermal Efficiency as a Function of Brake Horsepower for Methanol and Indolene.

the same location, but the efficiencies would all be higher for methanol. This increase is due to the larger heat of vaporization of methanol over indolene resulting in cooling of the charge during the compression stroke and a reduction of the compression work. Also, the flame speed for methanol is higher resulting in lower heat losses. For this study, thermal efficiencies would be 15% higher at WOT for methanol as compared to indolene, and even higher at the lower throttle settings.

B.3 Methane and Natural Gas Results

Another alternate fuel currently of interest for automotive engines is methane. Tests were conducted with methane and natural gas to provide additional methane multicylinder engine performance and emissions data.

The results from operation on methane and natural gas at 1000 and 2000 rpm without EGR are presented in this section. The ambient conditions for the testing are listed in Table B4. The results are again presented as plots of engine performance parameters as functions of equivalence ratio as described earlier.

Performance and emissions results were obtained with both methane and natural gas because the natural gas used was lower in methane than that found in other areas of the country. The composition of the natural gas is given in Table B5. Although the majority of the natural gas was methane (76%), there were high percentages of ethane and propane in the fuel. The performance and emissions results from operation at 2000 rpm are presented first, followed by the 1000 rpm results.

Plots of Bhp, Bsfc, and Bte as functions of equivalence

Table B4: Methane and Natural gas Test Conditions

Fuel	Speed rpm	$T_{ambient}$ F	$P_{ambient}$ in. Hg	T_{intake} F	$T_{coolant}$ F
Methane	2000	70	29.71	85-105	190-205
Natural Gas	2000	74	29.61	90-105	195-205
Methane	1000	72	30.14	85-110	185-200
Natural Gas	1000	74	30.02	85-120	190-200

Table B5: Composition of Natural Gas

Component		Volume %
CH_4	methane	76.33
C_2H_6	ethane	13.57
C_3H_8	propane	5.31
C_4H_{10}	butane	1.46
C_5H_{12}	pentane	0.31
C_6H_{14}	hexane	0.10

ratio for methane and natural gas at 2000 rpm are shown in Figures B66 through B71. Power output from both fuels was about the same and was approximately 37 Bhp at ϕ=1 (about 10 Bhp less than that from liquid methanol). As can be seen in Figures B66 and B67, the power output from methane decreased more at lean misfire than that from natural gas, and lean misfire occurred at about ϕ 0.65 for both fuels. The minimum Bsfc for methane and natural gas were 0.36 and 0.37 lbm/bhp-hr, respectively, and occurred at about ϕ=0.8. The maximum Bte for both fuels was about 33% and also occurred at ϕ=0.8 and WOT. As can be seen in Figures B70 and B71, the WOT Bte curves were very flat, and operation on either fuel at equivalence ratios beteen ϕ =1.0 and ϕ=0.7 resulted in very

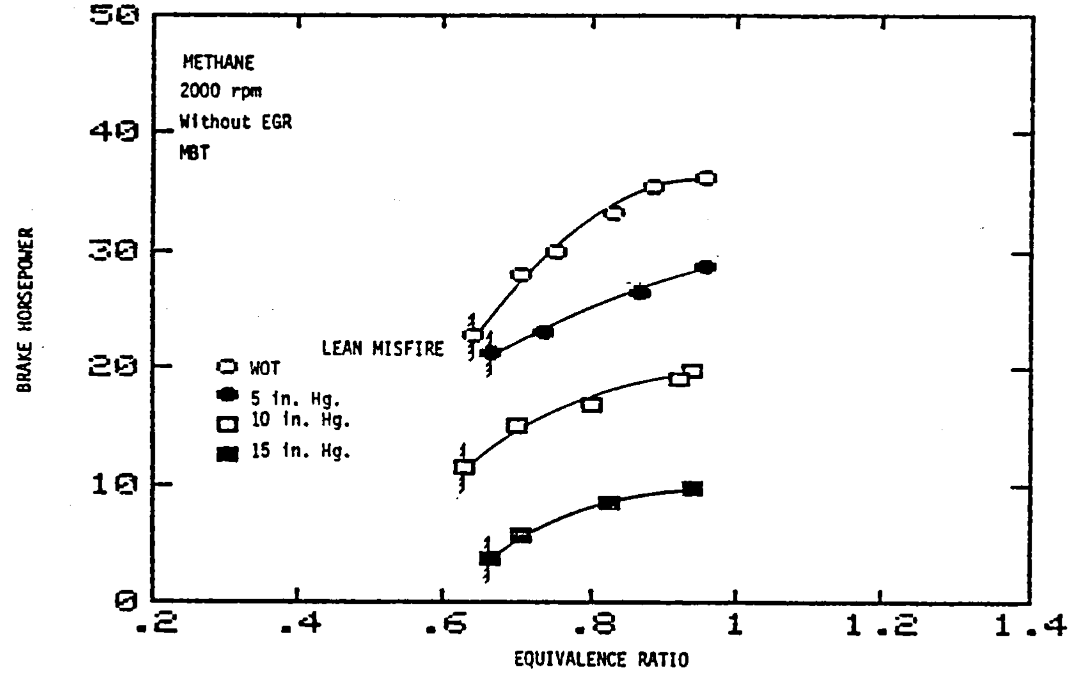

Figure B66: Effect of Equivalence Ratio on Brake Horsepower at Various Throttle Settings for Methane at 2000 rpm, without EGR.

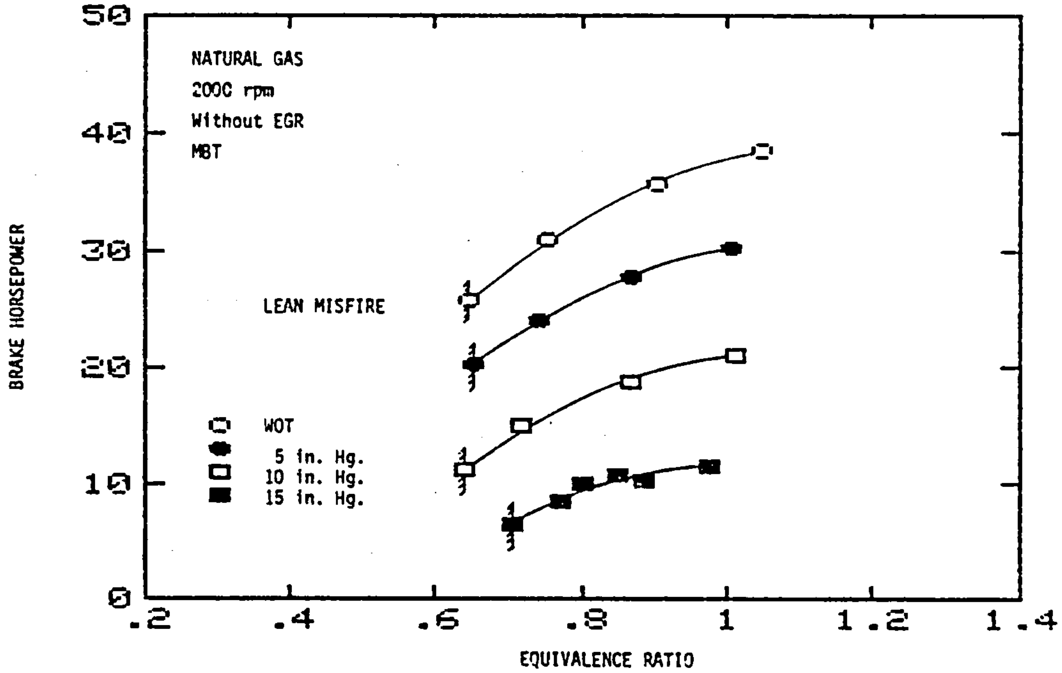

Figure B67: Effect of Equivalence Ratio on Brake Horsepower at Various Throttle Settings for Natural Gas at 2000 rpm, without EGR.

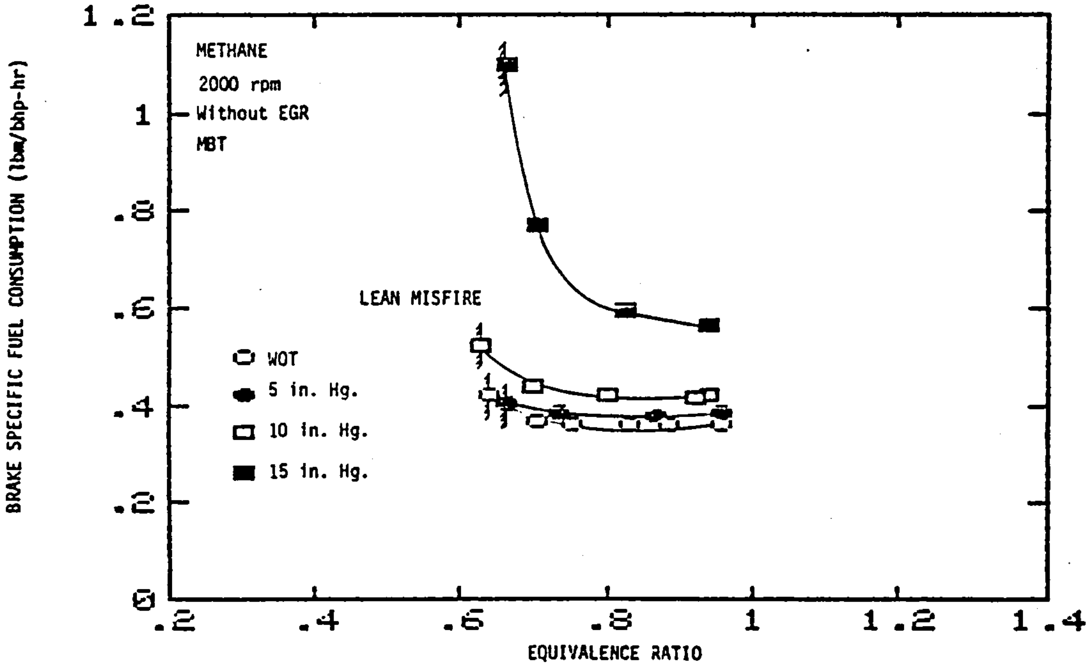

Figure B68: Effect of Equivalence Ratio on Brake Specific Fuel Consumption at Various Throttle Settings for Methane at 2000 rpm, without EGR.

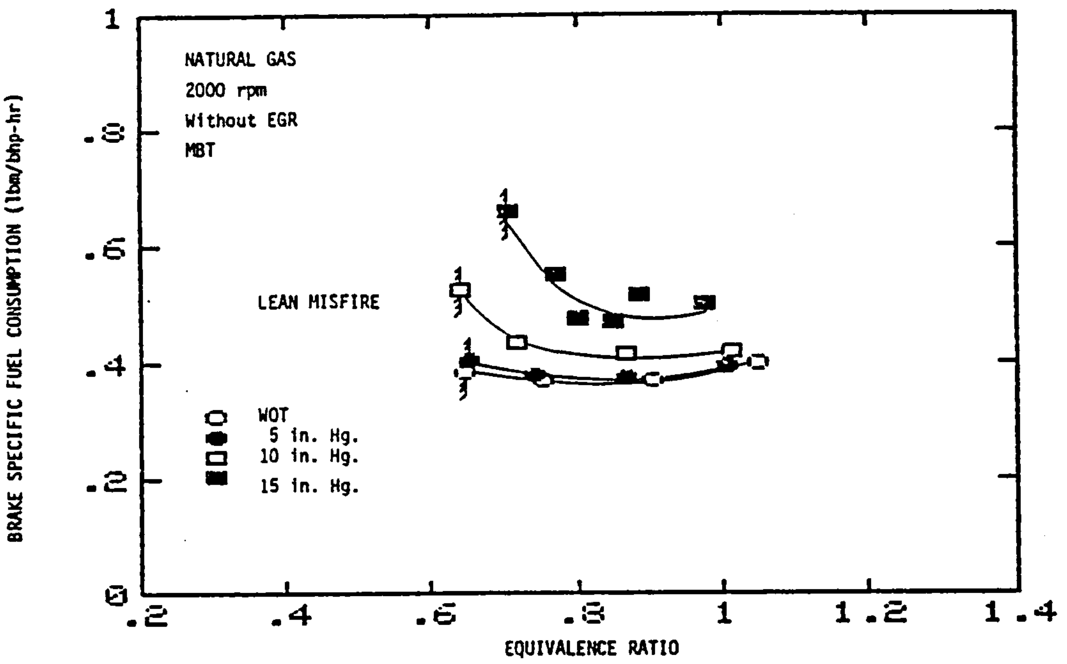

Figure B69: Effect of Equivalence Ratio on Brake Specific Fuel Consumption at Various Throttle Settings for Natural Gas at 2000 rpm, without EGR.

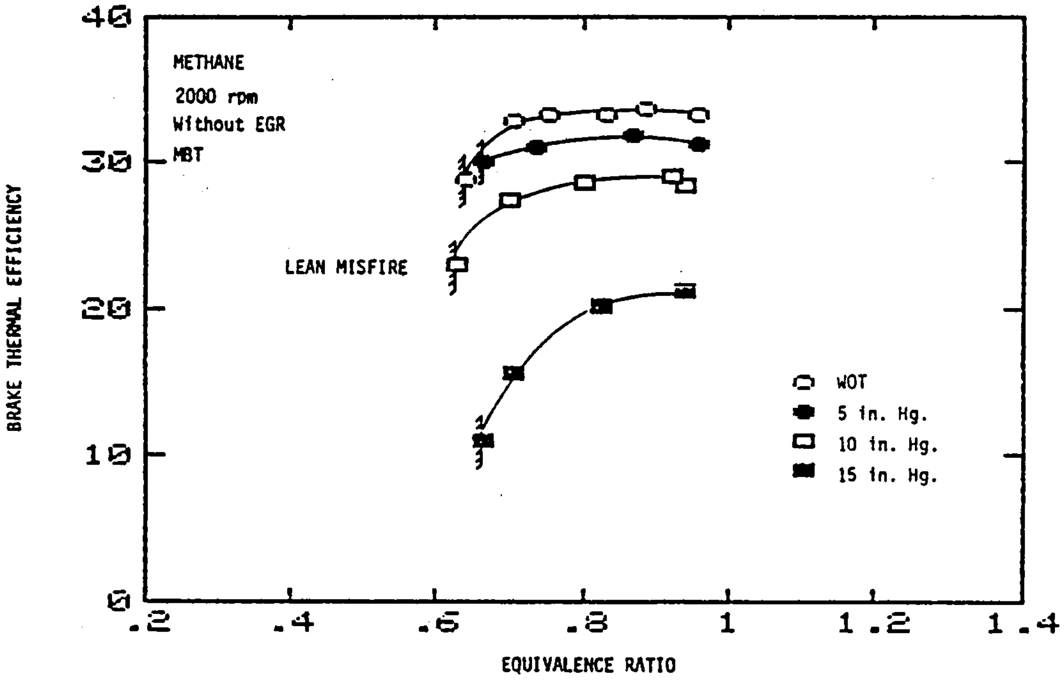

Figure B70: Effect of Equivalence Ratio on Brake Thermal Efficiency at Various Throttle Settings for Methane at 2000 rpm, without EGR.

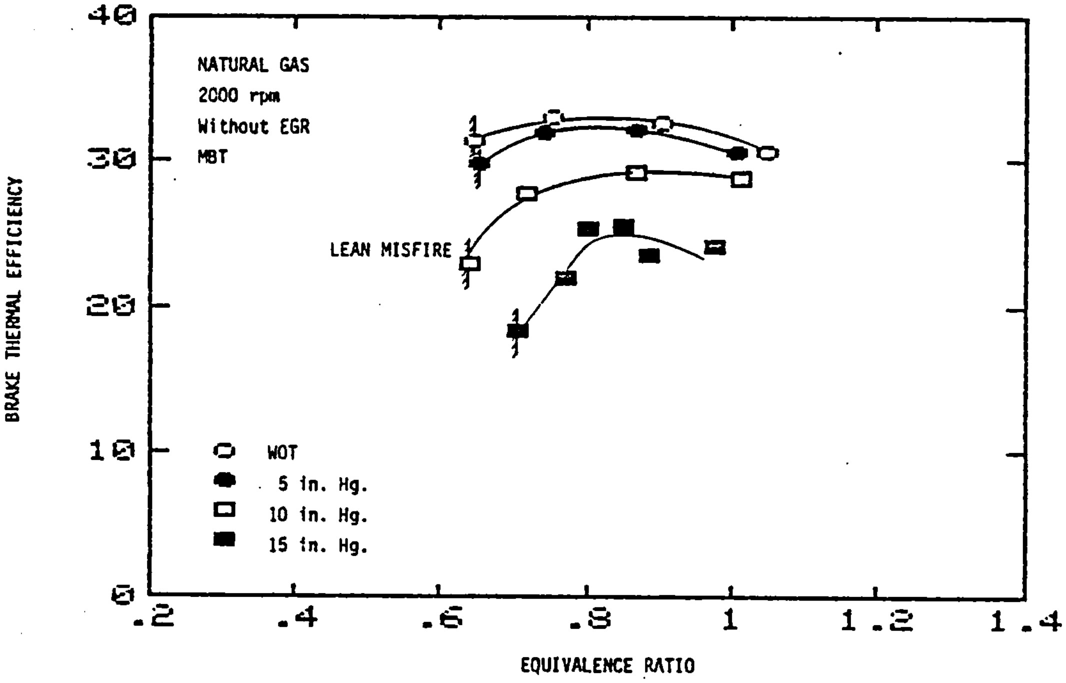

Figure B71: Effect of Equivalence Ratio on Brake Thermal Efficiency at Various Throttle Settings for Natural Gas at 2000 rpm, without EGR.

little change in the thermal efficiency.

Plots of hydrocarbons, NO_x, aldehydes, CO, CO_2, and O_2 as functions of ϕ are shown in Figures B72 through B83. Hydrocarbon emissions were a function of equivalence ratio but weren't affected by throttle setting. The minimum hydrocarbon emission occurred at about $\phi = 0.9$ for both fuels with operation on natural gas resulting in the lower emission. NO_x emissions from the two fuels, shown in Figures B74 and B75, were a function of both throttle setting and equivalence ratio with the maximum emission occurring at WOT and $\phi = 0.9$. NO_x emissions from methane were higher than those from natural gas. Aldehyde emissions from the two fuels were about the same.

CO, CO_2, and O_2 emissions were only a function of equivalence ratio and were similar from both fuels. Although no testing was done at rich equivalence ratios with methane, it appears that the CO emissions from methane began to increase at leaner equivalence ratios than for natural gas.

Overall, engine performance on methane and natural gas was about the same. This would be expected since natural gas is mostly methane.

The results from operation on methane and natural gas at 1000 rpm without EGR are presented in Figures B84 through B101 and are arranged in the same order as before. The results from operation on methane and natural gas at 1000 rpm were very similar to the results at 2000 rpm. Thus, these performance and emission results follow without further discussion.

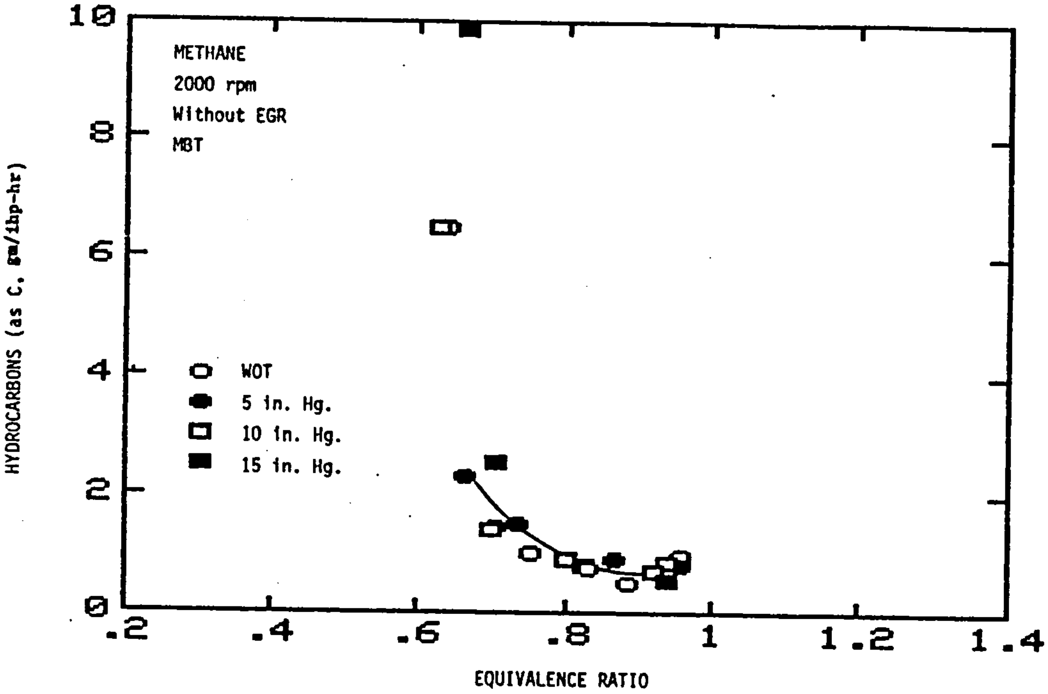

Figure B72: Effect of Equivalence Ratio on Hydrocarbon Emissions at Various Throttle Settings for Methane at 2000 rpm, without EGR.

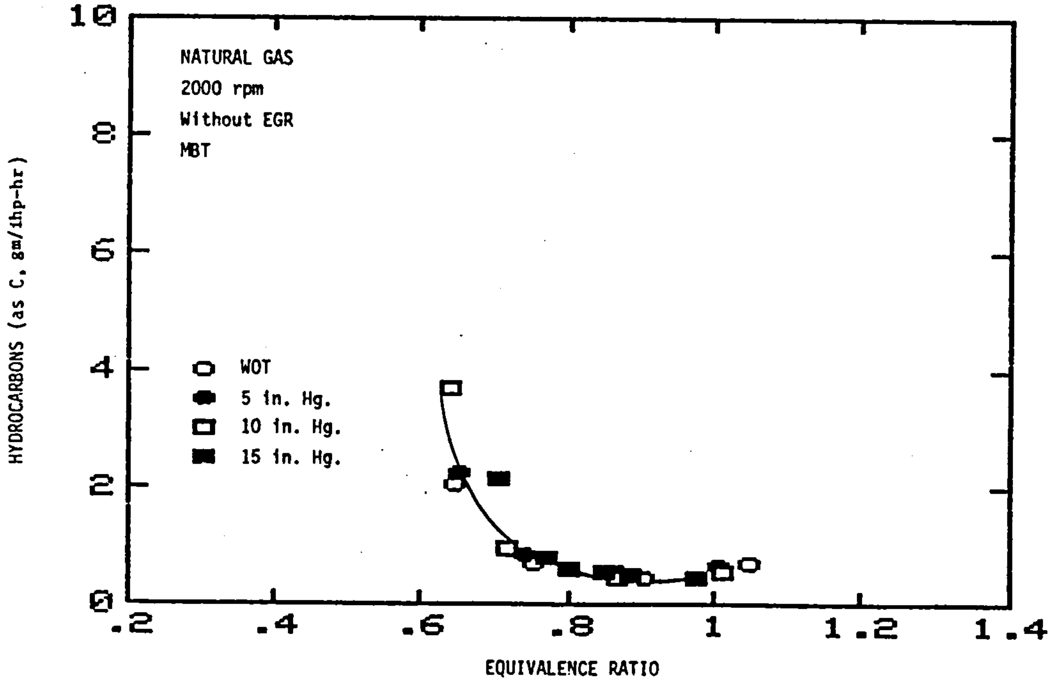

Figure B73: Effect of Equivalence Ratio on Hydrocarbon Emissions at Various Throttle Settings for Natural Gas at 2000 rpm, without EGR.

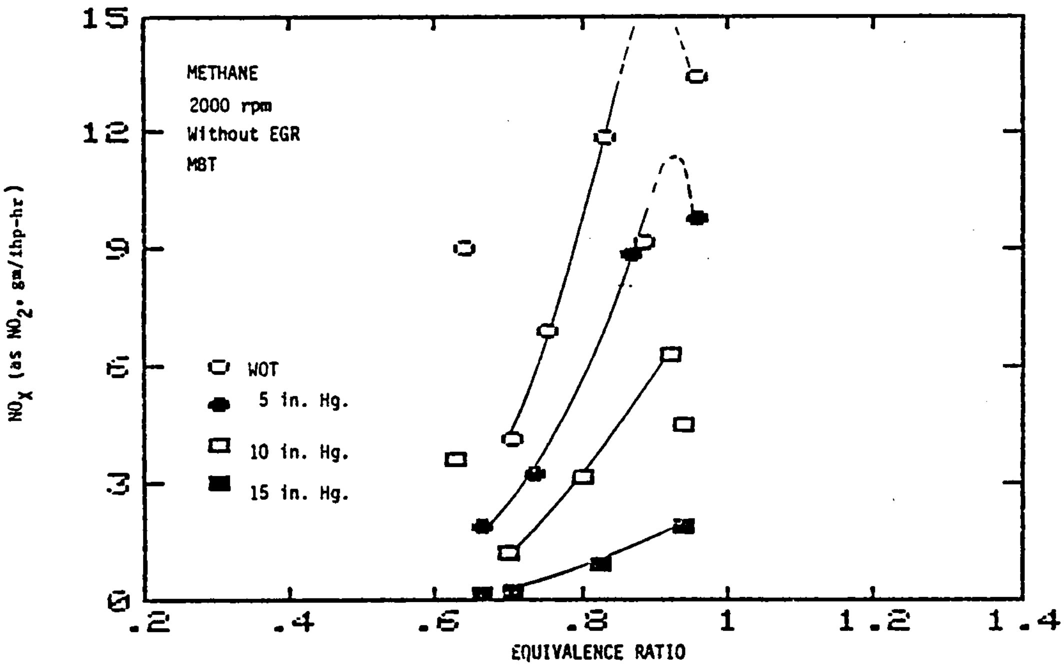

Figure B74: Effect of Equivalence Ratio on Oxides of Nitrogen Emission at Various Throttle Settings for Methane at 2000 rpm, without EGR.

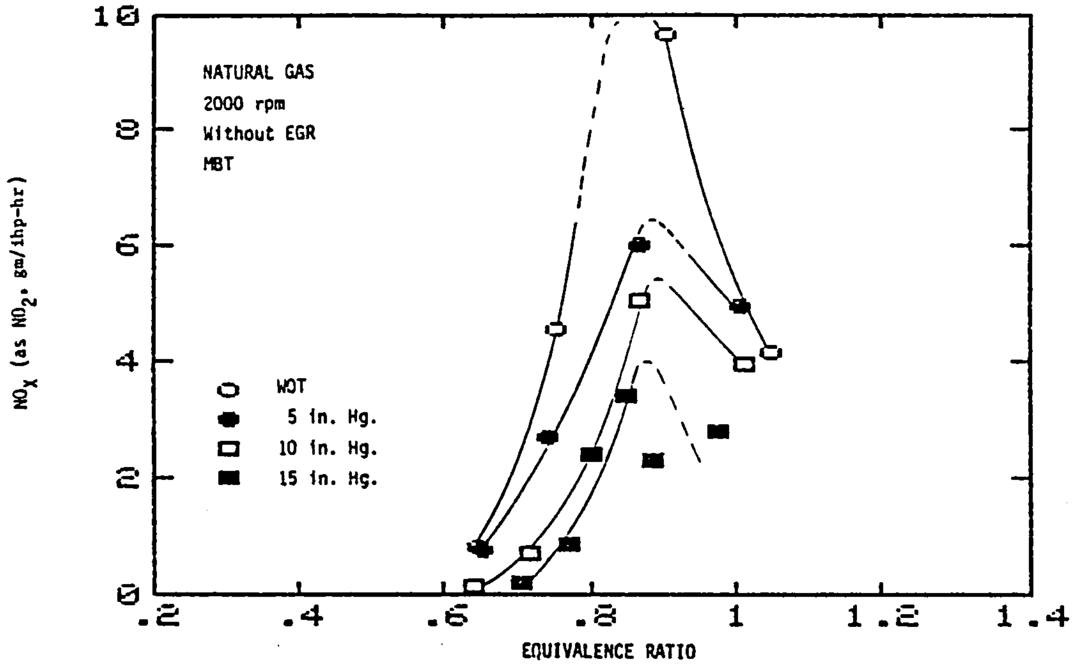

Figure B75: Effect of Equivalence Ratio on Oxides of Nitrogen Emissions at Various Throttle Settings for Natural Gas at 2000 rpm, without EGR.

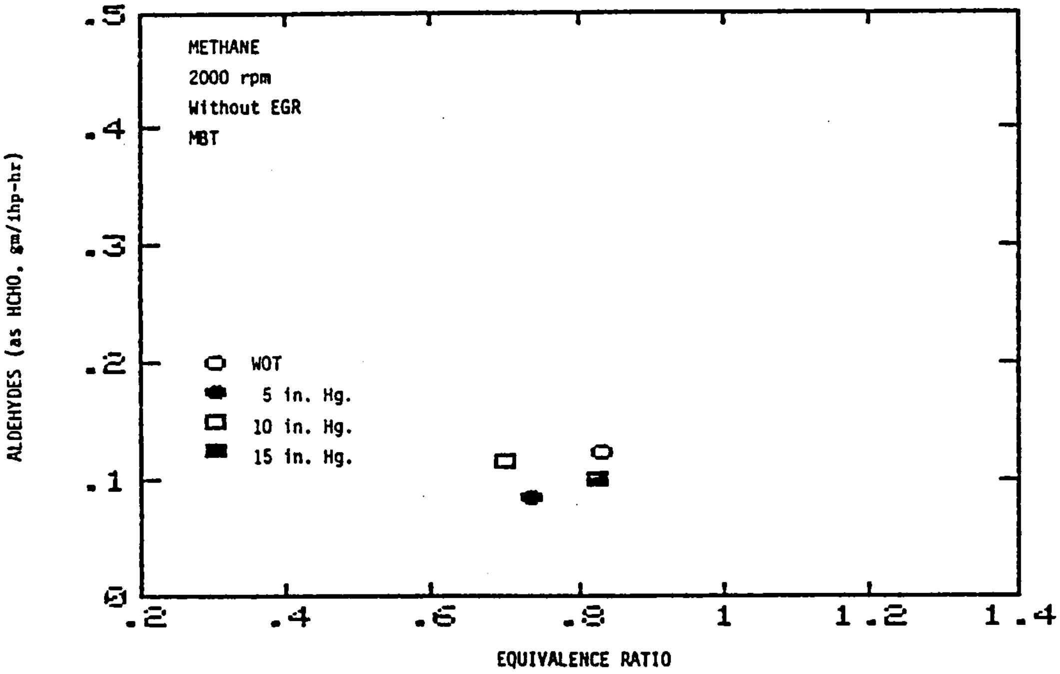

Figure B76: Effect of Equivalence Ratio on Aldehyde Emissions at Various Throttle Settings for Methane at 2000 rpm, without EGR.

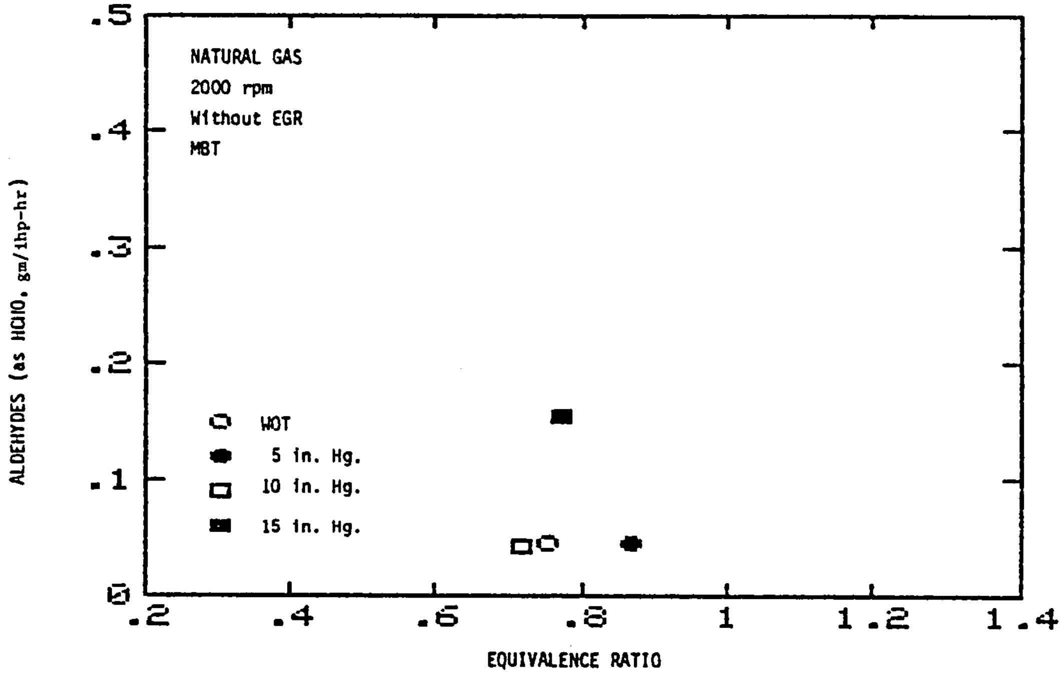

Figure B77: Effect of Equivalence Ratio on Aldehyde Emissions at Various Throttle Settings for Natural Gas at 2000 rpm, without EGR.

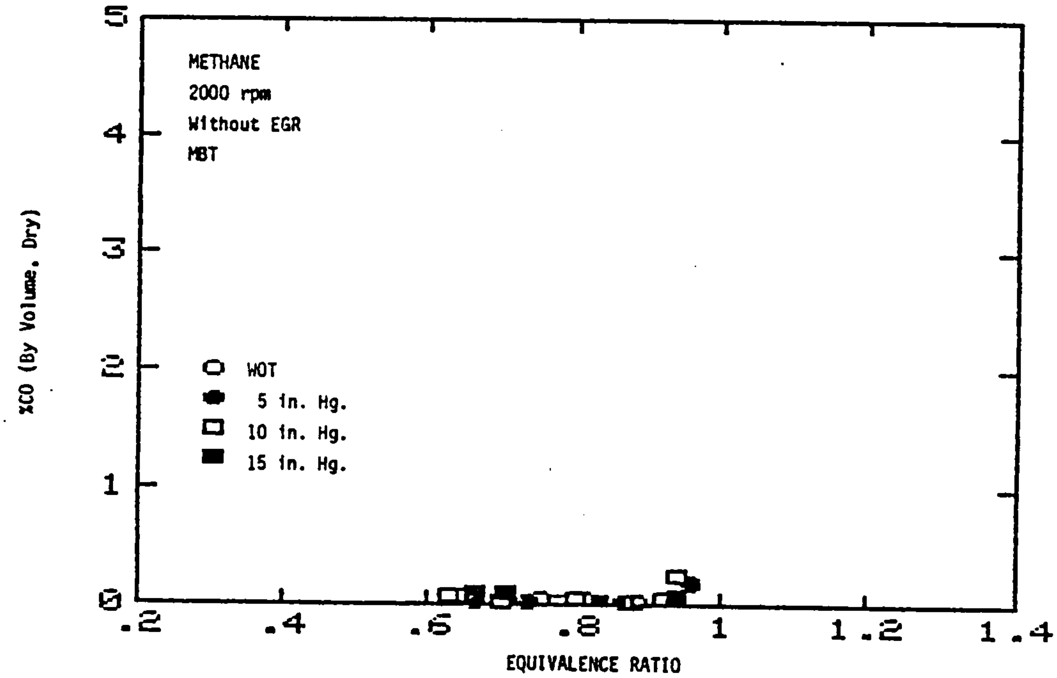

Figure B78: Effect of Equivalence Ratio on Carbon Monoxide Emissions at Various Throttle Settings for Methane at 2000 rpm, without EGR.

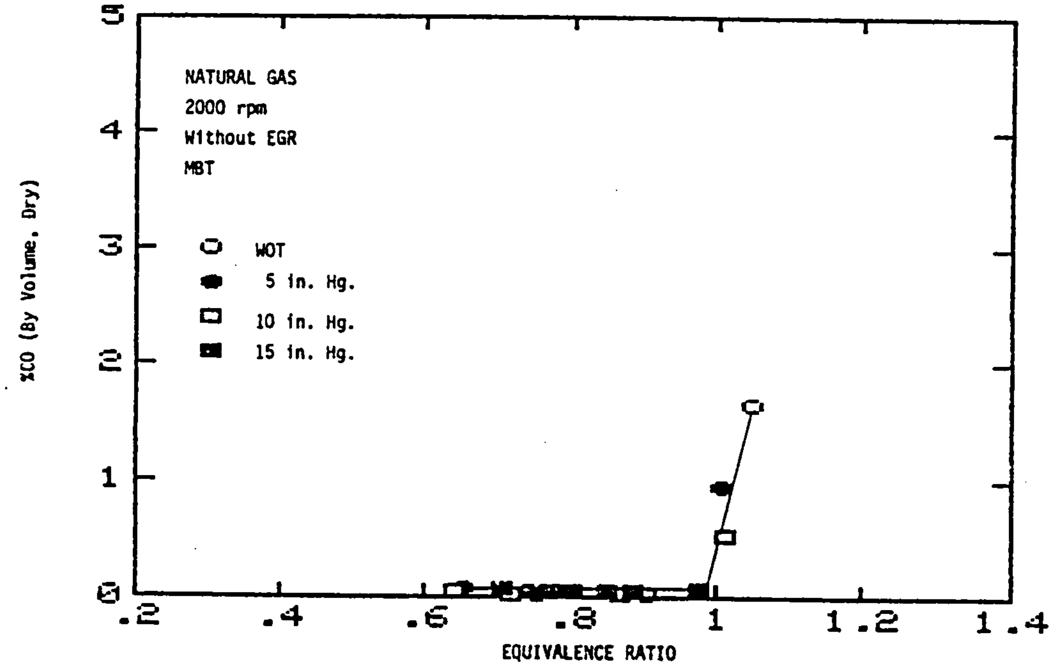

Figure B79: Effect of Equivalence Ratio on Carbon Monoxide Emissions at Various Throttle Settings for Natural Gas at 2000 rpm, without EGR.

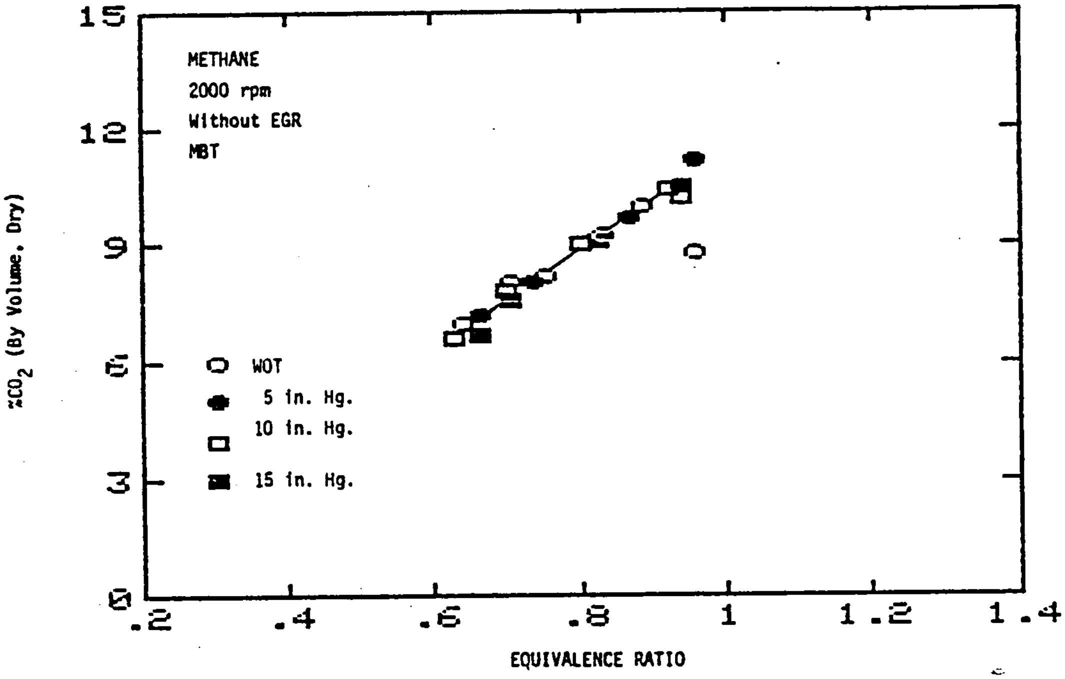

Figure B80: Effect of Equivalence Ratio on Carbon Dioxide Emissions at Various Throttle Settings for Methane at 2000 rpm, without EGR.

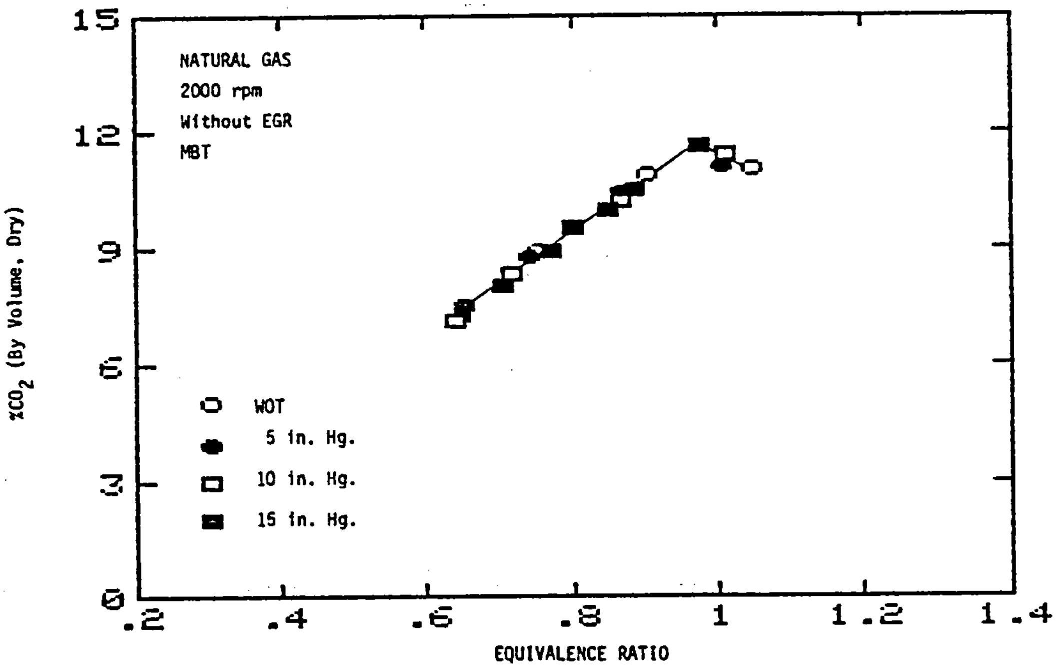

Figure B81: Effect of Equivalence Ratio on Carbon Dioxide Emissions at Various Throttle Settings for Natural Gas at 2000 rpm, without EGR.

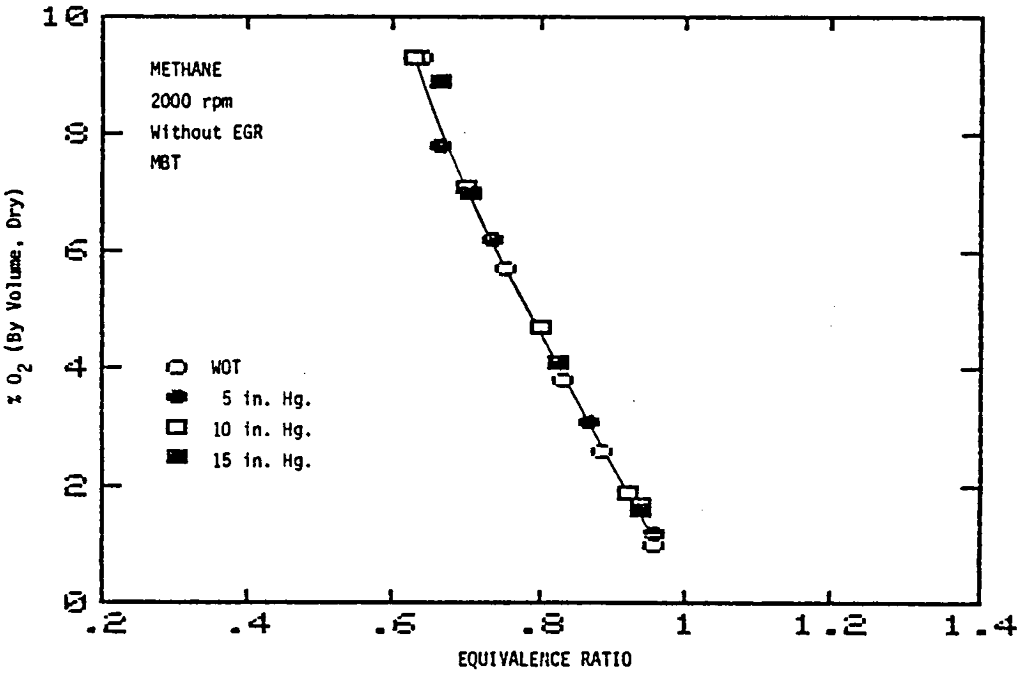

Figure B82: Effect of Equivalence Ratio on Oxygen Emissions at Various Throttle Settings for Methane at 2000 rpm, without EGR.

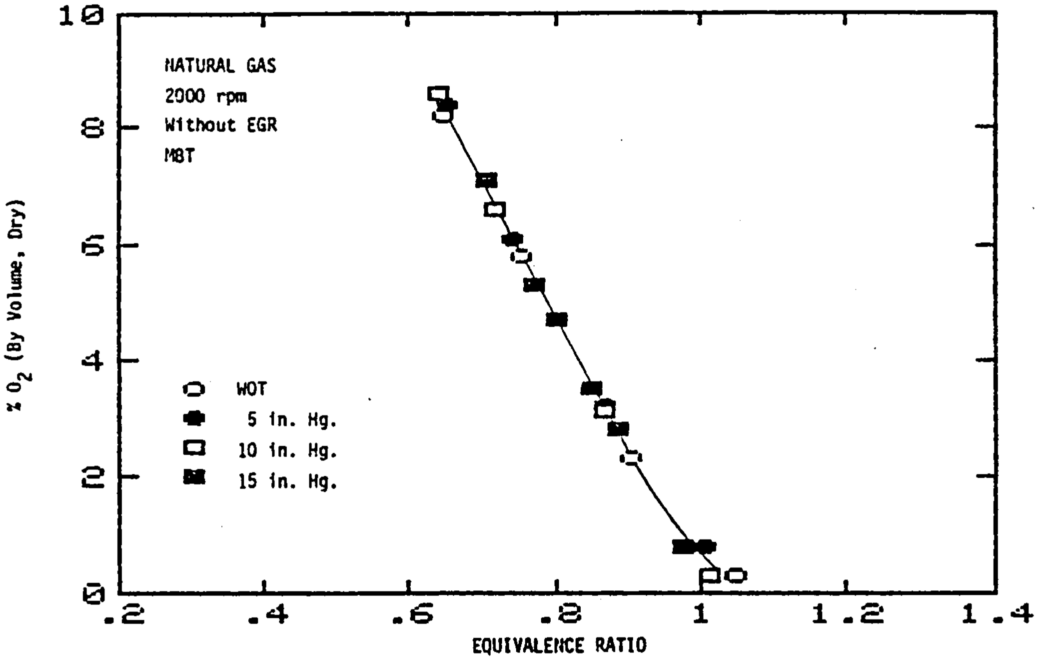

Figure B83: Effect of Equivalence Ratio on Oxygen Emissions at Various Throttle Settings for Natural Gas at 2000 rpm, without EGR.

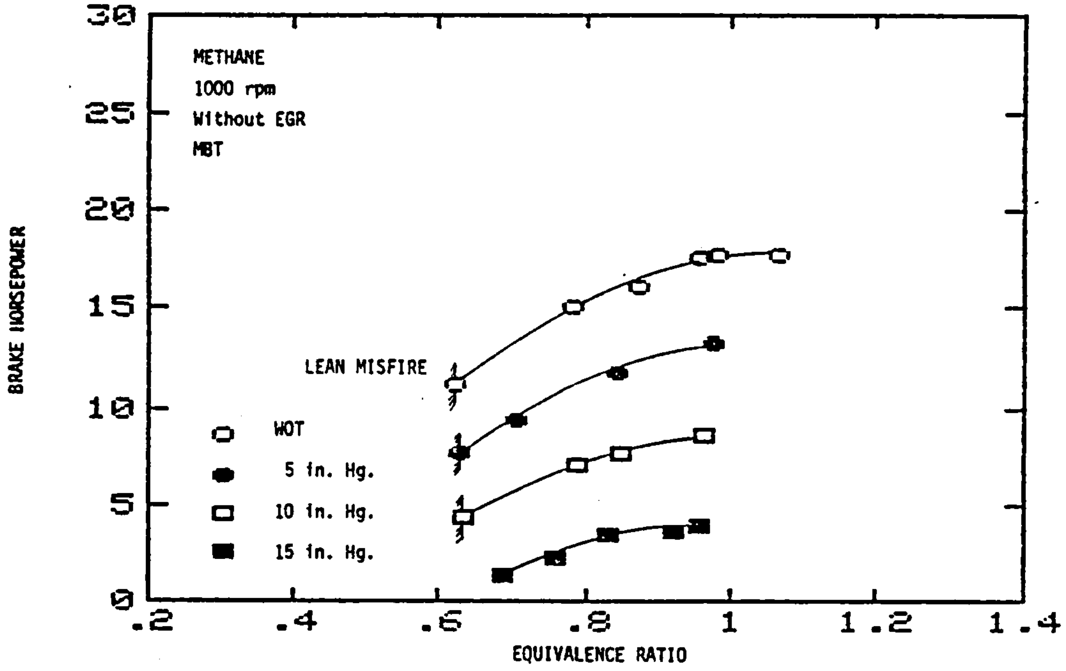

Figure B84: Effect of Equivalence Ratio on Brake Horsepower at Various Throttle Settings for Methane at 1000 rpm, without EGR.

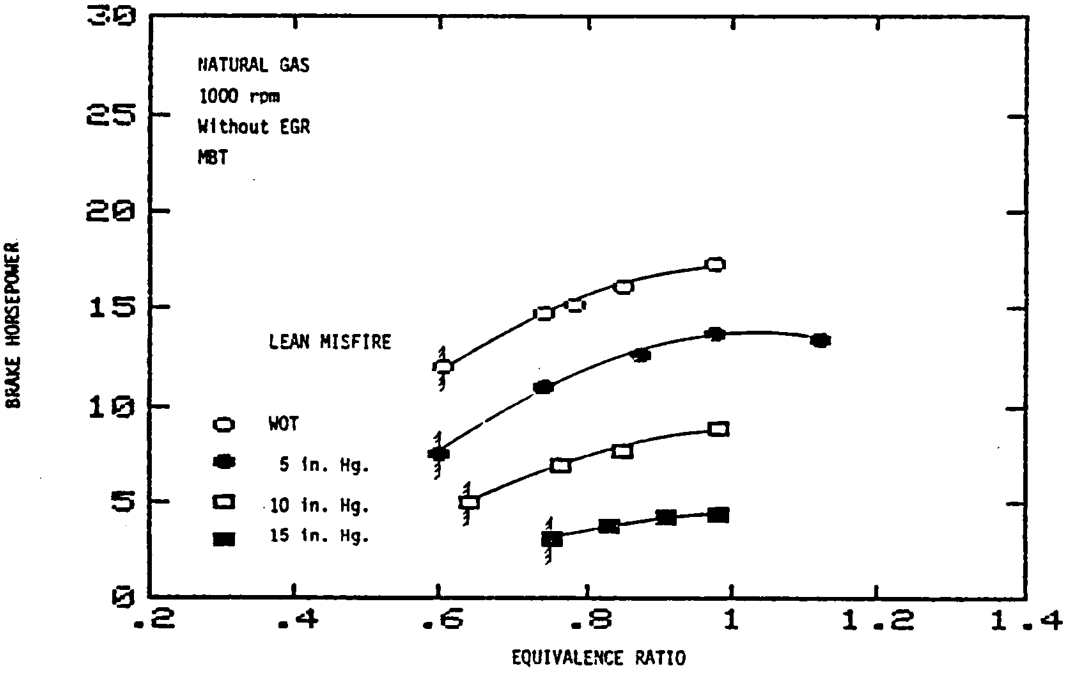

Figure B85: Effect of Equivalence Ratio on Brake Horsepower at Various Throttle Settings for Natural Gas at 1000 rpm, without EGR.

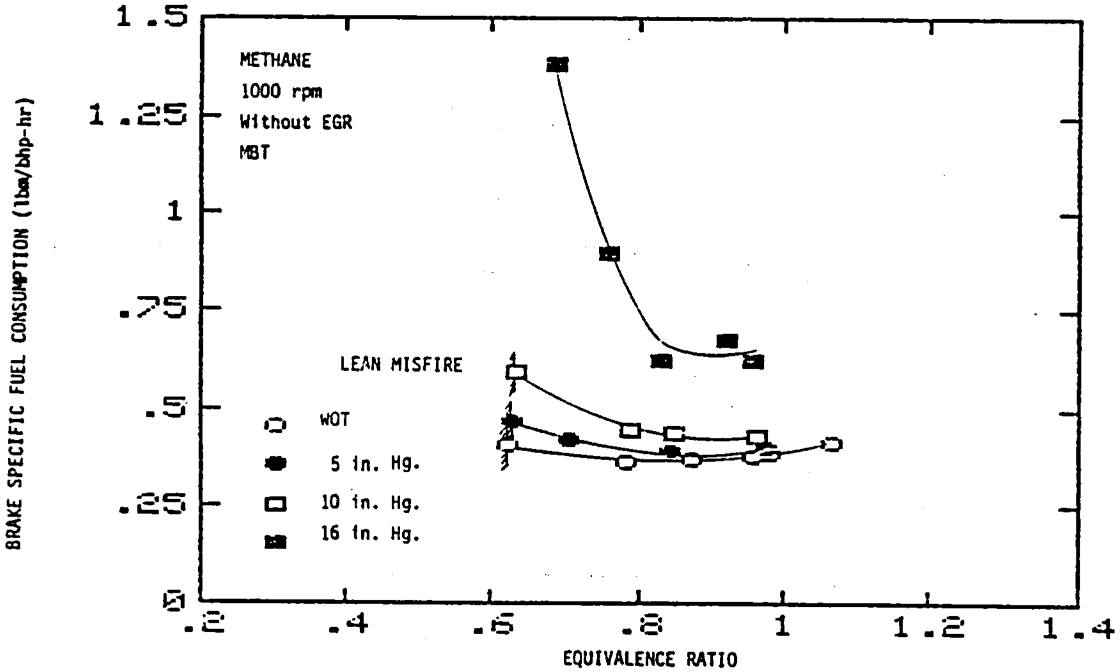

Figure B86: Effect of Equivalence Ratio on Brake Specific Fuel Consumption at Various Throttle Settings for Methane at 1000 rpm, without EGR.

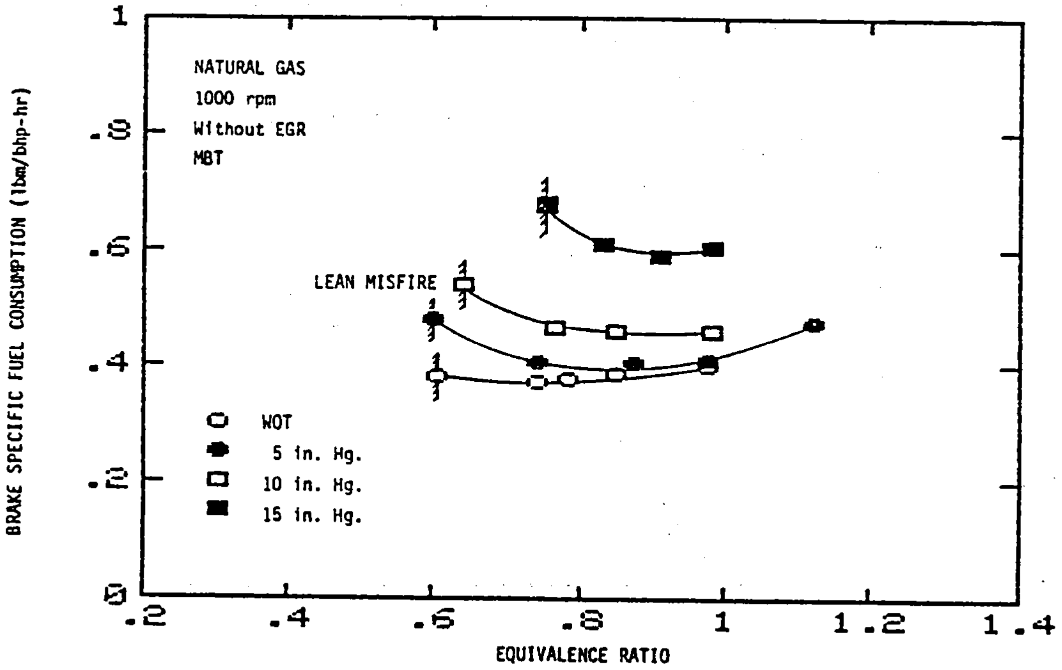

Figure B87: Effect of Equivalence Ratio on Brake Specific Fuel Consumption at Various Throttle Settings for Natural Gas at 1000 rpm, without EGR.

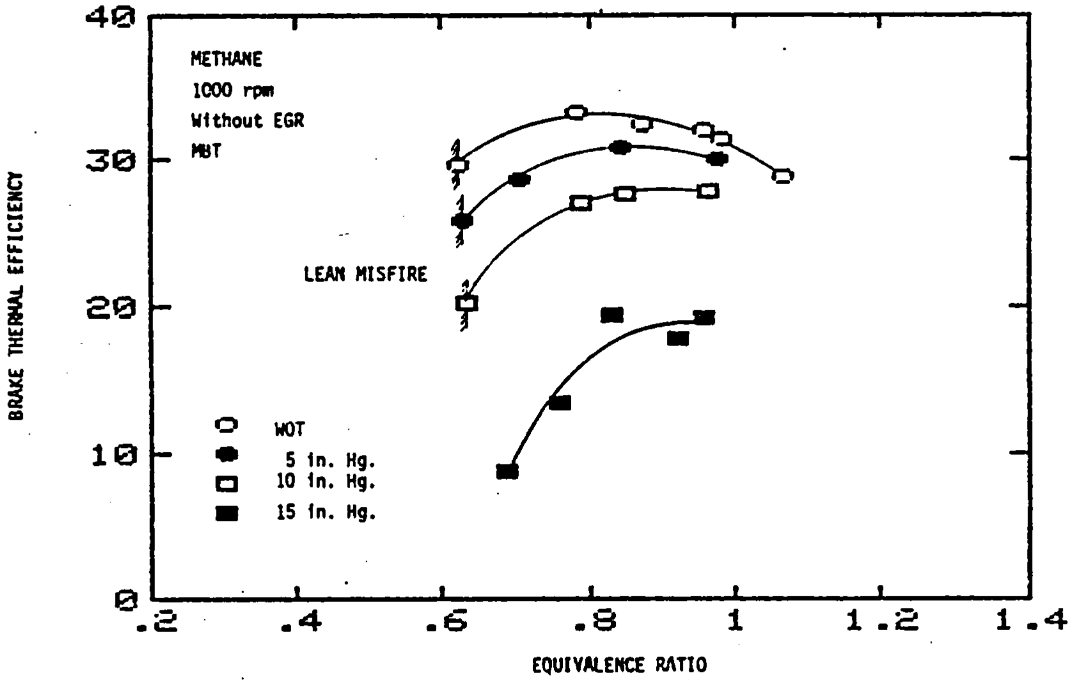

Figure B88: Effect of Equivalence Ratio on Brake Thermal Efficiency at Various Throttle Settings for Methane at 1000 rpm, without EGR.

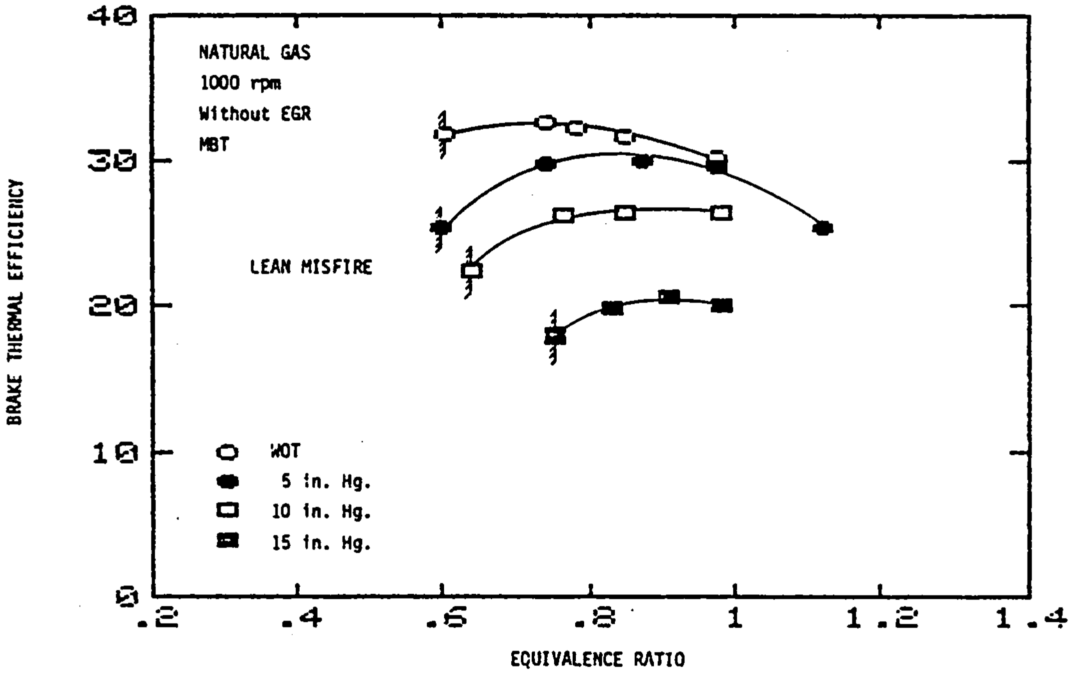

Figure B89: Effect of Equivalence Ratio on Brake Thermal Efficiency at Various Throttle Settings for Natural Gas at 1000 rpm, without EGR.

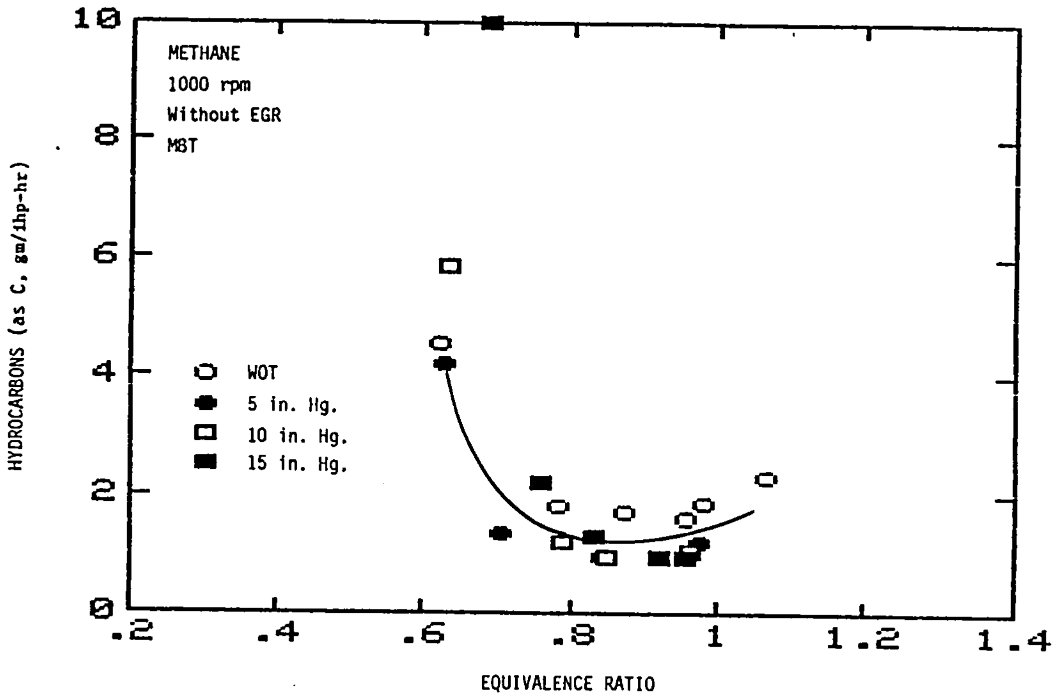

Figure B90: Effect of Equivalence Ratio on Hydrocarbon Emissions at Various Throttle Settings for Methane at 1000 rpm, without EGR.

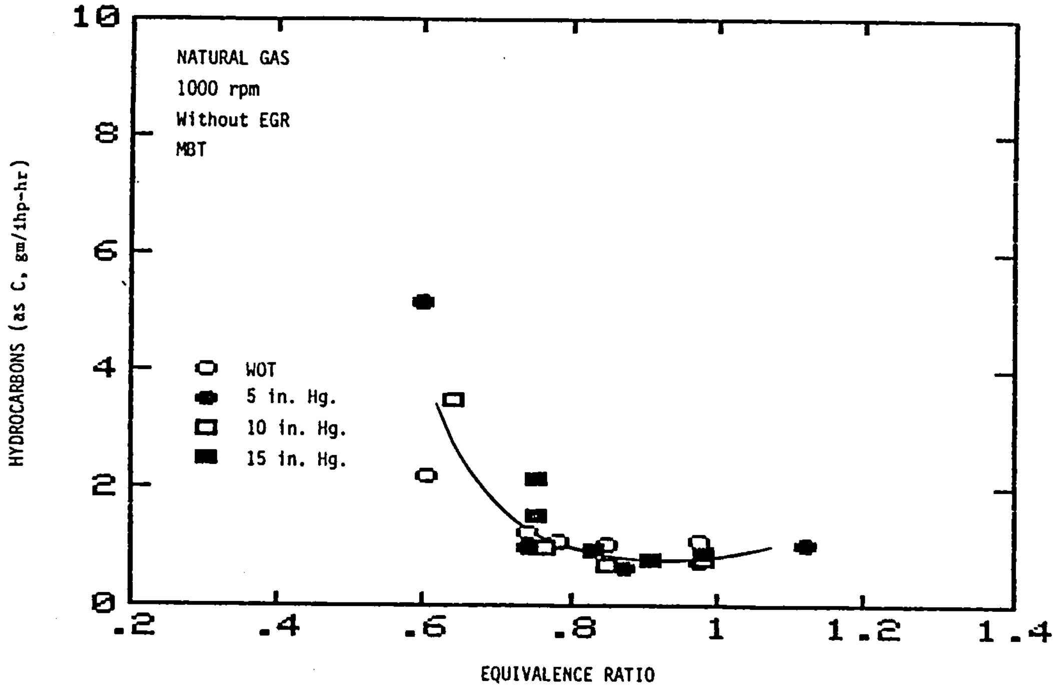

Figure B91: Effect of Equivalence Ratio on Hydrocarbon Emissions at Various Throttle Settings for Natural Gas at 1000 rpm, without EGR.

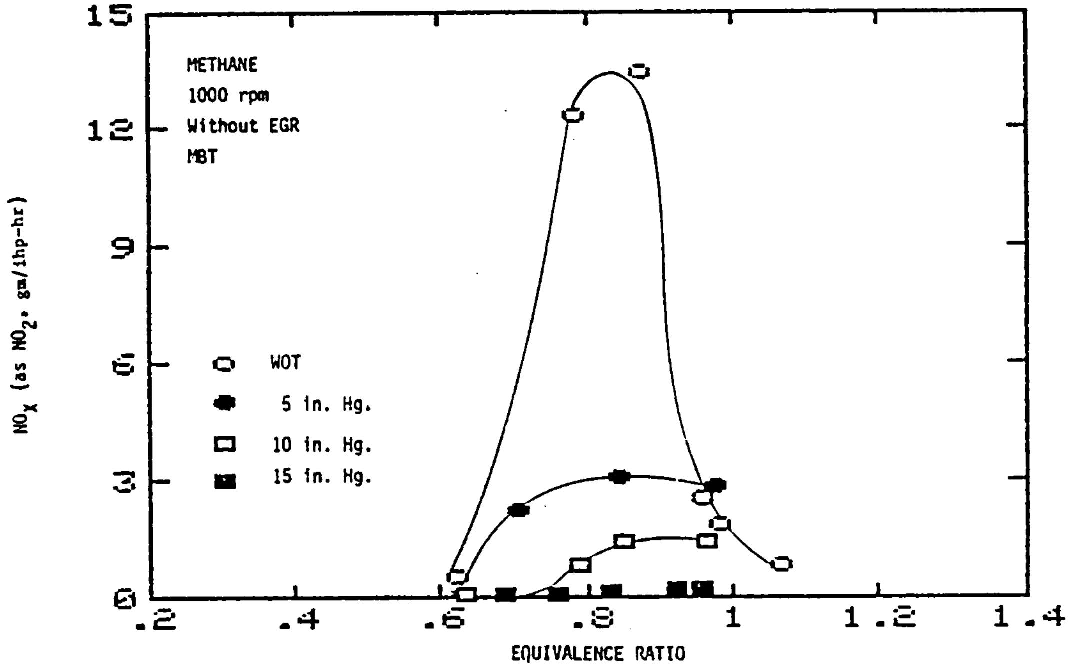

Figure B92: Effect of Equivalence Ratio on Oxides of Nitrogen Emissions at Various Throttle Settings for Methane at 1000 rpm, without EGR.

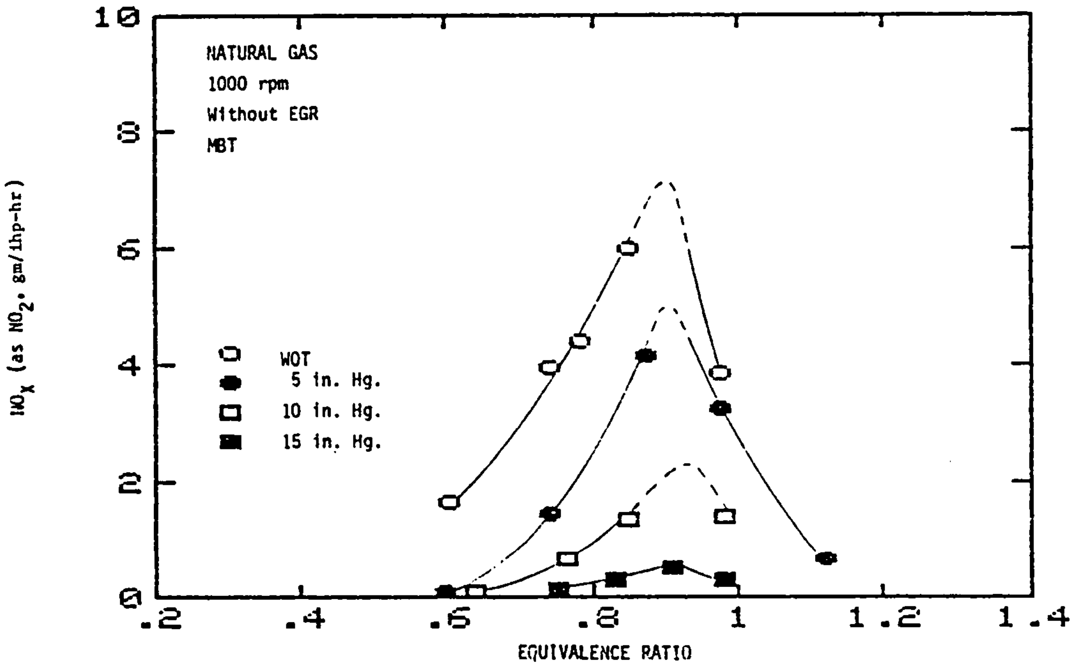

Figure B93: Effect of Equivalence Ratio on Oxides of Nitrogen Emissions at Various Throttle Settings for Natural Gas at 1000 rpm, without EGR.

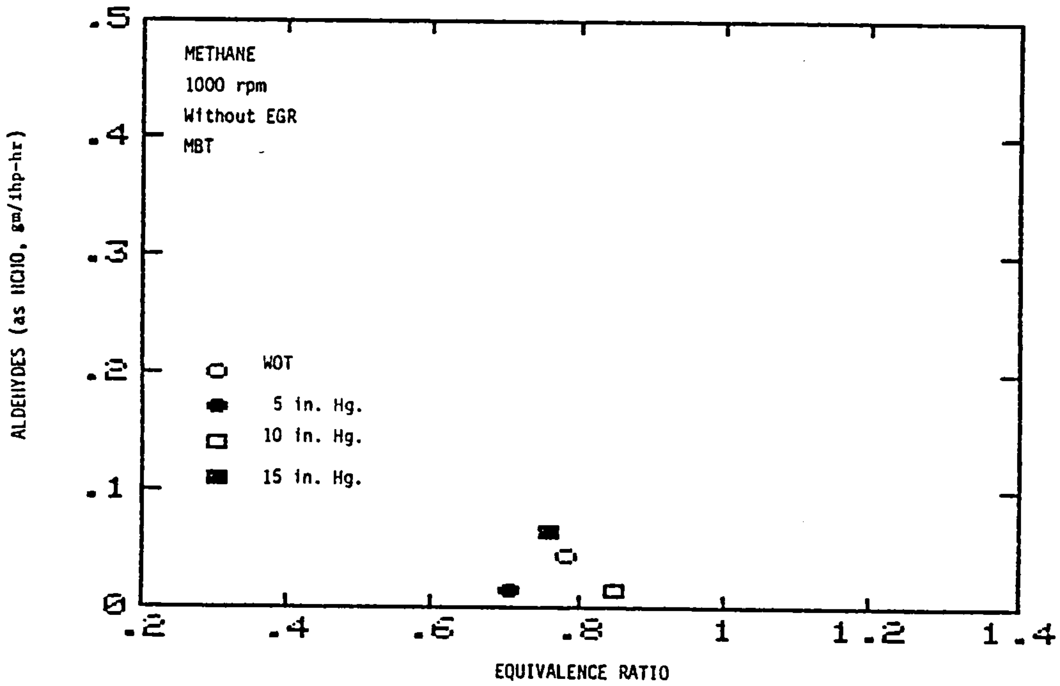

Figure B94: Effect of Equivalence Ratio on Aldehyde Emissions at Various Throttle Settings for Methane at 1000 rpm, without EGR.

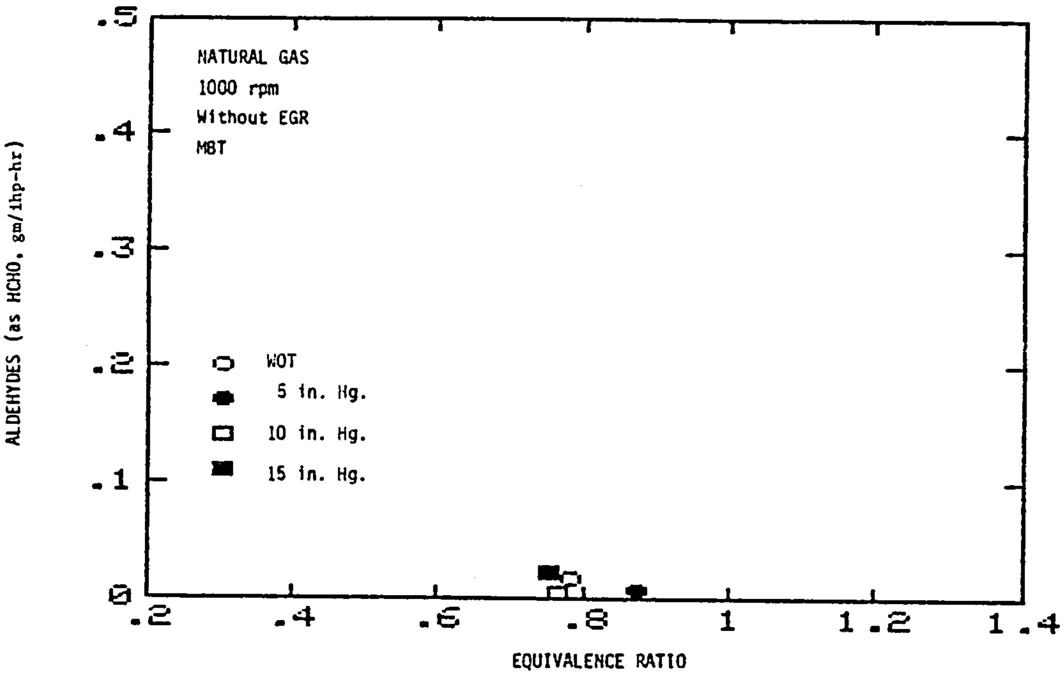

Figure B95: Effect of Equivalence Ratio on Aldehyde Emissions at Various Throttle Settings for Natural Gas at 1000 rpm, without EGR.

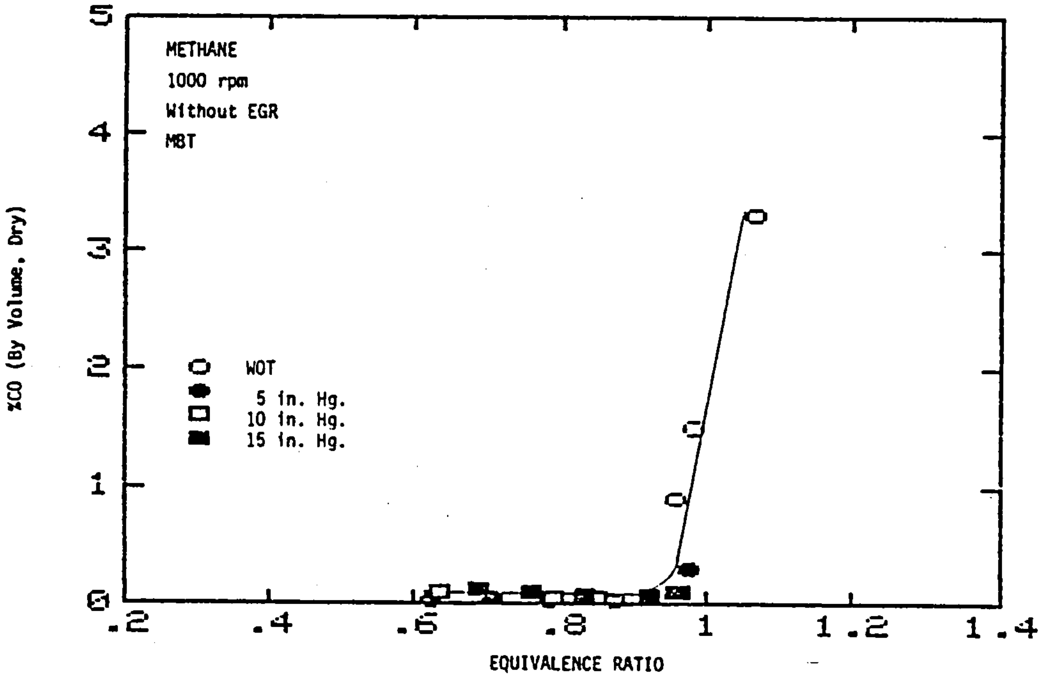

Figure B96: Effect of Equivalence Ratio on Carbon Monoxide Emissions at Various Throttle Settings for Methane at 1000 rpm, without EGR.

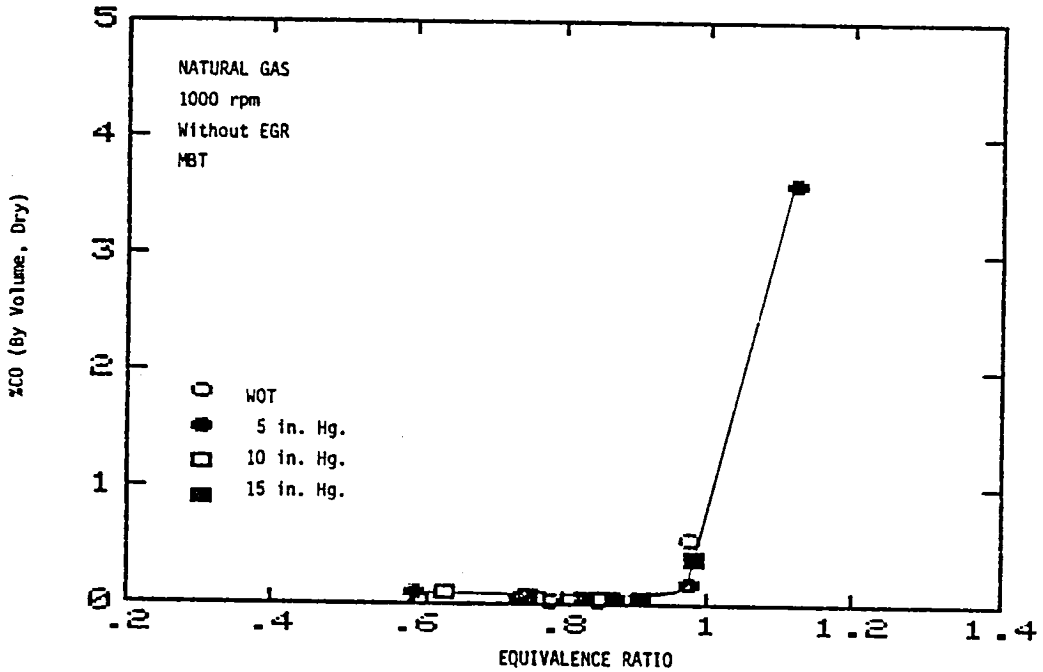

Figure B97: Effect of Equivalence Ratio on Carbon Monoxide Emissions at Various Throttle Settings for Natural Gas at 1000 rpm, without EGR.

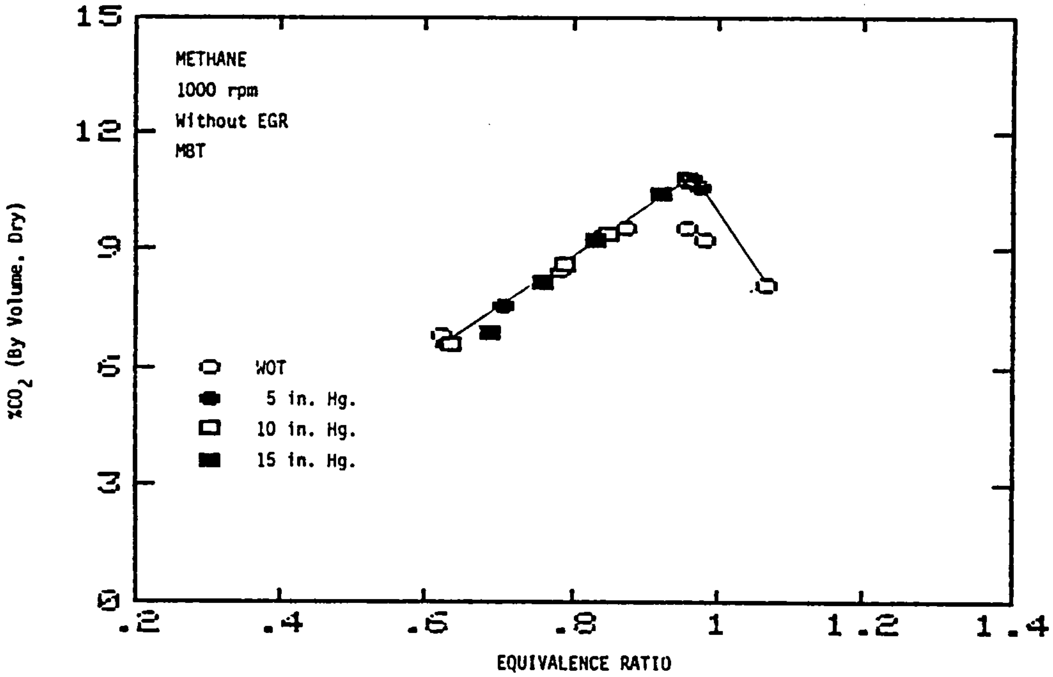

Figure B98: Effect of Equivalence Ratio on Carbon Dioxide Emissions at Various Throttle Settings for Methane at 1000 rpm, without EGR.

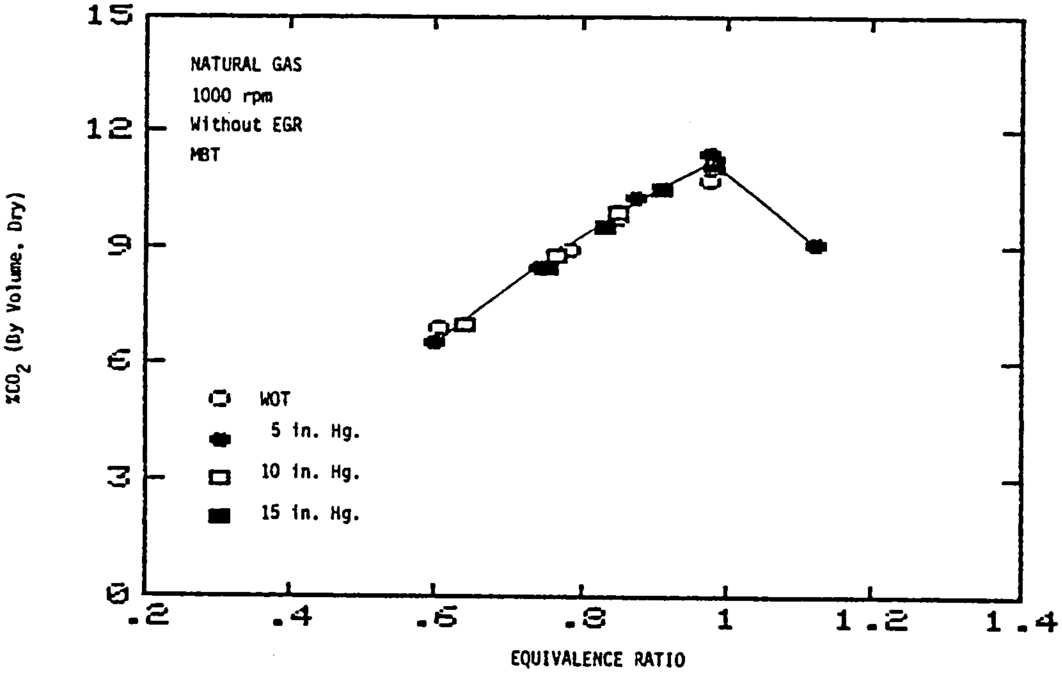

Figure B99: Effect of Equivalence Ratio on Carbon Dioxide Emissions at Various Throttle Settings for Natural Gas at 1000 rpm, without EGR.

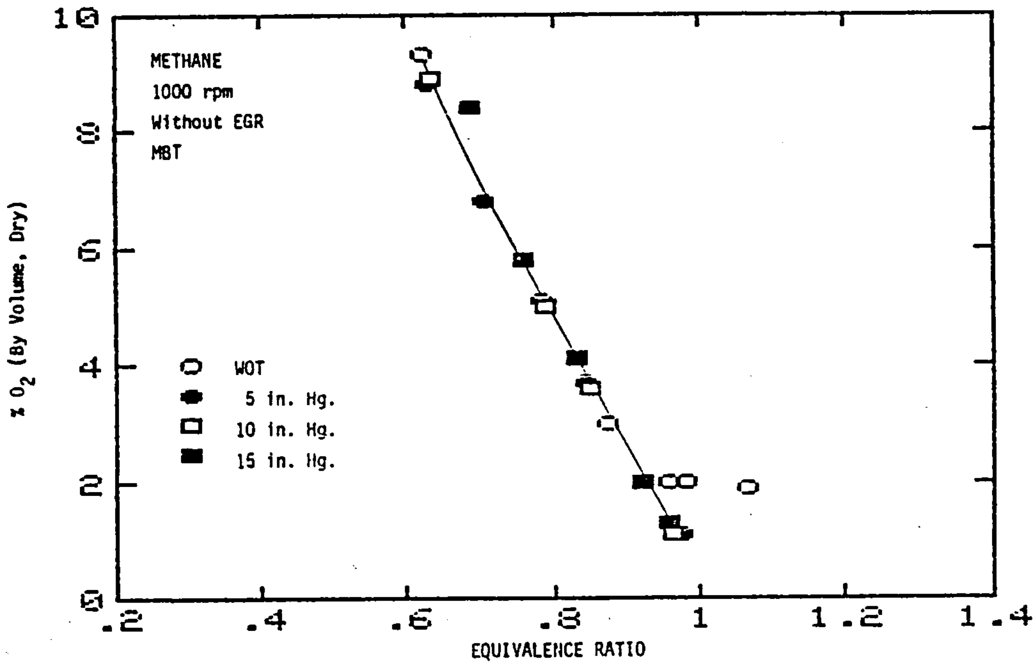

Figure B100: Effect of Equivalence Ratio on Oxygen Emissions at Various Throttle Settings for Methane at 1000 rpm, without EGR.

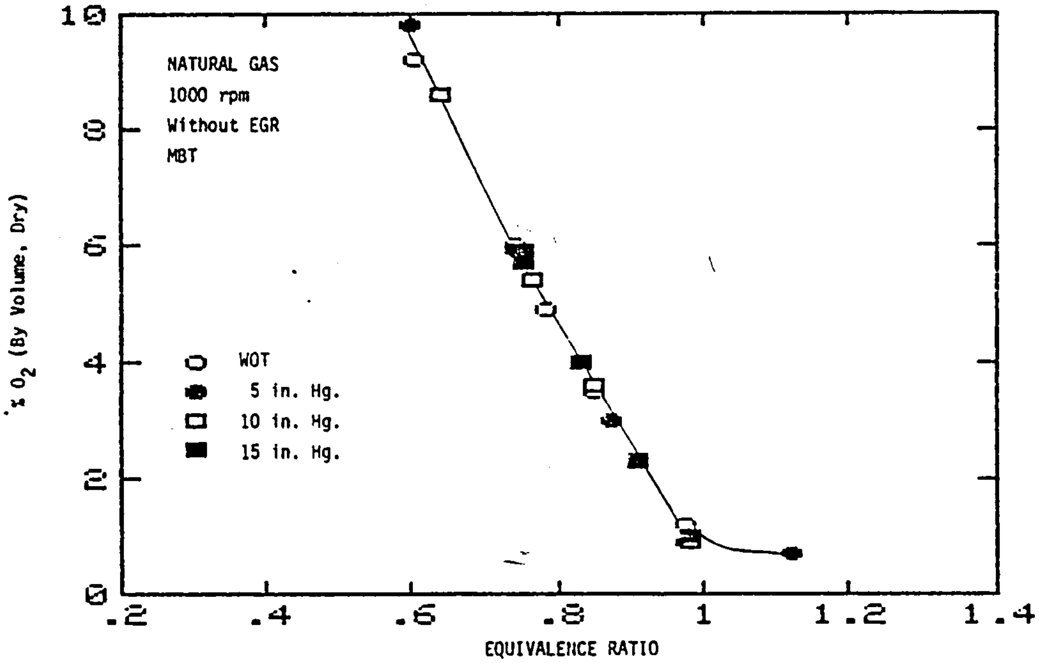

Figure B101: Effect of Equivalence Ratio on Oxygen Emissions at Various Throttle Settings for Natural Gas at 1000 rpm, without EGR.

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16. Abstract

Dissociated and steam reformed methanol were evaluated as automotive engine fuels. Advantages and disadvantages in using methanol in the reformed rather than liquid state are discussed.

Engine dynamometer tests were conducted with a four cylinder, 2.3 liter, spark ignition automotive engine to determine performance and emission characteristics operating on simulated dissociated and steam reformed methanol ($2H_2 + CO$ and $3H_2 + CO_2$ respectively), and liquid methanol. Results are presented for engine performance and emissions as functions of equivalence ratio, at various throttle settings and engine speeds.

Operation on dissociated and steam reformed methanol was characterized by flashback (violent propagation of a flame into the intake manifold) which limited operation to lower power output than was obtainable using liquid methanol. It was concluded that: an automobile could not be operated solely on dissociated or steam reformed methanol over the entire required power range - a supplementary fuel system or power source would be necessary to attain higher powers; the use of reformed methanol, compared to liquid methanol, may result in a small improvement in thermal efficiency in the low power range; dissociated methanol is a better fuel than steam reformed methanol for use in a spark ignition engine; and use of dissociated or steam reformed methanol may result in lower exhaust emissions compared to liquid methanol.

17. Key Words (Suggested by Author(s))
Alcohol fuel; Alternate fuel, hydrogen fuel,
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