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INVESTIGATION OF THE FUEL ECONOMY POTENTIAL ASSOCIATED WITH M85 FUELED VEHICLES

A Thesis

Presented for the

Master of Science

Degree

The University of Tennessee, Knoxville

Shean Patrick Huff

August 1993

This thesis is dedicated to my wife

Kim

who has been extremely supportive and more than understanding

over the past fourteen months, and to my parents

John and Patty Huff

who have always been there when I needed them.

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ABSTRACT

Due to an increasing threat of oil shortages and concerns about air quality, attempts have been made to reduce the United States' dependency on petroleum. Two possible solutions that have been receiving attention are to reduce the fuel consumption of modern automobiles and to use alternative fuels. This study considers both possibilities for a combined solution. To accomplish this, a stock gasoline vehicle was converted to operate on M85, a mixture of 85 percent methanol and 15 percent gasoline by volume, and concepts used to increase the vehicle's fuel economy were implemented in order to improve the conversion.

Under normal circumstances, a simple conversion of an engine, from gasoline to methanol operation, results in an increase in power-related performance. This increase is due to methanol's high latent heat of vaporization and low stoichiometric air/fuel ratio, which lead to increased volumetric efficiency. More complex conversions attempt to exploit methanol's high octane rating, which reduces the tendency for auto-ignition, through the use of turbo/superchargers and/or raising the compression ratio. For this study, it was desired to use these properties to increase the vehicle's fuel economy potential rather than its power-related performance. Therefore, the conversion was made such that the power-related performance of the developmental vehicle matched that of the baseline vehicle.

The properties of methanol mentioned above provide the opportunity to produce the same power-related performance of a gasoline engine from a smaller displacement methanol engine. Using this fact, the strategy selected was that of increasing the fuel economy potential through a reduction in the number of cylinders and thus the overall engine displacement. In order to reduce the number of variables changed and thus allow for a direct comparison of the developmental vehicle to the baseline vehicle, it was decided that the same engine, a Chevrolet 2.8 liter V-6, should be used for both the

baseline and developmental vehicles.

The developmental engine's displacement was reduced to 1.9 liters by modifying two cylinders so that they did not fire, and therefore converting the engine into a V-4 configuration. The two "dead" pistons were left in the engine for balancing purposes. However, their contribution to the engine's overall friction was reduced by removing the piston rings, and their pumping work was reduced by cutting slots in the piston crowns such that the crankcase gases could flow freely around the piston. The developmental engine was then outfitted with a turbocharger in order that the engine's power-related performance would match that of the baseline engine. Other than the modifications described above and those required for the conversion from gasoline to methanol fuel, the developmental vehicle's configuration was the same as that of the baseline.

After implementing the above modifications, the vehicle underwent various tests to determine the success of the conversion. It was found that an approximate increase of 19 percent was experienced in the gasoline equivalent fuel economy over the federal test procedure city and highway emissions test cycles. It was also found that increases of up to 21 percent could be experienced at steady state highway speeds.

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LIST OF SYMBOLS

C	Carbon
CDM	Calibration Display Module
$CH_{1.85}$	Gasoline
CL	Chemiluminescent
CO	Carbon Monoxide
CO_2	Carbon Dioxide
CVT	Continuously Variable Transmission
ECM	Electronic Control Module
EEPROM	Electronically Erasable Programmable Read-Only Memory
EGO	Exhaust Gas Oxygen
EPA	Environmental Protection Agency
FTP	Federal Test Procedure
ft-lb	Foot Pound
HC	Hydrocarbons
in^3	Cubic inch
J	Joule
kPa	Kilopascal
L	Liter
LV8	Scaled Load Parameter
MAF	Mass Air Flow
mph	Miles Per Hour
MTBE	Methyl Tertiary Butyl Ether
M85	85% Methanol and 15% Gasoline by Volume
NDIR	Nondispersive Infrared Analyzer
NO	Nitric Oxide
NO_x	Oxides of Nitrogen
NO_2	Nitrogen Dioxide
P_b	Brake Power
P_f	Friction Power
P_i	Indicated Power
psig	Pounds Per Square Inch, Gage
SAE	Society of Automotive Engineers
WOT	Wide Open Throttle

CHAPTER I
INTRODUCTION

In an attempt to reduce the United States' dependency on petroleum and to reduce atmospheric pollution, the automotive industry is working to develop production vehicles that operate on non-petroleum based fuels. One fuel receiving attention is methanol (1,2).

Methanol has several features which make it an attractive transportation fuel for the future. First, since methanol is a liquid at room temperature, existing pipelines, storage facilities and dispensing stations can be used with only minor modifications. This alleviates the need for costly new systems that would be needed for alternative fuels such as natural gas and electricity (3). Second, methanol can be produced from natural gas, coal, biomass, or municipal solid waste. These resources are in abundant supply in the United States. This fact gives both economic and political advantages by decreasing the United States' dependence on foreign resources. Also, even though natural gas is imported from Canada, reducing the U.S. dependence on the unstable Middle East region, for petroleum, is definitely an advantage. Third, methanol's high latent heat of vaporization, low stoichiometric air/fuel ratio, and high octane rating lead to increased engine volumetric efficiency and a decreased tendency to auto-ignite, which allow increased engine performance without the use of toxic fuel additives. This, along with the fact that methanol produces fewer hydrocarbons and oxides of nitrogen, gives methanol a favorable advantage in a world where environmental issues are becoming extremely important (4). It is for these reasons that the United States Environmental Protection Agency and the California Air Resources Board are viewing methanol as a clean fuel that can contribute to the reduction of atmospheric ozone formation (3).

Table 1 gives a direct comparison of methanol's and gasoline's properties. Methanol's thermodynamic properties generate an increase in usable engine power

Table 1–Comparison of Methanol and Gasoline Properties.

		Methanol	Gasoline
Molecular Weight ‡		32.04	100-105
Specific Gravity ‡		0.796	0.70-0.76
Boiling Temp. (°C) ‡		64	27-204
Vapor Pressure (kPa @ 38°C) ‡		31.7	55-103
Heating Value (MJ/kg @ 25°C) †	Higher	22.7	47.3
	Lower	20.0	44.0
Heat of Vaporization (kJ/kg) †		1103	305
Stoichiometric Air/Fuel Ratio (mass basis)		6.43	14.6
Potential Cooling Effect (°C) [1]*		148	25
Octane Rating ‡	Research	112	91-98
	Motor	91	83 - 90

[1] Assuming that all the fuel vaporizes (which may not occur).

‡ (5)
† (6, p915)
* (7, p74)

production over that of gasoline for the same engine displacement. One cause is that when using methanol, it is possible to obtain higher volumetric efficiencies than with gasoline because of methanol's large cooling effect which increases the density of the incoming fresh fuel/air charge, allowing more mass into each cylinder. Methanol's cooling effect is caused by its high latent heat of vaporization coupled with its low stoichiometric air/fuel ratio. Even though on a mole basis, methanol's heat of vaporization is slightly lower than gasoline's, gasoline has over twice as much mass of air to draw this energy from. Therefore, methanol has a greater cooling effect.

Another property of methanol, which can lead to an increase in power production, is its high octane rating. Methanol's high octane rating permits a higher compression pressure, allowing for turbo/supercharging, which also introduces more mass into each cylinder, and/or allows for higher compression ratios.

In view of the earth's dwindling natural resources, it is often more advantageous to increase the fuel economy potential of a vehicle rather than its power-related performance. However, since there is often a loss of power associated with methods of increasing the fuel economy, it is possible that the performance characteristics of methanol could offset these losses and allow for both acceptable fuel economy and performance.

CHAPTER II
BACKGROUND

On the Verge of an Oil Crisis

In 1993, with the industrial world growing at astounding rates, the global consumption of oil is extremely high, but how long can it last? Between the decreases in domestically produced oil and disruptions in oil supplies from the Middle East, the United States could face an oil shortage in the not too distant future (8).

The transportation sector will by far be hit the hardest. In 1988 transportation accounted for over 63% of all United States petroleum consumption and 23% more than the amount of crude oil produced in the United States (9, p139). Furthermore, between 1973 and 1988, oil consumption in the transportation sector increased 1.8% per year, while oil consumption in the industrial and residential/commercial sectors decreased by 2.3% and 2.5% per year respectively (10, p7).

Within the transportation sector, automobiles and light trucks have by far the largest share of the increasing consumption of oil. Figure 1 shows the comparison of oil consumption, between each mode of transportation, in 1988. It is evident that in the United States the automobile will remain the principle mode of personal transportation (11). Therefore, reducing the oil consumption of the automobile would greatly affect the overall future of the United States' oil consumption. Two ways to decrease the oil consumption of the automobile are to increase the vehicle fuel economy and implement the use of alternative fuels such as methanol.

Fuel Economy

In the past eighteen years, the fuel economy potential of automobiles has been a subject of some concern for both the consumers and the automotive manufactures. Due to increasing fuel prices, the automobile consumer has demanded vehicles with better

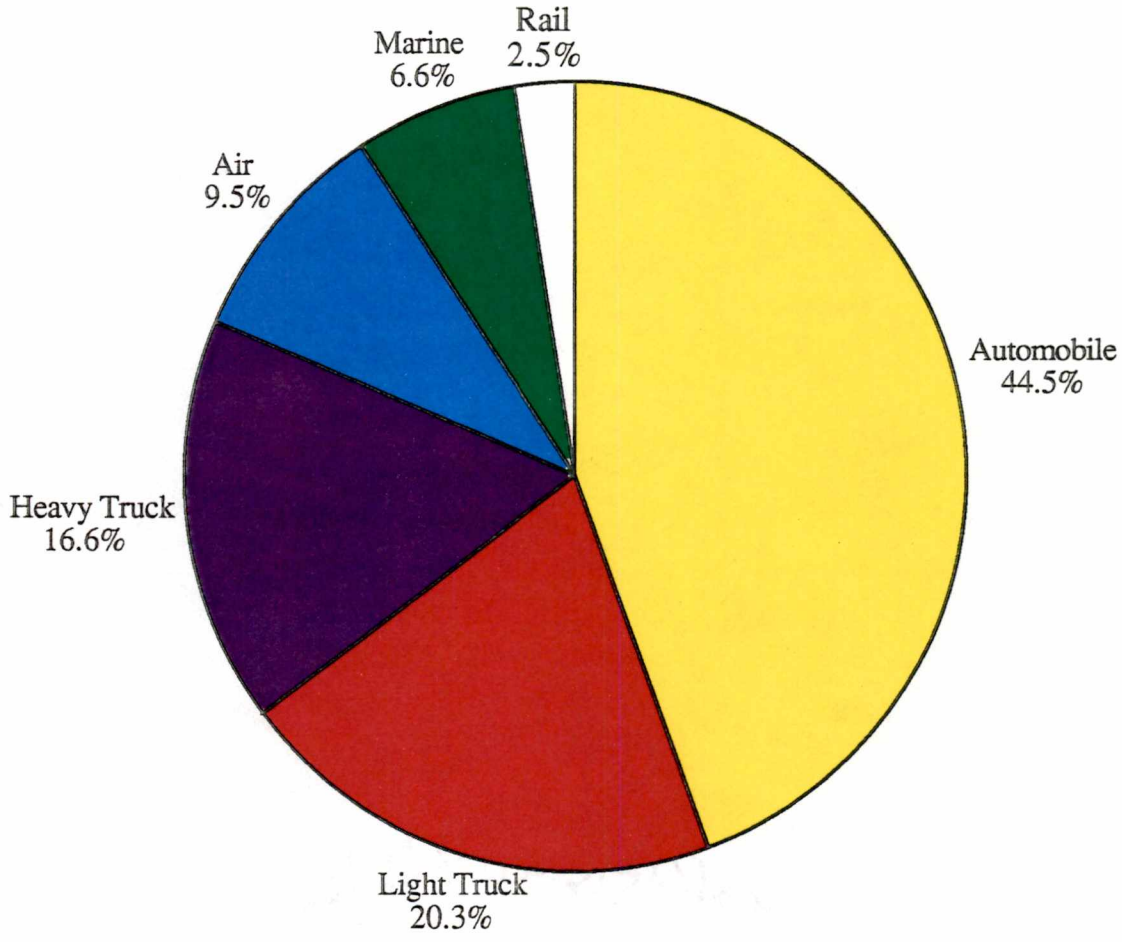


Figure 1—Percentage of Total Transportation Oil Usage within Each Mode in 1988 (9, p140).

fuel economy, and because of the world's decreasing oil reserves, the United States has passed legislation which requires manufactures to improve the fuel economy of their vehicles. In 1975 the US government enacted the Energy Policy and Conservation Act, a mandatory fuel economy statute requiring that automotive manufacturers selling cars in the United States increase the corporate average fuel economy of their new-car fleet to 27.5 miles per gallon in model years 1985 and after (12).

In the years between 1974 and 1984, both imported and domestically-produced cars showed a steady increase in fuel economy. The two major changes responsible for this increase were the reduction in average vehicle weight and the use of smaller displacement engines (12). Between 1974 and 1984, the average engine displacement was reduced 38%, however, improvements in powertrain technology prevented acceleration performance from dropping proportionately. In addition to the reduction in engine displacement, there has been an increase in the use of four and six cylinder engines, with even a few three cylinder engines being produced. Further fuel economy improvements have come through the use of overdrive and lockup transmissions and fuel injectors along with reductions in rolling resistance and aerodynamic drag. Other engine technologies currently used in automobiles for increasing fuel economy include: roller cam followers, general friction reduction, deceleration fuel restriction, compression ratio increase, throttle-body fuel injection, multiport fuel injection, overhead camshaft, four valves per cylinder, variable valve timing, advanced lubricants, turbo/supercharging, and diesel (13, p41).

Despite all the work done on increasing the fuel economy of passenger vehicles, many possibilities have yet to be exploited. According to Simpson (12), areas of new technology which present potential for fuel economy improvements in the future include the following: spark ignition engines with improved thermodynamics and reduced friction, improved and new concept diesel engines, continuously variable ratio

transmissions, increased use of electronic controls, further reductions in weight and aerodynamic drag, improved fuels and lubricants, and improvements in tires, electrical systems, hydraulic systems, and air conditioning.

The possibilities for improvements in the spark ignition engine include improved combustion efficiency, reduced mechanical friction, possible applications of variable valve timing, increased charge density, application of electronic controls, and reduced engine weight. Methods of increasing the efficiency of combustion include the use of lean/fast burn combustion designs, which improve fuel economy without having a negative effect on vehicle emissions. Mechanical friction can be reduced by the use of low friction piston rings, roller tappets and cam followers, reduced crank journal diameter, and reduction of parasitic losses due to the oil and coolant pumps. More extensive use of turbo/supercharging could be used to increase the charge density and volumetric efficiency. With the use of these improvements, smaller engines could be used in order to achieve higher fuel economy without reducing the power output (12, p17).

The use of compression ignition (diesel) engines in passenger cars is receiving renewed attention with improvements in combustion chamber design, friction reduction, and further use of direct injection, electronic controls, and turbocharging. Further increases in diesel engine usage will come with the development of an adiabatic diesel engine, in which the conventional cooling system is all but eliminated and the engine runs at much higher temperatures (12, p18).

Automatic transmissions will be improved using electronic controls and overdrive and lockup features. Other transmission technologies include the continuously variable transmission (CVT) which automatically changes ratio over a wide range in a continuous manner rather than by discrete steps of gears. The operation of the CVT is as described by Simpson (12). "Through the use of sensors and electronic

controls, the CVT will be able to provide the optimum transmission ratio for each particular driving condition. The greatest fuel economy benefits will accrue under light load operating conditions, where the CVT will automatically shift to the equivalent of a super overdrive ratio, greatly reducing engine friction losses. At the same time, CVT's will allow improved acceleration performance because of their wide range of ratios and smooth, stepless acceleration performance capabilities."

Further usage of electronic engine controls will allow the engine to optimize its performance under all driving condition while still keeping engine emissions at acceptable levels. Electronic controls will also be used to control transmission shifting and lockup, integrated speed control, anti-lock brakes, traction control, and suspension leveling devices (12, p18).

In addition, there are unlimited possibilities for research and development of alternative fuels which are in fact better suited for the internal combustion engine than gasoline. With the engine and fuel system optimized for methanol, it is possible to increase fuel economy by at least 15% above that of a gasoline fueled engine (12, p19).

Methanol

In recent years, the United States' growing dependence on foreign oil resources has brought about an interest in methanol as an alternative fuel. Methanol can be produced from both fossil and renewable domestic resources. Methanol, also known as wood alcohol, is a colorless, odorless, toxic liquid which can be used for automotive applications as a feedstock for the production of methyl tertiary butyl ether (MTBE), an octane-enhancing additive for gasoline, as a gasoline substitute, or in a blend with gasoline, most commonly as M85 (85% methanol, 15% gasoline).

Domestic resources from which methanol can be produced include natural gas, coal, residual oil, biomass, and municipal solid waste. The technology for producing methanol from natural gas is already present and requires only efficiency improvements

and scale-up to make methanol an economically viable alternative transportation fuel. Producing methanol from coal is also attractive considering the United States large coal reserve. Since sulfur is removed as a routine part of the process, recent concerns about sulfur oxide emissions from coal combustion can be solved by coal-to-methanol gasification. In addition, biomass represents an environmentally friendly and renewable resource for methanol production.

Methanol is produced through a thermochemical process known as gasification. In this process, biomass or coal is subjected to elevated temperatures and pressures to form a synthesis gas. After being cleaned to remove impurities, the synthesis gas reacts with a catalyst at elevated temperatures and pressures to form methanol. If natural gas is used, no cleanup is required before the catalytic reaction step.

Methanol possesses several properties which are advantageous for use as an automotive fuel. Its octane rating is higher than that of gasoline which helps to reduce engine knock and allows the use of higher engine compression resulting in greater fuel efficiency and reduced carbon dioxide (CO_2) emissions. Methanol's high heat of vaporization produces a peak flame temperature which is lower than that from gasoline. This helps to lower the emissions of oxides of nitrogen (NO_x) and also gives the opportunity for a higher engine output from a given engine volume. Therefore, given a specific engine power, a methanol engine could be both smaller and lighter than its comparable gasoline counterpart. Smaller engines allow for reduced friction, reduced pumping losses, and lighter vehicles, all of which help to reduce fuel consumption.

Unfortunately, two problems presently exist that have hindered the wide spread use of methanol as an automobile fuel. Methanol's energy density is approximately half that of gasoline which reduces the vehicle range for an equal volume of fuel. Also, methanol fueled vehicles are difficult to start at temperatures below 45°F due to low vapor pressure, however, the gasoline portion of M85 helps to a certain degree.

Research is currently being done to solve both the problems associated with low energy density and the cold start problems (14,15).

Friction

During the operation of an internal combustion engine, the combustion gases within the cylinders act on the pistons to produce work, which is often referred to as indicated work. Unfortunately, not all of the work can be transferred to the output shaft. The amount of work produced which is not available to the output shaft is called the friction work and accounts for about ten percent of the indicated work at full load and 100 percent at idle. Maximum usable torque and minimum specific fuel consumption, which is the fuel consumption rate per unit power output, are both directly affected by the losses due to friction work (6, p712).

Three sources within an engine account for the losses associated with friction work. One such source is called pumping work. This is the work required to draw the fresh fuel-air mixture through the intake system and into the cylinder, and to expel the burned gases from the cylinder and out the exhaust system. Included in the pumping work are valve flow work and throttling work.

The second source of friction work is called the accessory work. This is the work required to drive the various engine accessories which include the engine fan, water pump, oil pump, fuel pump, generator or alternator, secondary air pump for emissions control, power-steering pump, and air conditioning components.

The third source of friction work is the work associated with rubbing friction or the work required to overcome the resistance to relative motion of all the moving parts of the engine. Rubbing friction includes the friction between the piston rings, piston skirt, and the cylinder wall. It also includes bearing friction in the wrist pin, crankshaft, and camshaft, and friction in the valve mechanism.

The rubbing friction within an engine is reduced by the use of lubricants.

According to Heywood (6), there are three possible scenarios to consider when dealing with lubricated relative motion. The first possibility is that of boundary or solid friction which occurs when the lubricating film is reduced to a few molecules thick and cannot prevent metal-to-metal contact. This type occurs mainly during starting and stopping, but it is also present during operation at the piston ring and cylinder interface at top and bottom dead center, between heavily loaded parts, and between slow moving parts such as valve stems and rocker arms, and crankshaft timing gears and chains.

The second possible type of lubrication is hydrodynamic or viscous friction which occurs when the lubricant film is sufficiently thick to separate the surfaces completely. This type of lubrication is present between two converging surfaces, moving at relatively high speeds with a limited amount of load. Hydrodynamic lubrication is found in engine bearings, between piston skirts and cylinder liners, and between piston rings and liners for high sliding velocities.

The third type of lubrication occurs when hydrodynamic lubrication breaks down and boundary layer friction begins to come into play. This type of lubrication is called mixed or partial lubrication. The breakdown of hydrodynamic lubrication occurs when the thickness of the fluid film becomes about the same as the height of the surface asperities of the two objects in motion. Transition may be caused by sudden changes in load or speed, and also by mechanical vibrations. Mixed lubrication is present in connecting rod and crankshaft bearings where periodic metal-to-metal contact results from a sudden break in the oil film.

Engine friction is often measured through the use of a direct motoring test in which a dynamometer is used to measure the power required to motor a nonfiring engine. For this type of test, the engine's conditions are kept as close as possible to firing conditions. This is usually accomplished by conducting "grab" motoring tests where the engine is switched rapidly from firing to motored operation. Motoring tests

on a progressively disassembled engine can be used to identify the contributions that each major component of the engine makes to the total friction losses. One problem associated with this type of testing is that, during the test, only the compression pressure, not the firing pressure acts on the piston, piston rings, and bearings, which reduces the rubbing friction. General breakdown tests show the following approximate distribution of rubbing and accessory friction: piston assembly 50%, valve train 25%, crankshaft bearings 10%, accessories 15% (6, p725).

Figure 2 shows the results of a breakdown test on a typical four cylinder spark ignition engine. The results in this figure are reported in friction mean effective pressure (fmep), where fmep is defined as the theoretical constant pressure which, if exerted during each power stroke of the engine, would produce an amount power equal to the engine's friction power (16, p45). Also according to Obert (16), fmep, in pounds per square inch gage (psig), is equal to 150.8 times the friction torque, in foot-pounds (ft-lb), divided by the engine displacement, in cubic inches (in^3), for a four-stroke-cycle engine. By performing a unit conversion it is found that fmep, in gage kilopascals (kPa), is equal to 12.57 times the torque, in joules (J), divided by the engine displacement , in liters (L). Therefore, fmep is proportional to the torque required to overcome the friction divided by the engine's displacement.

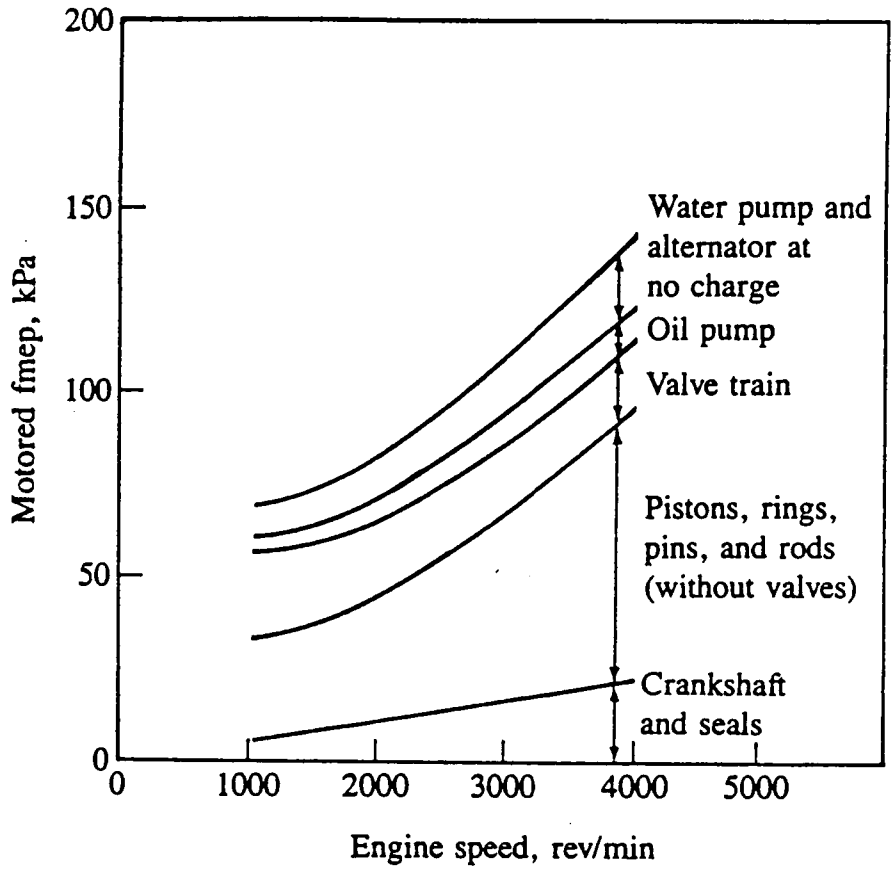


Figure 2—Motored Friction Mean Effective Pressure Versus Engine Speed for a Four-Cylinder Spark-Ignition Engine Breakdown Test (6, p726).

CHAPTER III
THE PROBLEM

Theoretical Frame Work

The fuel economy potential of an engine is related to its mechanical efficiency. An engine's mechanical efficiency is the ratio of the usable (brake) power to the total (indicated) power produced (16, p49). Due to friction losses within an engine, the brake power is less than the indicated power ($P_i = P_b + P_f$). Therefore, a reduction in friction losses will result in increased mechanical efficiency for a given indicated power.

Two contributions to the friction associated with the operation of an internal combustion engine are the mechanical friction and the pumping friction. The pumping friction can be divided into throttling friction and valve pumping friction. The throttling friction is of increasing importance with decreasing load and the valve pumping friction's importance increases with increasing engine speed (6, p728). Figure 3 shows the negative work loop associated with the intake stroke. Comparing Figure 3, parts (a) and (b), reveals that the size of the negative work loop is directly affected by the throttle position. Therefore, to reduce this negative work, it is advantageous to design an engine to run with the throttle as far open as possible for all working conditions, and most importantly at low speed and high load.

Since the friction losses are related to the number of pistons and the engine displacement, a reduction in the number of cylinders should produce a reduction in the overall friction losses by decreasing both the mechanical friction and the pumping friction, since the smaller engine would have to operate at higher manifold pressures to produce the same torque at the same speed.

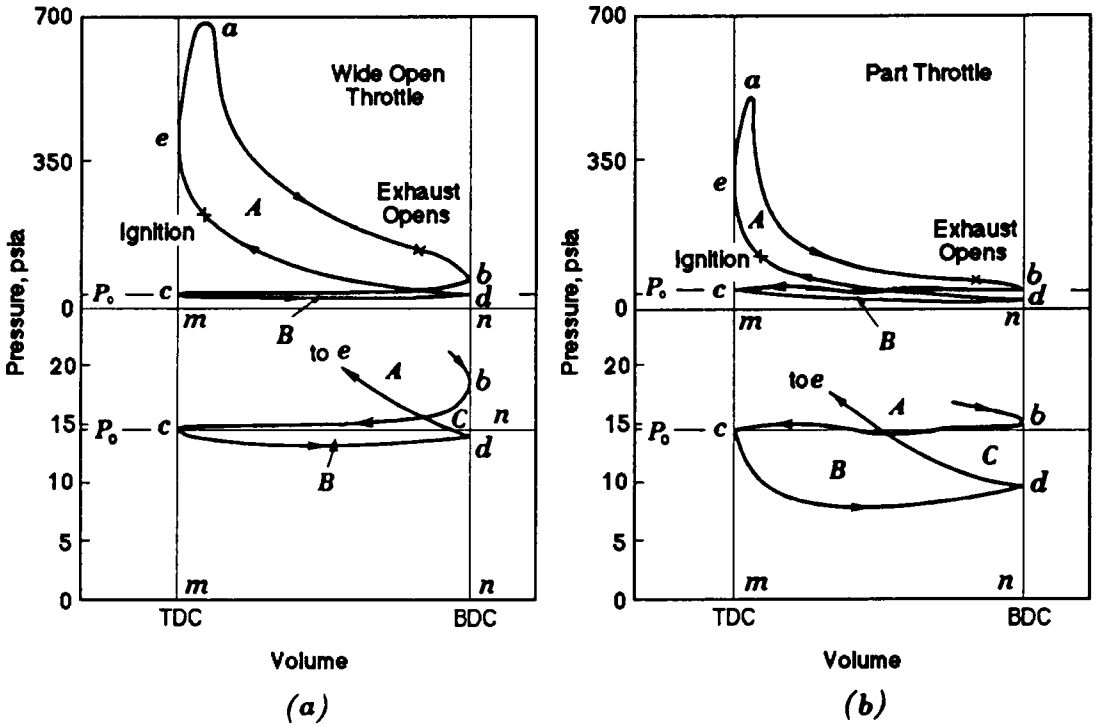


Figure 3—Comparison of Negative Work Loop Associated with Intake Stroke for (a) Wide Open Throttle and (b) Part Throttle (16, p156).

Statement of the Problem

The purpose of this study is to investigate the fuel economy potential of an M85 fueled vehicle that has been modified such that its performance is the same as a gasoline-fueled baseline vehicle. The strategy used to accomplish this involved reducing the engine's friction power by reducing the number of cylinders being used. In order to regain the loss in power associated with disabling the two cylinders and to take advantage of methanol's high octane rating, the engine was turbocharged. It was decided that in order to be able to directly compare the developmental engine to the six cylinder baseline engine, as few variables as possible should be changed. For this reason the same engine was used in the developmental vehicle as in the baseline, with two of the cylinders modified so that they did not fire and that they contributed as little as possible to the over all friction power of the engine.

If the information in Figure 2 is assumed to be representative of a typical 1.9 liter four cylinder engine, an approximation of the friction power can be found. If additional assumptions are made as to the friction reductions associated with each motored friction component in Figure 2 when comparing a 1.9 liter four and a 2.8 liter six cylinder engine, an approximation can also be made for the difference in the total friction between the four and six cylinder engine. Also gains associated with the four cylinder developmental engine over a six cylinder engine can be approximated.

Using information from Figure 1 at an engine speed of 2000 rpm (the engine speed for a vehicle speed of 55 miles per hour), the fmep associated with each component of the motored friction is found to be the following:

Crankshaft and seals	10.5 kPa
Piston assembly	33.7 kPa
Valve train	20.5 kPa
Oil pump	5.79 kPa
Water pump and alternator	11.1 kPa

The friction work is calculated from the fmep using the following equation:

$$T[J] = \frac{\text{fmep}\left[\frac{Pa}{cycle}\right] \cdot D[L]}{2\pi\left[\frac{rad}{rev}\right] \cdot x\left[\frac{rev}{cycle}\right] \cdot 1000\left[\frac{L}{m^3}\right]}$$

where T = friction work of torque
fmep = friction mean effective pressure
D = engine displacement
x = engine revolutions per cycle (revolutions per power stroke)

and the friction power is calculated using the following equation:

$$\text{Power}[hp] = \frac{2\pi\left[\frac{rad}{rev}\right] \cdot T[J] \cdot N\left[\frac{rev}{min}\right]}{60\left[\frac{sec}{min}\right]} \cdot \left(\frac{1.341}{1000}\right)\left[\frac{hp}{W}\right]$$

where Power = friction power
N = engine speed

Using these equations, the friction work and friction power can be calculated for the four cylinder engine as follows:

Component	Friction Work(J)	Friction Power(hp)
Crankshaft and seals	1.57	0.44
Piston assembly	5.01	1.41
Valve train	3.05	0.86
Oil pump	0.86	0.24
Water pump and alternator	1.64	0.46
Total	12.14	3.41

To calculate the increase in friction power associated with a six cylinder engine, assumptions must be made as to the increase in each friction component over that of the four cylinder engine. It is assumed that the friction associated with the crankshaft and seals would increase ten percent, the piston assembly, valve train, and oil pump friction

would increase by 50 percent, and the water pump and alternator friction would increase ten percent.

Again, in order to calculate the friction reduction associated with the four cylinder developmental engine, assumptions must be made for each components friction reduction. There is essentially no change in the friction associated with the crankshaft and seals, oil pump, and the water pump and alternator. Because the rings were removed from two of the cylinders and the rings account for 75 percent of the piston assembly friction (17), the overall reduction in piston assembly friction is 25 percent. Since two sets of valves were disengaged, the reduction in valve train friction is 33 percent. From these assumptions, the six cylinder and four cylinder developmental engines' friction power are as follows:

Component	Six Cylinder Friction Power(hp)	Developmental Four Cylinder Friction Power(hp)
Crankshaft and seals	0.48	0.48
Piston assembly	2.11	1.58
Valve train	1.29	0.86
Oil pump	0.36	0.36
Water pump and alternator	0.46	0.46
Total	4.71	3.75

These results show that there is a 28 percent reduction in motored friction power when going from a six cylinder to a four cylinder engine. The results also show that with the four cylinder developmental engine there is a 20 percent reduction in motored friction power from that of the six cylinder engine. The friction power of the developmental engine was measured directly during the course of this study to be 5.71 hp. Although this value is higher than that calculated above, the percent decrease in friction power from the six cylinder to the developmental four cylinder should be the same. Unfortunately, we were unable to quantify the actual reduction.

CHAPTER IV
EQUIPMENT

Developmental Engine

The engine under examination was a General Motor's V6 with a 2.8 liter displacement, of the type shown in Figure 4. The baseline engine was removed from a stock 1988 Chevrolet Corsica and was in a naturally aspirated, stock configuration.

The development engine was also from a 1988 Chevrolet Corsica, but it was extensively modified in an attempt to increase its fuel economy while keeping its power production the same as the baseline engine.

The first modification used converted the engine to run on only four of the six cylinders. With the engine in the orientation of Figure 4, the cylinders are numbered one-three-five, from left to right along the rear cylinder bank, and two-four-six, from left to right along the front cylinder bank. The two cylinders chosen to be disabled were cylinders numbered two and five. The motivations behind choosing these two cylinders were to modify one from each cylinder bank and to avoid modifying cylinders that were adjacent in the firing order. The firing order of the cylinders is one-two-three-four-five-six with the cycle being repeated every two revolutions of the crankshaft and with 120 degrees of rotation between the firing of successive cylinders. Since cylinders two and five would then be firing 360 degrees apart, it was anticipated that choosing these two cylinders would reduce the consequences of operating with uneven firing.

Several modifications were performed in order to improve the engine's operation. The fuel flow to the two disabled cylinders was cut off by simply disconnecting the electrical connection to their individual fuel injectors. Also, to avoid the ignition of any combustible mixture that might make its way into the cylinder, the spark plug to each cylinder was disconnected. However, since the ignition system is of

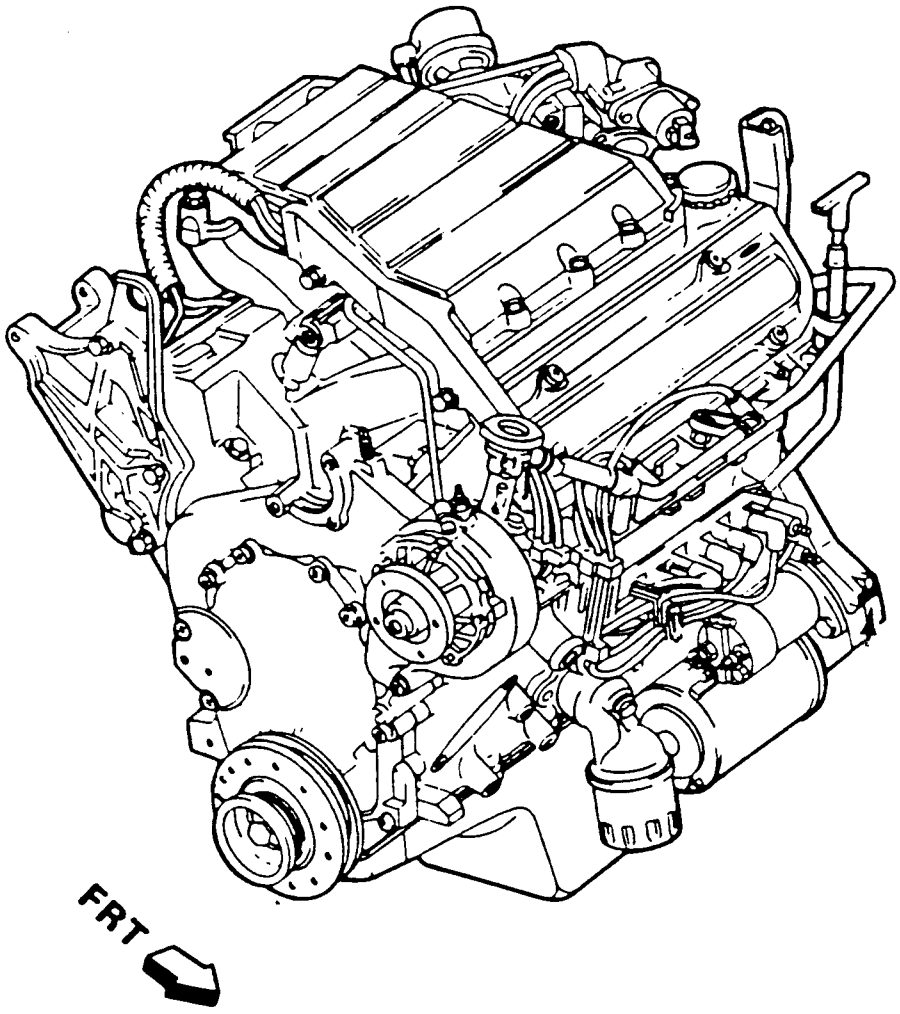


Figure 4—2.8 Liter Displacement General Motor's V6 Engine.

the "Direct Ignition System" design with a "waste spark," dummy spark plugs were provided to the spark plug wires of the disabled cylinders in order to avoid the need to modify the engine's ignition scheme.

The valves for the two disabled cylinders were kept shut to prevent fuel/air or exhaust mixtures from entering the cylinders and to reduce the engine's pumping work. Since oil is used to lubricate the cam followers, the cam followers could not be removed completely, because this would cause a loss of oil pressure. Therefore the cam followers were replaced with inserts that did not contact the cam shaft. Coil springs were used in the rocker arm assemblies, between the rocker arms and the rocker arm retaining nuts, to keep pressure on the push rods and cam follower inserts while accommodating thermal expansion of the components. These modifications are shown in Figures 5, 6, and 7.

To maintain engine balance the pistons in the disabled cylinders were left in the engine, however, they were modified in order to reduce the required effort from the engine. The rubbing friction was reduced by removing the piston rings and slots were machined in the crowns of the two pistons to allow the gases in the crankcase to flow freely around the piston in order to reduce the pumping work associated with moving the pistons. To make up for the material lost from the slot and the rings, a weighted insert was used in the piston wrist pins. The modified pistons are shown in Figures 8 and 9.

Finally, to regain the loss in wide-open-throttle performance due to the two disabled cylinders, the engine was fitted with a turbocharger. The turbocharger used was produced by Garrett Turbochargers for use with a 1.9 liter engine. This turbocharger used water cooled bearings and a vacuum actuated waste gate. Figure 10 shows the engine compartment of the developmental vehicle with the V-4 methanol engine.

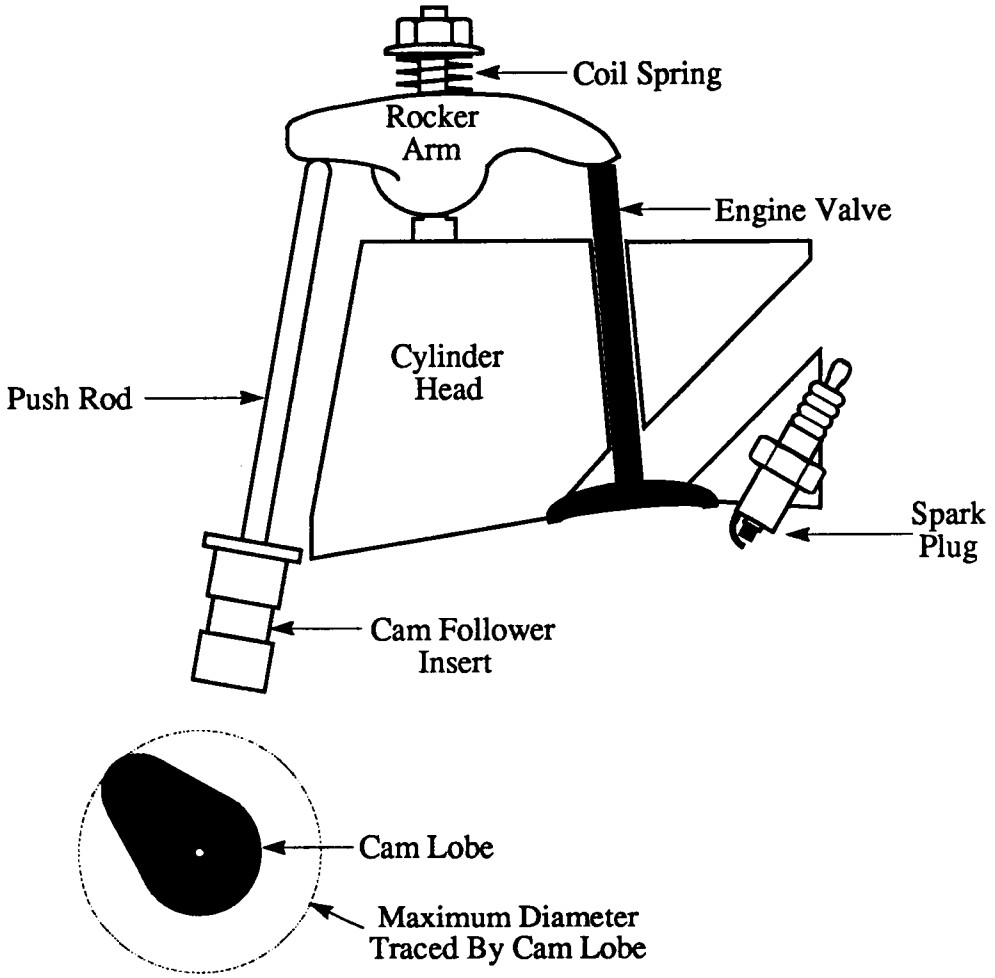


Figure 5—Sketch of Modifications to the Rocker Arm Assembly.



Figure 6—Insert Used to Replace Cam Followers for Nonfiring Pistons.

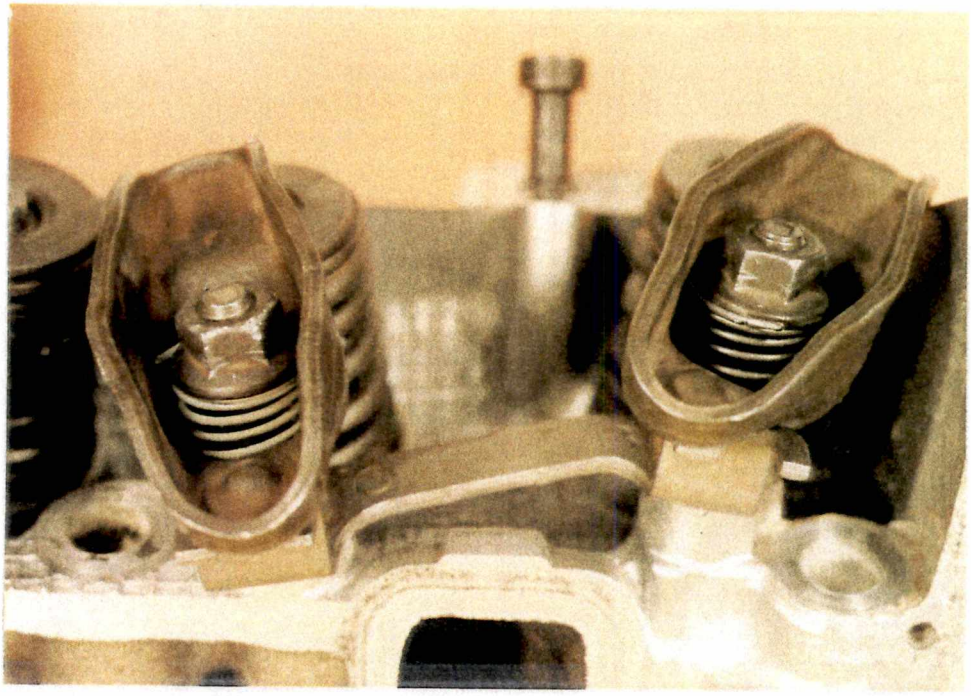


Figure 7—Coil Springs in Rocker Arm Assemblies.

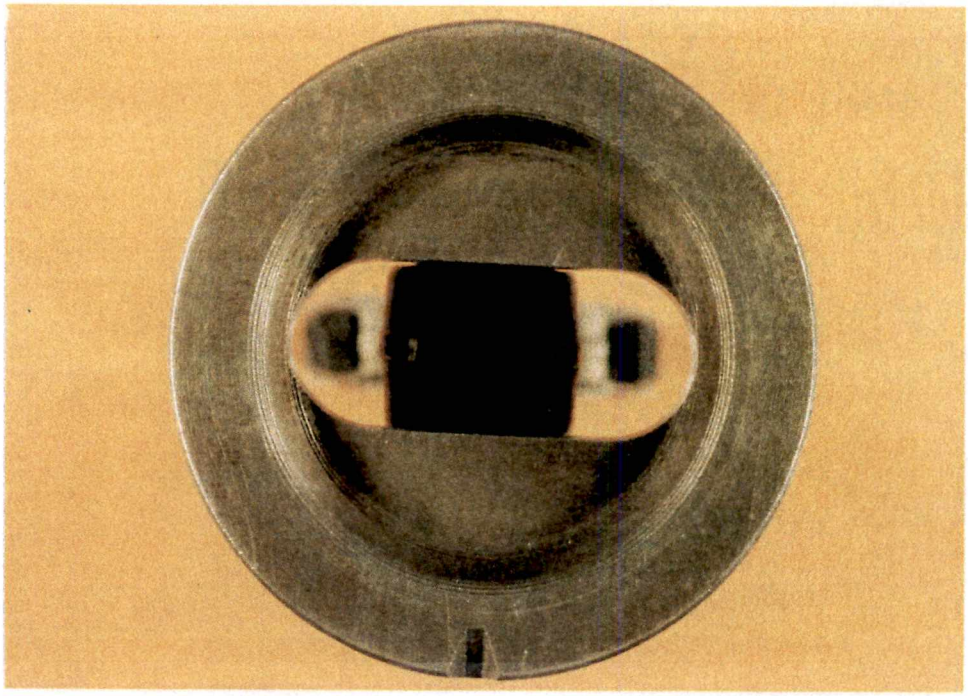


Figure 8—Machined Slot in the Crown of a Nonfiring Piston.

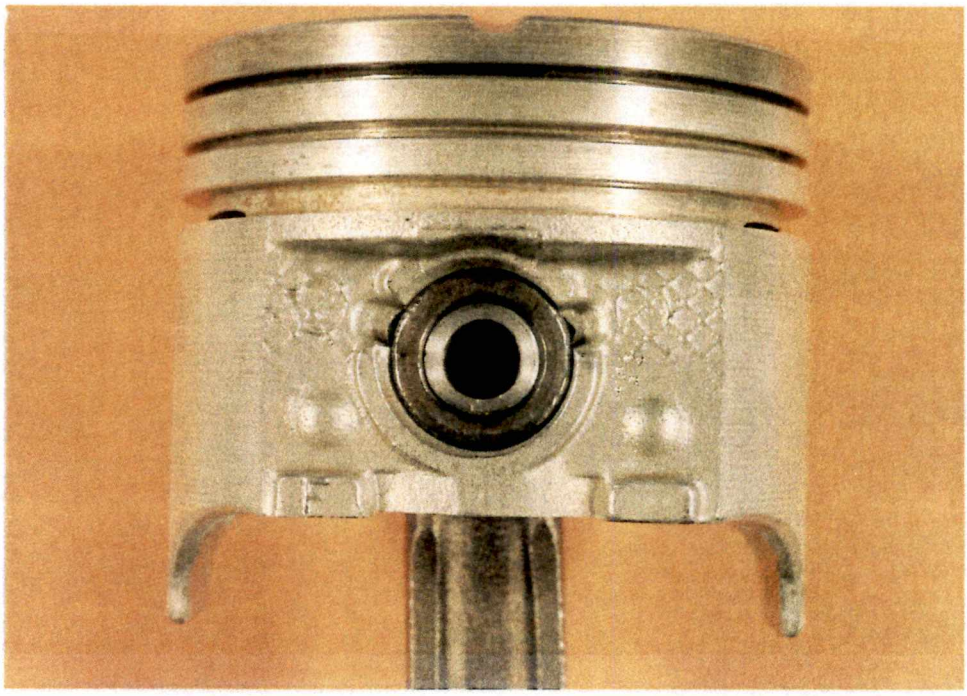


Figure 9—Weighted Wrist Pin in a Nonfiring Piston.

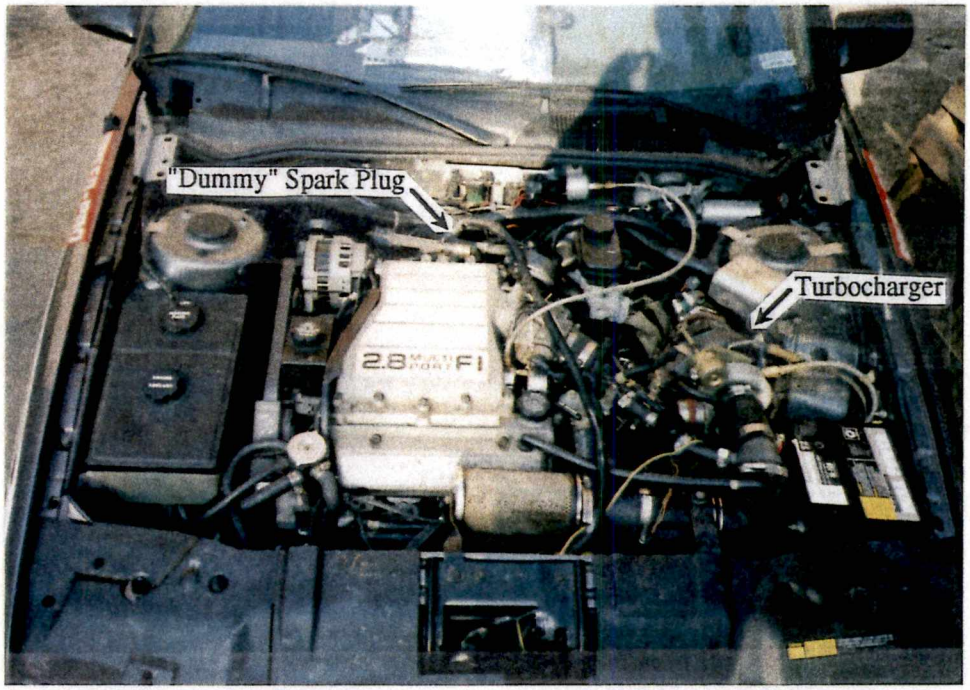


Figure 10—Engine Compartment of Developmental Vehicle.

Dynamometer Testing

A Calibration/Display Module (CDM) was used to modify the engine's Electronic Control Module (ECM) parameters. The CDM's functions were to collect and display data from the engine's ECM and allow modifications to designated regions of the ECM's memory. Before the CDM could be used to modify the ECM's memory, an Electronically Erasable Programmable Read-Only Memory (EEPROM) chip had to be incorporated into the ECM. General Motors Corporation provided a calibration guide which identified specific locations in the ECM's memory where parameters could be changed. Also, the CDM let the investigator keep track of whether or not the engine was in closed loop and if it was toggling between a rich and a lean fuel mixture (indicating under normal conditions that the engine was operating at nominally stoichiometric conditions).

A Duragage pressure/vacuum gage was used to monitor the engine's intake manifold pressure. A psychrometer was used to measure the wet and dry bulb temperatures, and a mercury barometer was used to measure the barometric pressure.

The engine's output was measured using a General Electric Type TLC-3544 D.C. engine dynamometer. This dynamometer operated either as a motor or generator depending on the desired output. When operating as a motor, the dynamometer drives the test engine, and when operating as a generator, the dynamometer absorbs the power output from the engine. The D.C. power produced by the dynamometer during the generator mode, was inverted to A.C. current and fed back into the power grid.

Test Vehicle

The vehicle used for all testing was a 1988 Chevrolet Corsica four door sedan equipped with a Muncie five-speed manual transaxle. In an attempt to reduce the carbon monoxide (CO), hydrocarbons (HC), and aldehydes during the engine warm-up period,

two modifications were made to the exhaust system. These modifications were developed when the vehicle was entered in the 1990 Methanol Challenge competition (18). In addition to the stock General Motor's catalytic converter, a smaller converter was added. This converter was a proprietary catalyst made specially for the University of Tennessee by the Engelhard Corporation. Also, in an attempt to heat up the catalytic converter as quickly as possible, a Warm-up Air Injection (WAI) system was installed. The WAI system used a pulse air valve which used the pressure pulses in the exhaust stream the induce an air flow. This system was activated 20 seconds after starting the engine and deactivated 100 seconds later. Modifications to the exhaust system are illustrated in Figure 11. In addition to the engine and exhaust system modifications, several changes were made to the fuel delivery system in order to make it compatible with methanol. These changes included a stainless steel fuel tank, special AC methanol fuel injectors, and a methanol compatible fuel pump.

Acceleration Test

The vehicle acceleration tests were performed on a 500 foot dragway using two timing schemes. A Vericom VC-200 performance computer, shown in Figure 12, was one timing scheme used. This commercially produced device used an accelerometer to measure the vehicles acceleration, speed, and distance traveled and included a timer which allowed preselected distances to be timed.

The second timing scheme, shown in Figure 13, involved the use of a long distance diffuse-reflective photoelectric sensor, produced by Cutler-Hammer, and a Penta P6000 programmable timer, produced by Acculex. The photoelectric sensor emits and detects an infrared beam which can be reflected off an object. The photoelectric sensor and the programmable timer were connected such that a signal was sent to the timer when the sensor detected an object. The timer was then programmed to start when receiving the first signal and to stop when receiving the second signal. Two

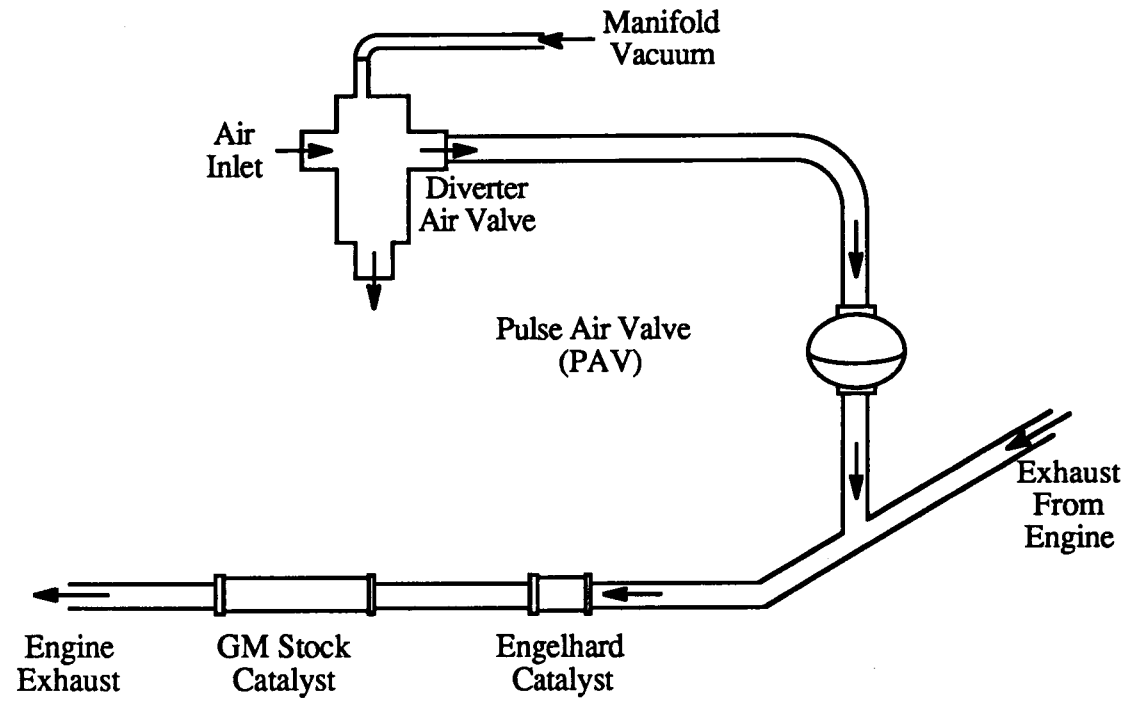


Figure 11—Sketch of Warm-Up Air Injection System and Engelhard Catalyst Used in Exhaust System.

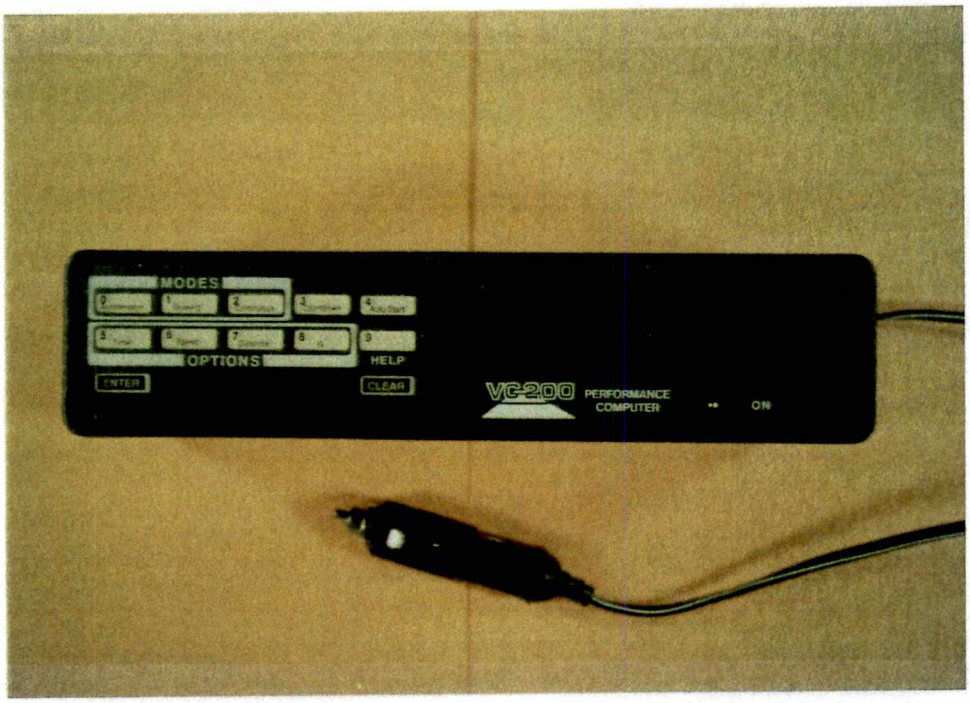


Figure 12—Vericom VC200 Performance Computer.

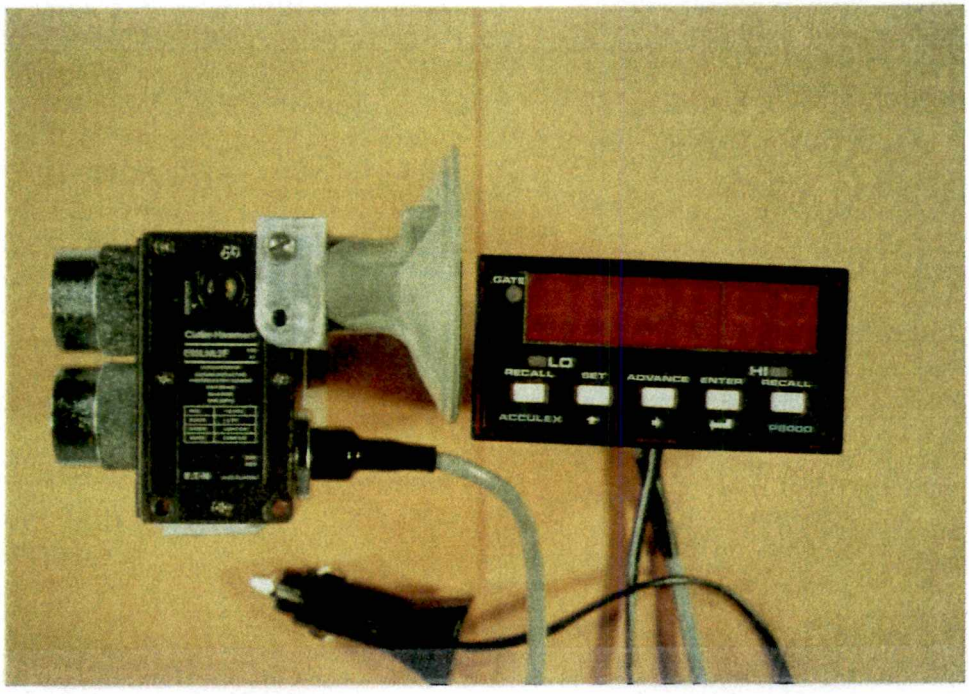


Figure 13—Infrared Timer.

objects were then placed next to the dragway, one at the starting line to trigger the timer to start, and one at the 500 feet mark to trigger the timer to stop. After confidence was developed in the VC-200, it was used to measure 0-60 mph acceleration times.

Emissions Test

The emissions tests were performed on the test vehicle at the EPA Environmental Research Center Annex, Research Triangle Park, North Carolina. These facilities are shown schematically in Figure 14. A chassis dynamometer was used to simulate road load conditions and was enclosed in a temperature controlled test chamber.

Fuel Economy Test

The location for the preliminary fuel economy testing was a 30 mile stretch on Interstate 75 between Knoxville and Chattanooga. The fueling station used was a level parking lot in the vicinity of the start of the test course. The amount of fuel used during the test was measured using a 1000 milliliter graduated cylinder. A Labeco Tracktest Fifth Wheel was used to measure the vehicle speed and the distance traveled during the test and is shown in Figure 15.

The final fuel economy test was performed on the test track at The Transportation Research Center, East Liberty, Ohio. As in the preliminary fuel economy test, the amount of fuel consumed during the test was measured using a 1000 milliliter graduated cylinder. Also, the same Tracktest Fifth Wheel was used in the test to measure the vehicle speed and the distance traveled.

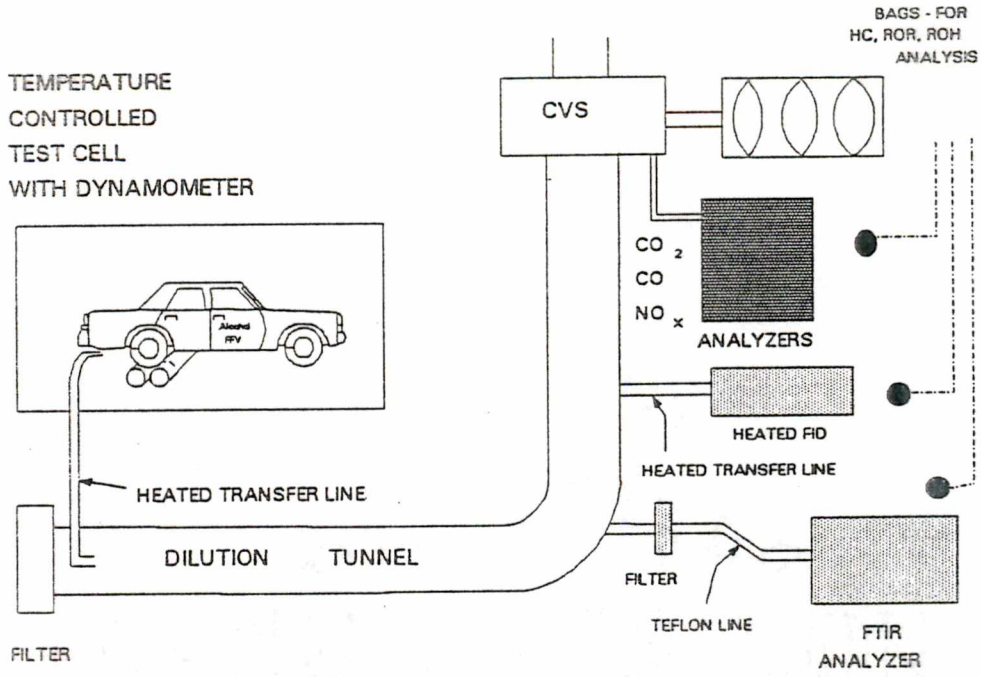


Figure 14—Schematic of EPA Vehicle Tailpipe Emissions Testing Facilities (19).



Figure 15—Tracktest Fifth Wheel.

CHAPTER V
PROCEDURE

Dynamometer Testing

Before the dynamometer testing was begun, the engine's computer control system had to be slightly modified for four cylinder operation. The engine's control system uses a mass air flow (MAF) sensor which generates a scaled load parameter (LV8) which is proportional to the mass of air inducted into each cylinder per intake stroke. The LV8 parameter is then used for the calculations of basic spark timing and fuel injection pulse widths.

With the engine operating on only four cylinders, and the engine's control system being unaltered, the engine's ECM assumes that the air flow is being shared by six cylinders. To solve this problem the LV8 parameter needed to be multiplied by 1.5 to make the six cylinder look-up tables valid for the four cylinder engine. Fortunately, there is an accessible LV8 scaling multiplier available in the ECM's EEPROM which was appropriately adjusted.

During the dynamometer test, the engine's torque output was measured at various engine speeds with the throttle in the wide open position. Before each test, the dynamometer's torque gage was calibrated. This involved first recording the value indicated by the torque gage and using that value to be the zero reading. Then a 150 foot-pound torque was put on the dynamometer by placing a 50 pound weight on a three foot moment arm. The torque gage was then adjusted such that the span, or difference in the indicated torque and the zero value, was 150 foot-pounds. After the dynamometer was calibrated, the test engine was checked again to ensure its readiness. The engine oil and coolant were checked, all foreign objects were removed from the engine's vicinity, and the exhaust fan was turned on.

With the engine ready and the dynamometer calibrated, the test was begun. At the beginning of the test, measurements were taken for the barometric pressure, and the wet and dry bulb temperatures.

The dynamometer was put in the speed mode, which allowed the operator to adjust the speed from zero to 5000 RPM using a variable potentiometer. With the transmission that connected the engine to the dynamometer set at a one to one gear ratio, the dynamometer was turned on at a low speed to check that the transmission was indeed in gear and the engine was rotating in the proper direction. After adjusting the speed to 1500 RPM, the ignition switch for the engine was turned on. The throttle was then adjusted to approximately three percent of wide open throttle (WOT). At this point, the engine was allowed to warm up such that it would enter the feedback control loop for the fuel injection system. When in this loop, the engine used the output of the exhaust gas oxygen (EGO) sensor to keep the air/fuel mixture close to the stoichiometric value.

Once the engine had warmed up to its proper operating temperature, the data collection was begun. Measurements were taken for the torque output of the engine at 400 RPM increments between 1200 and 4800 RPM. For each data point, the dynamometer was adjusted to the desired speed while the engine's throttle was kept between three and ten percent of WOT. With the speed set, the engine's throttle was taken to 100 percent of WOT and the system was allowed to stabilize. Once the torque output stabilized, a reading was taken and also the intake manifold pressure was measured. Then the throttle was immediately set back to between three and ten percent of WOT. The dynamometer was then adjusted to the next speed and the process was repeated. The engine's coolant temperature was closely monitored to avoid overheating. If the temperature became too high, the speed was lowered and the engine allowed to cool. After all the data points were taken, the engine and dynamometer were shut down

and the dynamometer's calibration was checked again to verify that it had not shifted while running the test.

After generating the torque-speed data for the developmental engine, a motoring test was run to determine the friction power of the engine. This was done by first warming up the engine to operating temperature and then cutting off the power to the ignition system. This was done across the same speed range of the engine as the engine torque test.

Acceleration Test

This test was performed in order to find what capability the developmental engine had to accelerate the test vehicle. First, a 500 foot section of the dragway was measured and marked for the test run. Targets for the infrared timer were placed at each end of the 500 foot distance. The photoelectric sensor was set up to deliver a twelve volt signal when it detected the beam being reflected back from an object, and the trigger for the programmable timer was set for a negative slope. This means that the timer was triggered when the sensor's output changed from twelve volts to zero volts, indicating the trailing edge of a detected object. For this reason, the trailing edge of the two targets were placed exactly at the start and finish lines such that the timer could measure the time for the vehicle to travel 500 feet.

By adjusting the vehicle's position until the infrared timer just detected the edge of the starting target, the vehicle was placed exactly at the starting line. After resetting the infrared timer, the VC-200 was configured to measure the elapsed time for a 500 foot run, and it was set to start automatically when it detected the first movement of the vehicle. With a second person holding the test vehicle to insure that it did not drift from the starting position, the driver prepared for the acceleration run.

For each run, the driver attempted to attain the maximum possible acceleration by revving the engine to high RPM's and quickly releasing the clutch, without causing

excessive drive wheel spin. Gear changes were made at approximately 5000 RPM. Several test runs were made in order to find the best possible time.

Emissions Test

Prior to conducting the emissions test, the vehicle was allowed to soak at 70°F overnight to assure a uniform temperature of the engine and exhaust system. Immediately before the test, the dynamometer rollers were run for a short period as a warm up procedure. The vehicle was then placed on the chassis dynamometer and chained down.

During the emissions test the WAI system was activated at 20 seconds after the vehicle was started and remained active until it was turned off 100 seconds later. Exhaust gases from the test vehicle were directed through a flexible stainless steel line, which was enclosed within a heat jacket to prevent temperatures inside from falling below 235°F, to a dilution tunnel. The raw exhaust gases were thoroughly mixed with filtered room air within the dilution tunnel. A constant volumetric flow rate, of about 650 cubic feet per minute, was maintained through the system. Gas samples of diluted exhaust were then sampled from the tunnel.

Total hydrocarbons were measured using a real-time, heated flame ionization detector (FID). This device detects approximately 70 percent of the unburned methanol in the exhaust. The concentrations of CO and CO_2 were determined through the use of nondispersive infrared analyzers (NDIR). The concentration of nitric oxide (NO) was found directly using a photo chemiluminescent (CL) gas analyzer to measuring the light emissions given off when NO reacted with ozone in the reaction chamber. Nitrogen dioxide (NO_2) was measured by first converting it into NO before introducing it into the reaction chamber, therefore measuring the sum ($NO+NO_2$), which is commonly referred to as NO_x (19). The speed versus time plots for the federal test procedure (FTP) city and highway driving cycles are shown in Figures 16 and 17 respectively.

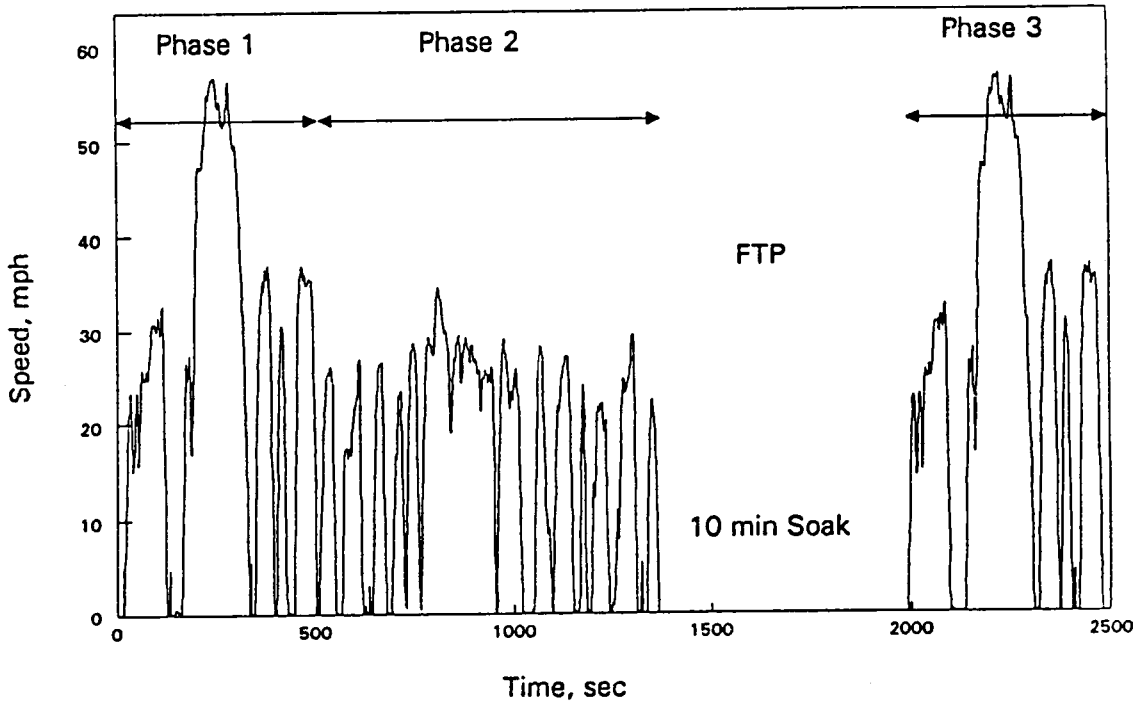


Figure 16—FTP City Driving Cycle.

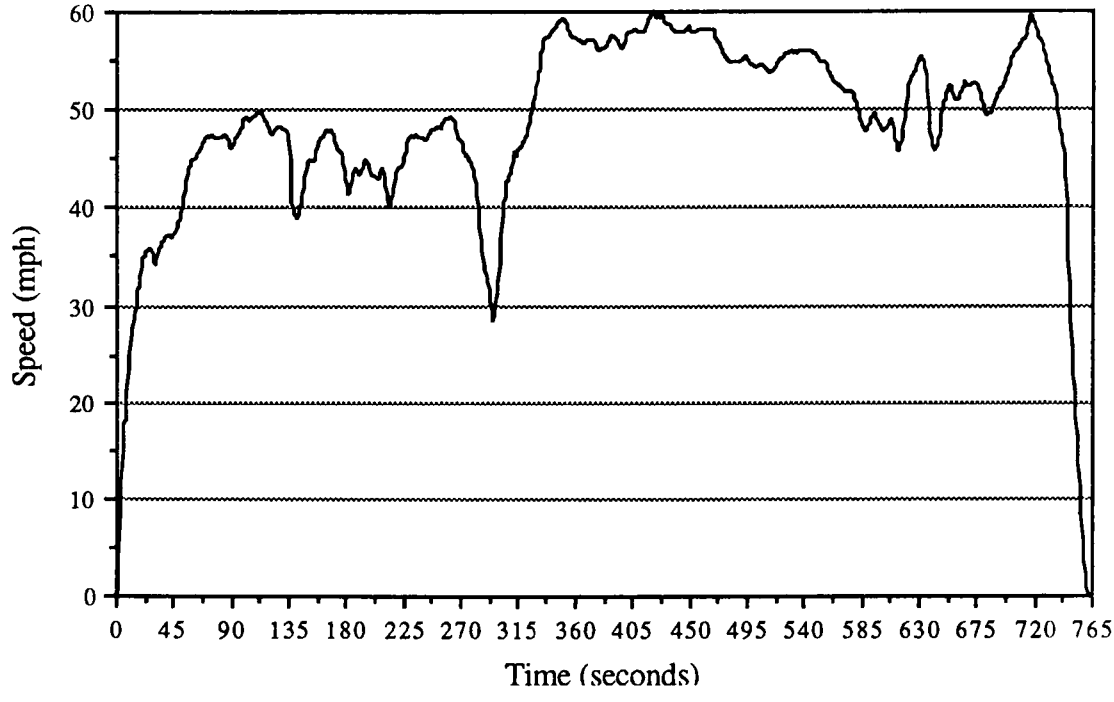


Figure 17—FTP Highway Driving Cycle.

Fuel Economy Calculations for Emission Tests

The following is compiled from information obtained from the reference (20). Calculating the average fuel economy of a vehicle over an EPA emissions cycle is accomplished by measuring the grams of carbon per mile in the exhaust, in the forms of hydrocarbons (HC), CO, and CO_2, and back calculating knowing the amount of carbon (C) contained in a gallon of fuel. The measured HC is assumed to be gasoline which, according to the EPA, is represented as $CH_{1.85}$ (20). The amount of carbon contained in the exhaust in the forms of aldehydes and methanol not measured by the FID is assumed to be negligible. The first step is to find the total mass of a gallon of fuel. The fuel being considered is M85, which is defined as 85% methanol and 15% gasoline by volume. Therefore, the mass per gallon of M85 is calculated as follows:

density of gasoline = 2798 gms/gal
density methanol = 2989 gms/gal

$$\frac{\text{mass of gasoline}}{\text{gallon of M85}} = 0.15 \text{ gallons of gasoline} \bullet 2798 \text{ gm/gal} = 419.7 \text{ gms}$$

$$\frac{\text{mass of methanol}}{\text{gallon of M85}} = 0.85 \text{ gallons of methanol} \bullet 2989 \text{ gm/gal} = 2540.7 \text{ gms}$$

density of M85 = 2960.4 gms/gal of M85

$$\text{mass fraction of gasoline in M85} = \frac{419.7 \text{ gms gasoline}}{2960.4 \text{ gms M85}} = 0.1418$$

$$\text{mass fraction of methanol in M85} = \frac{2540.7 \text{ gms methanol}}{2960.4 \text{ gms M85}} = 0.8582$$

The next step is to calculate the gram fraction of carbon for the fuel and for each of the three measured exhaust constituents.

M85

$$0.1418 \bullet \frac{12.011 \text{ gms C}}{13.876 \text{ gms gasoline}} = 0.1227 \frac{\text{gms C}}{\text{gm M85}} \text{ from gasoline}$$

$$0.8582 \bullet \frac{12.011 \text{ gms C}}{32.043 \text{ gms methanol}} = 0.3217 \frac{\text{gms C}}{\text{gm M85}} \text{ from methanol}$$

mass fraction of carbon in M85 = 0.444 gm C/gm M85

HC

$$HC = \frac{12.011 \text{ gms C}}{13.876 \text{ gms HC}} = 0.866 \text{ gm C/gm HC}$$

CO

$$CO = \frac{12.011 \text{ gms C}}{28.010 \text{ gms CO}} = 0.429 \text{ gm C/gm CO}$$

CO_2

$$CO_2 = \frac{12.011 \text{ gms C}}{44.009 \text{ gms } CO_2} = 0.273 \text{ gm C/gm } CO_2$$

The final step is to calculate the miles per gallon of M85.

$$MPG = \frac{(0.444 \text{ gm C/gm M85})(2960.4 \text{ gm M85/gal M85})}{[0.866(HC) + 0.429(CO) + 0.273\ (CO_2)] \text{ gms C/mile}}$$

where HC = gms/mile of HC emitted
CO = gms/mile of CO emitted
CO_2 = gms/mile of CO_2 emitted

Fuel Economy Test

In order to gain some confidence of the success of the engine modifications to increase the fuel economy, a preliminary fuel economy test was performed at 55 miles per hour (mph) on a public interstate highway. A relatively level stretch of interstate was found on Interstate 75 south just outside of Knoxville. It was decided to start the test at exit number 81, go approximately 30 miles, turn around, and return to the starting point. A level parking lot near the exit was used as the refueling station.

The test vehicle's fuel tank was filled to a known level, and the vehicle's location was noted such that upon returning, the same location could be used when checking the amount of fuel consumed. This was done in order to insure that if the vehicle was not exactly level during refueling, it would be in the same orientation as during the initial fill.

The engine's control scheme contains a routine analogous to the accelerator pump in a carburetor and it was desired to reduce the throttle movement in order to avoid this routine. The method for holding the vehicle's speed constant involved trying to anticipate changes in road conditions and reacting ahead of time. Therefore, the

throttle position was noted on the CDM when the desired speed was reached and throughout the remainder of the test, an attempt was made to minimize the deviation from the noted throttle position while continuing to hold the vehicle's speed constant.

In earlier work on the vehicle, it had been discovered that when the fuel tank was filled such that the fuel level was at the fill cap, the fuel level would mysteriously fall gradually back down into the tank. After investigation, it was discovered that when the fuel level was filled up to the fill cap, fuel vapor flowed into the evaporative emissions system which is designed to collect fuel vapors out of the gases leaving the fuel tank. The problem was alleviated by disconnecting the vapor canister from the evaporative emissions system and clamping the end of the loose hose such that the trapped vapors would not allow the fuel the enter the system.

After the vehicle was refueled, the vapor canister was reconnected and the test was begun. During the test, the vehicle speed was monitored and kept within two mph of 55 mph. At the end of the test, the vehicle was returned to the refueling site and the amount of fuel required to return the fuel level to where it was at the beginning of the test was measured. The fuel was measured using a 1000 milliliter graduated cylinder.

The final fuel economy tests were performed in the same manner as the preliminary fuel economy test, however, they were performed on a seven and a half mile high speed test track at the Transportation Research Center of Ohio. In order to consume approximately the same amount of fuel for each steady state speed test, a calculation was performed to estimate the time required for each test if the total time for all three tests was chosen to be three hours. For this approximation, it was assumed that the vehicle would achieve the same fuel economy at each speed. From these assumptions the following times were calculated for each test: one hour and 12 minutes for the 45 mile per hour test, 57 minutes for the 55 mile per hour test, and 50 minutes for the 65 mile per hour test.

CHAPTER VI
RESULTS AND DISCUSSION

Dynamometer Testing

The primary objective of the dynamometer testing was to configure the development engine such that its torque versus speed curve matched that of the baseline engine. This became a very eventful task.

After the engine was converted to operate using only four cylinders, a torque test was performed using the turbocharger used on the test vehicle during the SAE Methanol Marathon and Methanol Challenge student design competitions. The development engine's torque output was significantly lower than the baseline engine's output at engine speeds of 2800 RPM or below. The difference ranged from 58 foot-pounds at 1200 RPM to ten foot-pounds at 2800 RPM. However, the development engine's torque was ten and 13 foot-pounds higher at an engine speeds of 3200 and 3600 RPM, respectively. This was an indication that the turbocharger being used was too large and could not reach the proper operating speed at low engine speeds. This was also indicated by the lack of manifold boost at low engine speeds. A smaller turbocharger was then chosen and put on the engine.

A new torque test was performed after the turbocharger was changed. In this test, the developmental engine's torque matched the baseline engine's torque within four foot-pounds at engine speeds between 2400 and 3600 RPM. Unfortunately, at 4000 RPM a sudden loss in engine power was experienced and the test had to be terminated.

It turned out that a hole had developed in the crown of one of the four operating pistons. The aluminum which used to occupy the hole had dispersed throughout the engine, causing damage to the cylinder head, engine bearings, and the oil pump. The

damage to the piston and cylinder head can be seen in Figures 18 and 19. The cylinder heads that were being used on the engine, at the time of the failure, had been treated with a ceramic coating. After close examination of the damaged cylinder head, it was discovered that the ceramic coating, in the damaged cylinder, had become delaminated from the head. It was believed that the delaminated ceramic could not cool adequately and developed a hot spot which caused preignition which eventually blew a hole in the crown of the piston.

After the engine was rebuilt, it was run at various speeds for approximately five hours to break in the new parts and to allow the new piston rings to seat. After being broken in, the engine's torque output was again tested.

The developmental engine was able to match the baseline engine's torque exactly at 2000 RPM, and at engine speeds of 2400 RPM and above the developmental engine's torque output exceeded that of the baseline engine. The results of the test are tabulated in Table 2, and the power versus speed curves are plotted in Figure 20.

It was believed that with the use of an automatic transmission, the reduced torque below 2000 RPM would not affect the vehicle's performance since the vehicle would not operate at engine speeds less than 2000 RPM at WOT.

Also shown in Figure 20 is the power curve for the six cylinder methanol engine which was used for the Methanol Challenge Competition (18). As can be seen, when the power capabilities of methanol are exploited, it is possible to obtain a sizable increase in power. However, it has been the purpose of this investigation to exploit the fuel economy potential of methanol.

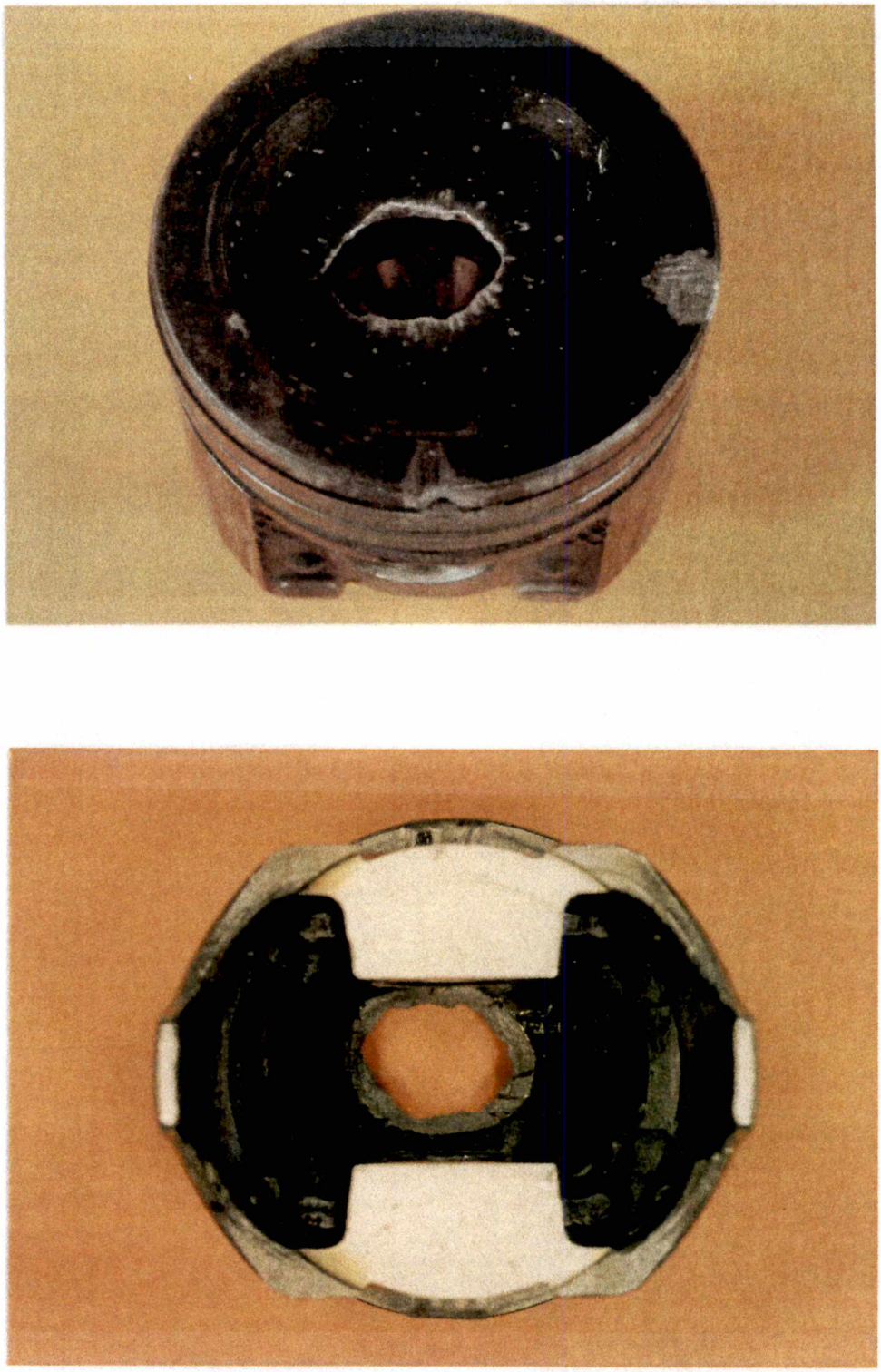


Figure 18—Piston Damaged During Dynamometer Testing.

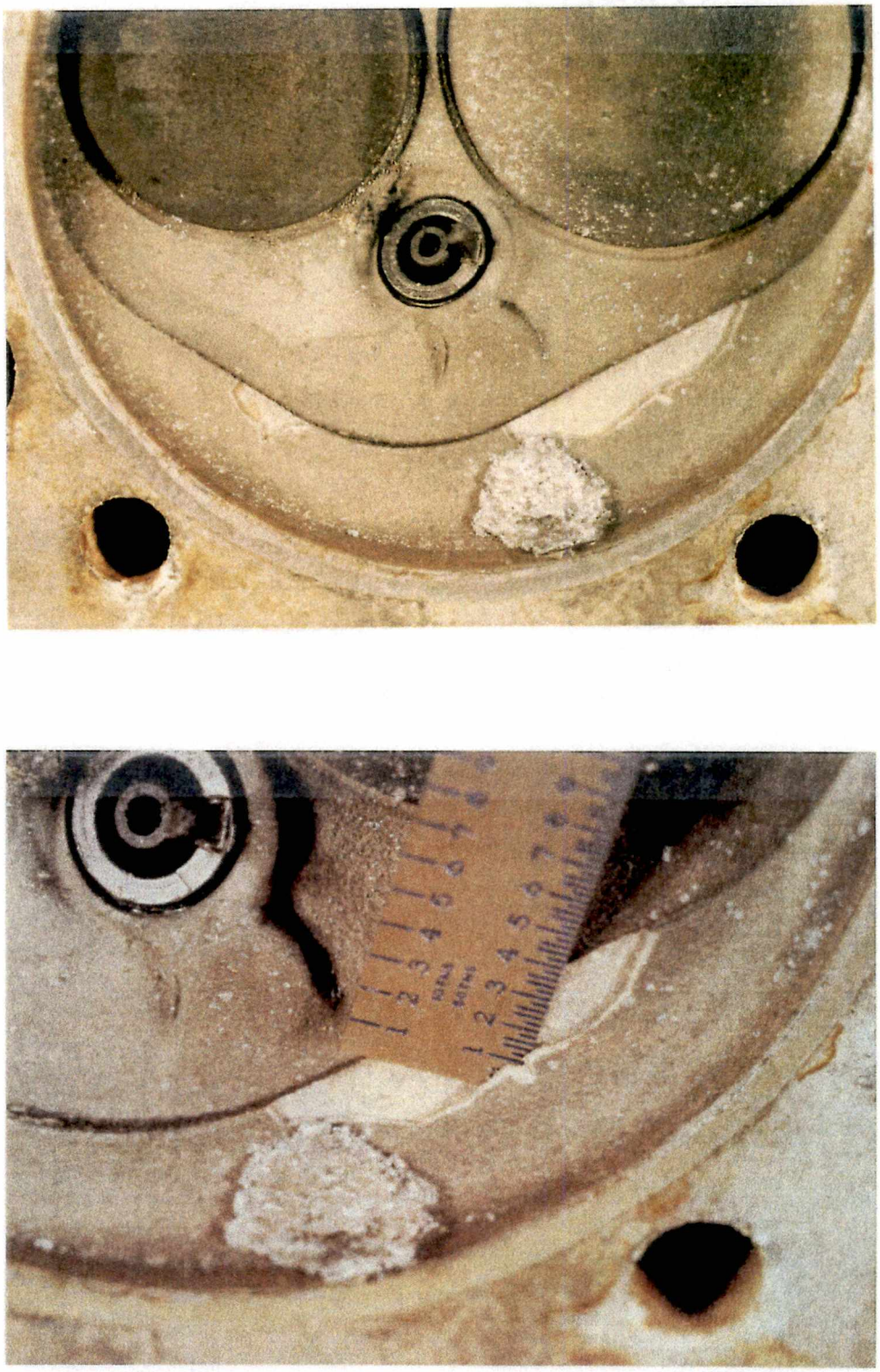


Figure 19—Damage to Cylinder Head (Note Missing Spark Plug Electrodes).

Table 2–Results from Final Dynamometer Test.

	V-6 Gasoline Baseline Engine		V-4 Methanol Development Engine		
Date	2/16/89		8/6/92		
Barometric Press (in Hg)	29.6		29.44 @78.5 °F		
Dry Bulb Temp (°F)	76		78.5		
Wet Bulb Temp (°F)	61		71.5		
Engine Speed (rpm)	Torque (ft - lbf)	Power (HP)	Torque (ft - lbf)	Manifold Pressure (psig)	Power (HP)
1200	137	31.30	–	–	–
1600	140	42.65	113	3.0	34.42
2000	142	54.07	142	6.0	54.07
2400	142	64.89	148	6.0	67.63
2800	145	77.30	152	6.5	81.03
3200	149	90.78	158	7.0	96.27
3600	149	102.13	160	8.0	109.67
4000	144	109.67	158	8.0	120.33
4400	136	113.94	–	–	–
4800	137	125.21	–	–	–

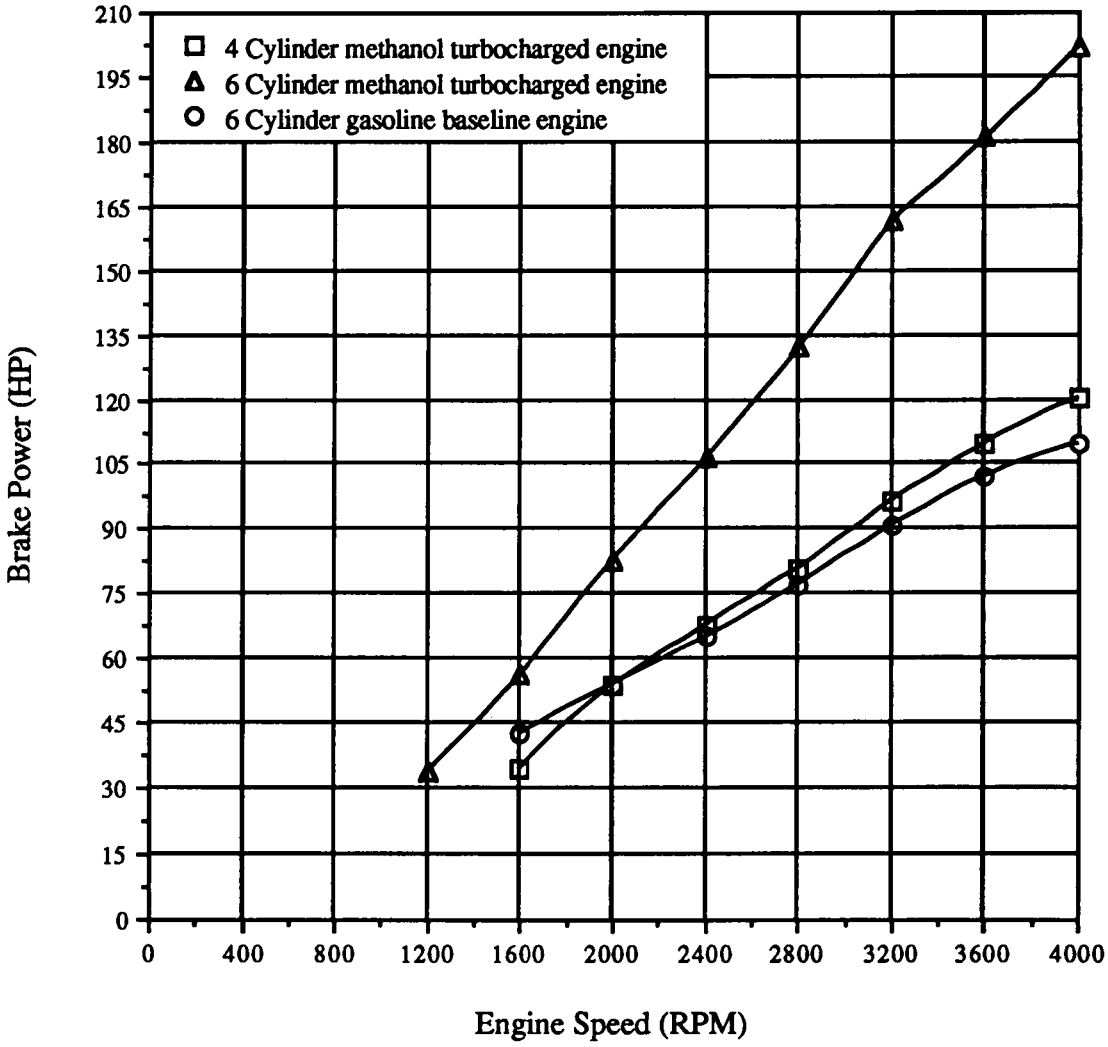


Figure 20—Horsepower Versus Speed Curves for the V-4 and V-6 Methanol Engines and the V-6 Gasoline Baseline Engine.

Acceleration Test

The objective of the acceleration test was to show that the performance characteristics of the test vehicle, with the development engine installed, were not significantly different than that of the stock vehicle. Since the acceleration characteristics of the vehicle are greatly dependent on the driver several acceleration runs were made using the same driver.

Based on the test results the 0-500 feet acceleration time for the four cylinder methanol developmental vehicle was found to be 10±0.15 seconds. For the six cylinder gasoline baseline vehicle the time was found to be 10±0.25 seconds. In addition to the 0-500 feet acceleration tests, 0-60 miles per hour acceleration runs were made. The results for these tests were twelve seconds for the four cylinder methanol and eleven seconds for the gasoline baseline. These results, along with the results from the dynamometer tests, verified that the developmental vehicle had essentially the same acceleration performance as the baseline vehicle.

Emissions Test

The purpose of performing the emissions test was to demonstrate that a lean burn tactic had not been used in order to achieve an increase in the vehicle's fuel economy. During the first attempt to run an emissions test, a problem developed with the test vehicle's engine. After close examination, it was decided that the problem was most likely caused by faulty fuel injectors. In order to test this hypothesis, the fuel injectors were removed from the engine and a static flow test was run. The results of the flow test indicated that two of the injectors being used were faulty and were not able to deliver the appropriate amount of fuel to the engine. New fuel injectors were then installed and another emissions test was performed.

The results from the emission test and the EPA's certification data for the 1988

Corsica are shown in Table 3. The results indicate that the vehicle was definitely not running lean. In fact, the high CO and low NO_x values indicate that the engine was actually running rich. Unfortunately this meant that the fuel economy was being reduced.

Fuel Economy

This section includes all the fuel economy results obtained during the course of this study. These include results from the preliminary fuel economy test, data calculated from FTP emissions tests, and also the final fuel economy test. In reporting the results, four sources are referred to. The vehicle equipped with the V-4 methanol developmental engine is referred to as the four cylinder methanol developmental vehicle. The vehicle equipped with the V-6 gasoline baseline engine used in the dynamometer and acceleration comparisons is referred to as the six cylinder gasoline baseline vehicle. The remaining two data sources are the EPA certification data for the 1988 Chevrolet Corsica and the results obtained from the 1989 gasoline fueled Corsica used as the control vehicle in the 1990 Methanol Challenge. When reporting the fuel economy results of an M85 fueled vehicle, it is customary to report the gasoline equivalent fuel economy in which the two fuels are compared on an equal energy basis. To arrive at the gasoline equivalent fuel economy, the M85 fuel economy is multiplied by the ratio of gasoline's heating value (32.16 MJ/liter) to the heating value of M85 (18.34 MJ/liter), which is 1.754 (20).

The preliminary fuel economy test was the first to be performed and its purpose was to give an indication as to the success of the project before going to the expense of running a controlled fuel economy test. The test vehicle's fuel economy improved as expected and the results are shown in Table 4. From these results, it can be seen that the gasoline equivalent fuel economy of the developmental vehicle was from 11 to 17 percent higher than that of the gasoline baseline vehicle.

Table 3–Emissions Results from EPA's FTP City Cycle (grams/mile).

	Date	HC	NO_x	CO	CO_2
EPA Certification for 1988 Corsica(21)		0.35	1.53	3.1	—
4 Cylinder Methanol Developmental Vehicle	2/12/92	0.40	0.99	5.17	395.26
	5/25/93	0.63	0.49	13.69	332.96

Table 4–Highway Fuel Economy Test Results (mpg).

Date	**6 Cylinder Gasoline Baseline Vehicle**	**4 Cylinder Methanol Developmental Vehicle***
7/7/93	37.0	—
12/31/92	—	43.4 (17%)
5/13/93	—	43.4 (17%)
5/27/93	—	41.1 (11%)

*Gasoline equivalent fuel economy. (Numbers in parenthesis indicate percent increase over gasoline baseline.)

As discussed earlier, fuel economy data were generated during both the FTP city and FTP highway emission tests. These results are shown in Table 5 along with the data for the gasoline baseline vehicle and the control vehicle used during the 1990 Methanol Challenge. From this table it can be seen that over both the EPA city cycle the development vehicle showed a fuel economy increase of 13 percent over the EPA certification data and 20 percent over the Methanol Challenge control vehicle. Over the FTP highway cycle the developmental vehicle's fuel economy showed a 14 percent increase over the EPA certification data and an 18 percent increase over the Methanol Challenge control vehicle.

The final test run for determining the fuel economy of the vehicle was the steady state test on a closed loop track. The results from this test are given in Table 6. From these results it can be seen that there was a 14, 13, and 3.4 percent increase in fuel economy respectively at 45, 55, and 65 miles per hour. The results from all the vehicle fuel economy tests are compiled graphically in Figures 21 and 22. Figure 21 compares the developmental vehicle's fuel economy with the six cylinder gasoline vehicle for the steady speed tests and with the EPA certification data for the FTP city and highway tests. Figure 22 is similar to Figure 21 except it uses the Methanol Challenge control vehicle as the bases for comparison.

Uncertainty Analysis

An uncertainty analysis was performed on the preliminary and final fuel economy tests. The two parameters scrutinized were the distance traveled and the volume of fuel consumed. From the specifications for the Track Test Fifth Wheel, the uncertainty of the distance measurement was taken to be one percent. Since the fuel consumption was found by measuring the amount of fuel required to refill the fuel tank after a test, the uncertainty of the fuel consumption measurement was found by determining the uncertainty in the refueling procedure. The method used to determine

Table 5–Emission Test Fuel Economy Results (mpg).

	FTP City	FTP Hwy
EPA Certification for 1988 Corsica (21)	21.1	37.3
Methanol Challenge Control Vehicle (21)	19.9	35.8
4 Cylinder Methanol Developmental Vehicle*	23.8	42.4
Increase Over EPA Certification Data	13%	14%
Increase Over Control Vehicle	20%	18%

*Gasoline equivalent fuel economy.

Table 6–Steady State Fuel Economy Test Results (mpg).

	45 mph	55 mph	65 mph
6 Cylinder Gasoline Baseline Vehicle	38.6	34.9	31.2
Methanol Challenge Control Vehicle (21)	40.1	32.5	29.2
4 Cylinder Methanol Developmental Vehicle*	44.1	39.4	32.2
Increase Over Baseline Vehicle	14%	13%	3.4%
Increase Over Control Vehicle	10%	21%	10%

*Gasoline equivalent fuel economy.

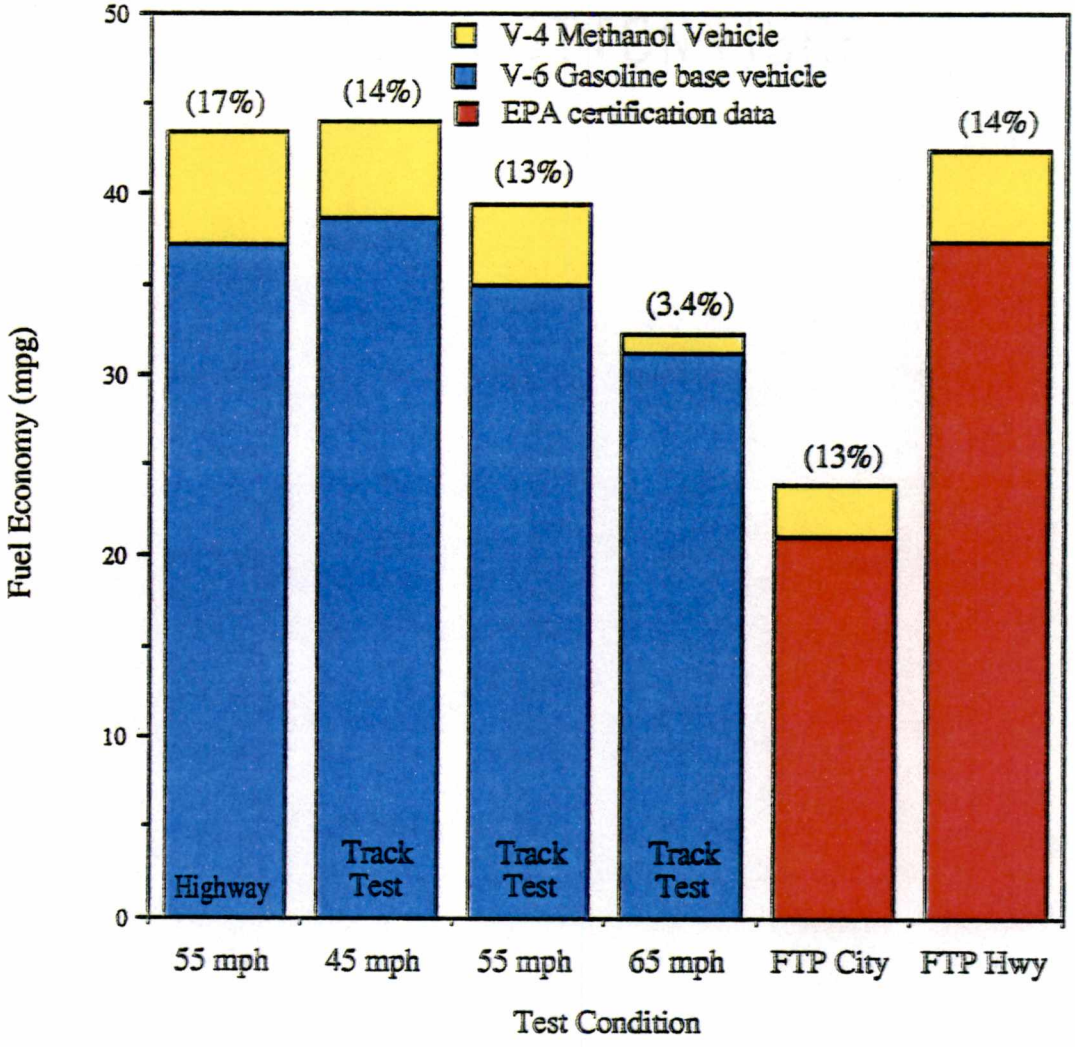


Figure 21—Fuel Economy Results Compared to the V-6 Gasoline Baseline Vehicle and the EPA Certification Data (Numbers in parenthesis represent percent increases).

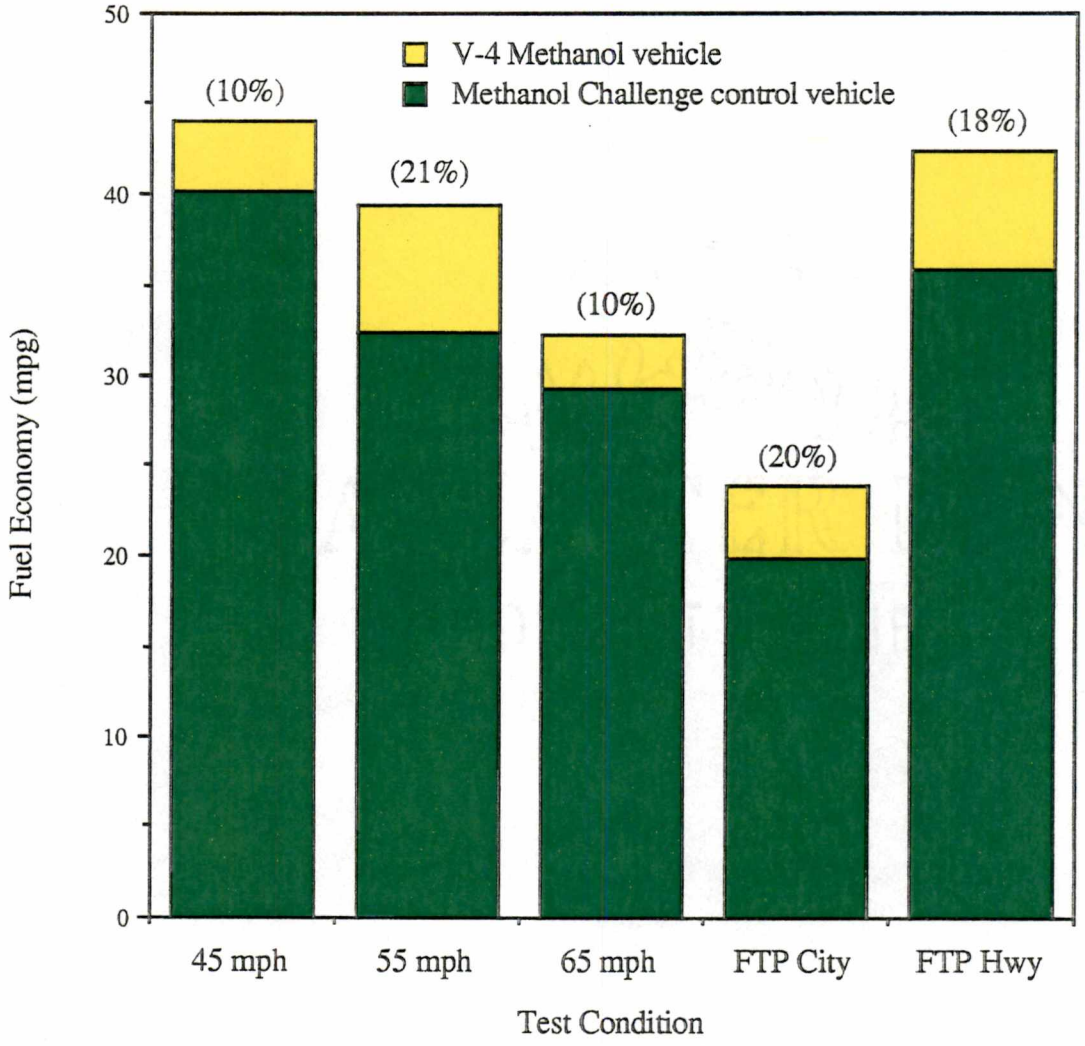


Figure 22—Fuel Economy Results Compared to the Methanol Challenge Control Vehicle (Numbers parenthesis represent percent increases).

this uncertainty involved repeated trials of the preliminary fuel economy test for the six cylinder gasoline baseline vehicle. A total of six trials were made using the same procedure. The results from these trials are given in Table 7.

A statistical analysis was conducted on the fuel economy test results in the manner outlined in Holman (22). First the mean and sample standard deviation were both calculated for the amount of fuel consumed. The mean was found to be 6.70 liters and the sample standard deviation was found to be 0.215 liters. Using Chauvenet's criterion the results from trial number three were disregarded because it was 1.95 times the sample standard deviation above the mean. (For a sample size of six, Chauvenet's criterion specifies that any data point out side the range of 1.73 times the sample standard deviation around the mean can be disregarded.) With trial number three disregarded, the new mean was calculated to be 6.79 liters and the sample standard deviation was found to be 0.0726 liters. Therefore, there is a 95% probability that the fuel consumption's deviation from the mean will be less than 0.142 liters. Also at 55 mph the mean fuel economy was found to be 37.0±0.3 mpg with a 95% confidence. Using these results, the uncertainties for each fuel economy test were calculated and are given in Table 8.

Table 7–Repeated 55 mph Highway Fuel Economy Trials.

Trial No.	Distance Traveled (miles)	Fuel Consumed (liters)
1	66.4*	6.76
2	66.4*	6.76
3	66.5	6.285
4	66.4	6.91
5	66.2	6.72
6	66.4*	6.785

*Distance not measured but taken to be the average of distance measured during other trials

Table 8–Steady State Fuel Economy Results and Uncertainty Analysis.

	Date	Speed (mph)	Place	Distance (mi)	Fuel (gal)	Fuel Economy (mpg)	Uncert.
4 Cyl. Methanol Develop. Vehicle	12/31/93	55	Hwy	66.4	2.68	24.8	1.9%
	5/13/93	55	Hwy	66.5	2.69	24.7	1.9%
	5/17/93	45	Track	68.0	2.71	25.1	1.9%
	5/17/93	55	Track	53.0	2.36	22.5	2.0%
	5/17/93	65	Track	53.0	2.89	18.4	1.8%
	5/27/93	55	Hwy	66.2	2.83	23.4	1.8%
6 Cyl. Gasoline Baseline Vehicle	7/6/93	55	Hwy	66.4*	1.79	37.2	2.6%
	7/7/93	55	Hwy	66.4*	1.79	37.2	2.6%
	7/16/93	45	Track	68.0	1.76	38.6	2.6%
	7/16/93	55	Track	53.0	1.52	34.9	2.9%
	7/16/93	65	Track	53.0	1.70	31.2	2.7%
	7/26/93	55	Hwy	66.5	1.66	40.0	2.7%
	7/26/93	55	Hwy	66.4	1.83	36.4	2.5%
	7/26/93	55	Hwy	66.2	1.78	37.3	2.6%
	7/26/93	55	Hwy	66.4*	1.79	37.0	2.6%

*Distance not measured but taken to be the average of distance measured during other trials

CHAPTER VII
CONCLUSIONS

Based on the results of this study, two conclusions can be drawn. It has been shown that the properties of methanol can lead to wide-open-throttle performance comparable to that of gasoline but from a smaller displacement engine. However, the use of a turbocharger, as in this study, results in reduced low end torque which may be unacceptable for use with a manual transmission.

It has also been shown that if methanol's properties are used to produce an increase in the vehicle's fuel economy potential rather than its power-related performance, sizable fuel economy increases can be accomplished. Increases in gasoline equivalent fuel economy of approximately 19 percent can be achieved over the FTP city and highway cycles, and gasoline equivalent fuel economy increases of up to 21 percent can be achieved at steady state highway speeds.

CHAPTER VIII

RECOMMENDATIONS

Based on the results of this study, the following recommendations are made:

1. Improve the reduced engine displacement concept by using a production four cylinder engine instead of modifying a V-6 engine.
2. Improve the low end torque with the use of a supercharger rather than a turbocharger.
3. Develop a system for measuring instantaneous fuel consumption which could be combined with the output from the fifth wheel in order to obtain real time fuel economy samples. With this type of system, fuel economy data could be obtained over the entire operating range of the vehicle. Also, a statistical analysis could then be performed on the data in order to determine the actual uncertainty.

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APPENDIX

Table A1–Results from Dynamometer Test Using Turbocharger from Methanol Challenge.

	V-4 Methanol Development Engine	
Date	6/23/92	
Barometric Press (in Hg)	29.18 @ 77°F	
Dry Bulb Temp (°F)	79.5	
Wet Bulb Temp (°F)	65.5	
Engine Speed (rpm)	Brake Torque (ft - lbf)	Manifold Pressure (psig)
1200	79	0.5
1600	94	1.0
2000	106	2.0
2400	119	4.0
2800	135	5.5
3200	159	8.5
3600	162	8.5

Note: Test terminated because turbo was too big.

Table A2–Results from Dynamometer Test when Piston Failed.

	V-4 Methanol Development Engine	
Date	6/26/92	
Barometric Press (in Hg)	—	
Dry Bulb Temp (°F)	—	
Wet Bulb Temp (°F)	—	
Engine Speed (rpm)	Brake Torque (ft - lbf)	Manifold Pressure (psig)
1200	85	0.0
1600	107	2.5
2000	133	5.5
2400	142	6.5
2800	147	6.5
3200	153	7.0
3600	157	7.5
3800	101	—

Note: Test terminated because of engine trouble.

Table A3–Results from Dynamometer Motoring Test.

	V-6 Gasoline Baseline Engine		V-4 Methanol Development Engine	
Date	2/16/89		8/6/92	
Barometric Press (in Hg)	29.6		29.44 @78.5 °F	
Dry Bulb Temp (°F)	76		78.5	
Wet Bulb Temp (°F)	61		71.5	
Engine Speed (rpm)	Torque (ft - lbf)	Power (HP)	Torque (ft - lbf)	Power (HP)
1200	–	–	–	–
1600	–	–	-14	-4.26
2000	–	–	-15	-5.71
2400	–	–	-17	-7.77
2800	–	–	-24	-12.79
3200	–	–	-28	-17.06
3600	–	–	-33	-22.62
4000	–	–	-38	-28.94

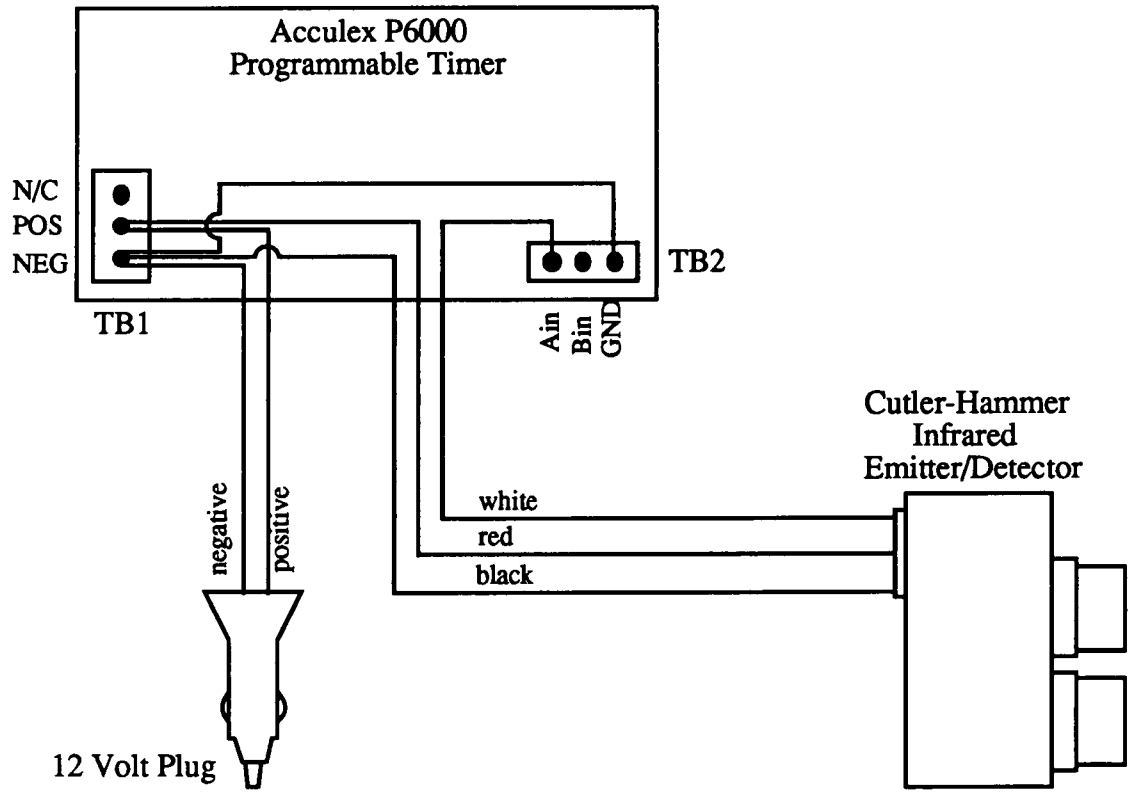


Figure A1—Schematic of Infrared Timer.

Table A4–Results from Acceleration Test for 4 Cylinder Methanol Developmental Vehicle.

0-500 Foot Acceleration

Infrared Timer (sec)	VC200 (sec)	Max Speed (mph)	Comments
10.925	10.79	55.3	"Bogged" off line
—	10.81	54.7	Loss of traction
—	10.11	54.5	"Good" run
8.289	9.96	52.3	Seemed Fast
10.319	10.32	54.3	Two people in car
10.02	10.09	54.7	ok
9.877	9.86	54.8	

0-60 MPH Acceleration

VC200 (sec)	Comments
12.19	not maxed out
11.92	traction loss
11.97	ok

Table A5–Results from Acceleration Test for 6 Cylinder Gasoline Vehicle.

0-500 Foot Acceleration

Infrared Timer (sec)	VC200 (sec)	Comments
—	9.77	6000 rpm shift
—	9.69	5000 rpm shift
9.839	9.78	
9.795	9.72	
9.908	9.74	
9.784	9.81	
10.041		
9.655		5500 rpm shift
10.128		5000 rpm shift
9.749		
10.21		

0-60 MPH Acceleration

VC200 (sec)	Comments
10.79	
10.84	5000 rpm shift
10.59	
10.47	

Table A6–Fuel Injector Static Flow Test.

Injector No.	Volume measured (milliliters)	Time (sec)	Flow rate (ml/s)	Deviation from Maximum
47	93.5	16.97	5.51	-7.38%
49	93.0	16.81	5.53	-7.01%
54	96.0	23.74	4.04	-32.03%
59	95.0	16.01	5.93	-0.26%
62	94.5	16.05	5.89	-1.03%
71	91.5	15.38	5.95	0.00%

Note: Pressure in fuel rail was 52 psig

Table A7–Emission Test Results from 2/12/93.

Cycle	City			Highway
Bag	1	2	3	1
Vmix (ft^3)	5659	9763	5679	8554
Dilution factor	24.33	31.54	26.24	21.62
Fuel Economy (miles/gallon)	13.00	10.80	13.99	21.39
Test distance (miles)	3.59	3.89	3.59	10.26
HC (ppm)	57.58	2.94	4.75	1.98
HC (grams)	5.32	0.47	0.44	0.28
HC (grams/mile)	1.48	0.12	0.12	0.03
NO_x (ppm)	11.50	7.10	9.70	18.40
NO_x (grams)	3.76	4.00	3.18	8.46
NO_x (grams/mile)	1.05	1.03	0.89	0.82
CO (ppm)	216.00	37.00	89.00	23.00
CO (grams)	40.30	11.91	16.66	6.49
CO (grams/mile)	11.23	3.06	4.64	0.63
CO_2 (%)	0.43	0.34	0.41	0.52
CO_2 (grams)	1262.5	1720.99	1213.65	2307.50
CO_2 (grams/mile)	351.67	442.41	338.07	224.9
Fuel Economy (miles/gallon)	11.96			21.39

Table A8–Emission Test Results from 5/25/93.

Cycle	City			Highway
Bag	1	2	3	1
Vmix (ft^3)	5783	9885	5732	8606
Dilution factor	24.56	38.39	28.01	24.48
Fuel Economy(miles/gallon)	12.85	13.32	14.83	24.17
Test distance (miles)	3.59	3.89	3.59	10.20
HC (ppm)	55.12	5.08	31.11	4.12
HC (grams)	5.21	0.82	2.91	0.58
HC (grams/mile)	1.45	0.21	0.81	0.06
NO_x (ppm)	6.00	2.80	4.30	6.00
NO_x (grams)	2.25	1.80	1.60	3.28
NO_x (grams/mile)	0.63	0.46	0.45	0.32
CO (ppm)	353.04	103.03	371.04	88.08
CO (grams)	67.31	33.58	70.12	24.99
CO (grams/mile)	18.75	8.63	19.53	2.45
CO_2 (%)	0.41	0.27	0.36	0.45
CO_2 (grams)	1236.18	1357.26	1055.11	1999.63
CO_2 (grams/mile)	344.34	348.91	293.9	196.04
Fuel Economy (miles/gallon)				

VITA

Shean Patrick Huff was born in Spartanburg, South Carolina on September 26, 1969. At the age of three weeks, he moved to Knoxville, Tennessee where he lived his childhood. Shean attended elementary school at First Lutheran Christian School and high-school at Central High-school in Knoxville. He graduated from high-school in June of 1987, and the following September he entered the University of Tennessee, Knoxville with a major in Mechanical Engineering. As an undergraduate, Shean was very active in scholastic and extra curricular activities. He played base drum with the University of Tennessee Marching Band his freshman year and was very active with the Baptist Student Union (BSU) throughout his undergraduate career. As a junior, Shean participated in the BSU summer missionary program and had the opportunity to travel to Africa for a month. Also, as an undergraduate, Shean met his future wife, Kim, who he dated for four years before marrying. Following his graduation with a Bachelor of Science degree in Mechanical Engineering in May 1992, he accepted a research assistantship and began study toward a Master of Science degree in Mechanical Engineering.

Shean received his Masters of Science degree in Mechanical Engineering in August 1993.