EMISSION CONTROL IN DIESEL ENGINES BY ALCOHOL FUMIGATION

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DISCLAIMER

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EXECUTIVE SUMMARY

INTRODUCTION

Exhaust emissions from diesel engines are a substantial source of air pollution in this country. In recognition of this fact, the Environmental Protection Agency has issued strict new regulations due to take effect in 1991 and 1994 that will drastically reduce the amount of some pollutants these engines will be allowed to emit. The technology is not currently available to produce diesel engines that can meet these regulations without large penalties in engine performance and efficiency. One technique that offers promise of being able to reduce emissions from both existing engines and new engines is alcohol fumigation.

PROBLEM STUDIED

The objective of this project was to investigate the use of alcohol fumigation to control the exhaust emissions from diesel engines. Previous testing by the authors and others has shown that there is considerable potential for reduction of particulates and oxides of nitrogen using this technique. Specifically, the project was directed to address the questions of operating cost, formulation of alcohol-water mixture and the potential for producing increases in other emissions.

Alcohol fumigation involves injecting alcohol or an alcohol-water mixture into the intake air stream of the engine. The injected alcohol and water vaporizes and is compressed along with the air in the cylinder. Diesel fuel is injected into the engine in the normal manner but a portion of the power from the engine is provided by the alcohol, typically 10-30%.

All of the previous work that has been done on alcohol fumigation has been under steady state operating conditions. This means the engine is operated at a fixed speed and load while data are collected. However, this is not typical of actual engine operation in trucks and buses. In order to test the emission reduction from alcohol fumigation under conditions typical of actual engine operation, the engine dynamometer and emission measurement system in the Internal Combustion Engine Laboratory at Iowa State had to be upgraded to allow transient testing. Most of the effort expended on this project was devoted to this upgrading. The result is a facility believed to be unique among U.S. universities in regard to its ability to simulate the EPA's Federal Transient Test Procedure.

RESULTS ACHIEVED

The optimum formulation for the alcohol additive that is injected into the engine appears to be about 80 proof ethanol. Higher proofs than this do not give as much reduction in oxides of nitrogen emissions and lower proofs give increases in carbon monoxide and unburned hydrocarbon emissions. There does not appear to be any difference between methanol and ethanol in their emissions reduction potential although a higher concentration of methanol would be required to give the same effect as 80 proof ethanol.

The cost of operating an engine with alcohol fumigation depends on the amount of emission reduction desired. The steady state data indicated that the engine could be operated with up to 30% of its torque supplied by the fumigated alcohol. However, to accomplish this would require very large flow rates of the alcohol-water mixture since the 80 proof mixture is only about 33% ethanol and ethanol has only about 65% of the energy content of diesel fuel. The transient tests, although not conclusive, indicate that if the alcohol is injected only on demand, then the amount of alcohol required to achieve significant reductions in emissions may be much smaller. Typically, the mass of the alcohol-water mixture consumed during the transient test is about 5-20% of the diesel mass and this is only about 33% ethanol. Thus, it may be possible to reduce emissions using an amount of ethanol equal to from 2-6% of the diesel fuel consumed. Reductions of 12.7% in oxides of nitrogen emission and 28.3% in particulates have been observed under transient conditions.

There is a definite increase in the amount of aldehydes produced by the engine when operated on alcohol. Although we did not determine which aldehyde species were present, since the alcohol used was ethanol we expect that the principal aldehyde was acetaldehyde. The aldehyde was observed to increase by as much as 2.5 times its diesel-only value. However, since the aldehyde emissions from diesel engines have not generally been considered to be a problem, this increase does not necessarily mean that higher aldehyde emissions would preclude alcohol fumigation.

One of the objectives of this project was to provide students with exposure to problems and technology in the transportation area. We were very successful in this regard. Four graduate students, Bennett Murray, Taner Tuken, Qiqing Jiang and Pradheepram Ottikkutti, worked on different aspects of this project. Three undergraduates also involved were Eric Hensley, William Carr and Darren Herum.

CONCLUSION

Based on our test results, we have determined that it is possible to reduce diesel emissions using alcohol fumigation. This is true even under transient conditions. Further testing is required to optimize the emission reduction but reductions of 12.7% in oxides of nitrogen and 28.3% in particulates have been observed. The engine used for this study was a four-stroke diesel engine that is characteristic of engines used for industrial and light truck use. It would be very useful to conduct a similar study using a two-stroke engine typical of most bus engines. Although large differences from the results observed for four stroke engines are not expected, the high scavenging air flow rates characteristic of two stroke engines might cause higher levels of unburned hydrocarbon emissions when using alcohol fumigation.

EMISSION CONTROL IN DIESEL ENGINES BY ALCOHOL FUMIGATION

CHAPTER 1. INTRODUCTION

Diesel engines are the most widely used power plant for over-the-road trucks and buses in the United States. These engines are also responsible for a large fraction of vehicle-generated pollutants. The Environmental Protection Agency (EPA) has mandated significantly lower diesel exhaust emission levels to take effect in 1991 and 1994 [1]. Table 1 shows the current and future standards for heavy-duty diesel engines. A large amount of research is currently being done to design and modify engines to meet the new regulations. Some cities where air pollution levels are exceptionally high are considering the curtailment or prohibition of vehicle use if maximum air pollution levels are exceeded. Such regulations will necessitate the retrofitting of existing units in the fleet in order to reduce the production of critical contaminants including oxides of nitrogen (NO_x), carbon monoxide (CO), unburned hydrocarbons (HC), and particulates.

Table 1. Environmental Protection Agency Heavy-Duty Diesel Standards

	1989	1991	1994
Hydrocarbons (g/hp-hr)	1.30	1.30	1.30
Carbon Monoxide (g/hp-hr)	15.50	15.50	15.50
Oxides of Nitrogen (g/hp-hr)	10.70	6.00	5.00
Particulates (g/hp-hr)	0.60	0.25*	0.10

*Value is 0.1 for urban buses

One technique that offers promise of being able to reduce emissions from both existing buses and new buses is alcohol fumigation. While it is well known that alcohol fumigation can reduce emissions, the mechanism for this action has not been established and the knowledge required to optimize the emissions reduction effect is not available. The objective of this project is to improve the understanding of the mechanisms involved in reducing emissions through the use of alcohol injection in order to reduce the pollution caused by diesel engines, especially in city buses. In particular, the research will answer the following questions that remain regarding its acceptability

as an emissions control device:

1. What are the operating costs of the technique?
2. What is the optimal additive formulation (mix of methanol, ethanol, and water?)
3. Does the technique produce any new, as yet unregulated, emissions?

PLAN OF ATTACK

At the start of this project the Internal Combustion Engines Laboratory at Iowa State University had a John Deere 4276T diesel engine installed on a dynamometer that could be used for this project. This engine had been used for an earlier alcohol fumigation project and a considerable amount of performance and emissions data were available. However, this engine could only be operated at fixed speeds and loads, a mode that is not typical of the way engines are used in vehicles. Therefore, it was recognized from the beginning that a large portion of the effort of this project would be directed toward upgrading the laboratory facilities to allow the engine to be operated over transient cycles.

The EPA specifies in the Code of Federal Regulations how the Federal Transient Test Cycle is to be conducted as well as how the emissions are to be measured [1]. Laboratories that have systems that satisfy these regulations can be certified by the EPA to evaluate engines. These systems are extremely expensive and cannot be justified unless the laboratory expects to be in the business of engine certification. Our intent was to develop a simplified system, at much lower cost, that would allow engine modifications to be evaluated and compared without being certified.

As mentioned above, the engine to be used for this work had been used previously for another project on alcohol fumigation. This work was sponsored by Midwest Power Concepts, Inc. of Radcliffe, Iowa -- a firm represented on the advisory board for this project. Midwest Power Concepts produces an alcohol fumigation system that it is currently developing to market as part of an emission control system for urban buses. The work described later in this report was performed using an injector and fuel supply system provided by Midwest Power Concepts although the fuel control was modified so that it could be directly controlled by our computer.

Midwest Power Concepts also encouraged us to place the primary emphasis of the project on transient testing. The original intent of the project was to conduct an extensive series of steady state tests but we decided that the effort would be better spent on transient testing of different strategies for injecting the alcohol. This decision was also influenced by the fact that a considerable amount of steady state data was already available for this engine. Some of this data

will be discussed later in the report especially in regard to questions about the optimum additive formulation.

The report is structured as follows. A brief background discussion of diesel emissions and measurement procedures is given as part of this introduction. Then the experimental apparatus and the procedures used for conducting the tests are presented. Finally the results of the tests are discussed and a summary of the conclusions is provided.

BACKGROUND

This section provides the essential background information for understanding work in the area of diesel exhaust emissions measurement. An overview of diesel emissions is given in the first section followed by an explanation of transient dynamometer tests. Then the concepts of dilution tunnels and constant volume sampling (CVS) systems are discussed.

DIESEL EMISSIONS

The primary source of air pollution from a diesel engine is the products of combustion that are discharged through the exhaust pipe. While the crankcase breather and fuel tank breather are both contributors, the exhaust pipe emissions comprise from 65 to 85 percent of the engine's pollutants. The engine exhaust contains particulates, unburned hydrocarbons (HC), oxides of nitrogen (NO_x), carbon dioxide (CO_2), carbon monoxide (CO) and traces of alcohols, aldehydes, ketones, phenols, acids, esters, ethers, epoxides, peroxides and other oxygenates. This study focuses on the measurement of particulates, NO_x, HC and CO.

Particulates in diesel exhaust are made up of carbonaceous soot particles that have hydrocarbons adsorbed and condensed onto their surfaces. The hydrocarbon part of the particulates is referred to as the soluble organic fraction (SOF). This name comes from the fact that a solvent is used to remove the organic fraction from the carbonaceous core in the analysis of a particulate sample. The amount of SOF in a particulate sample can be influenced by the manner in which the sample is collected. The amount of hydrocarbons that adsorb and condense onto the surfaces of the particulate will depend on the length of time between formation and collection and the temperatures in the collection system. The sources of hydrocarbons that make up the SOF have been shown to be both unburned fuel and engine oil [2], while the insoluble fraction is a product primarily of the unburned paraffin, olefin and aromatic fuel components.

The amount of NO_x produced by diesel engines is comparable to the amount produced by

spark-ignition engines. Typical concentrations range from 500 to 1000 parts per million or 20 grams per kilogram of fuel. Ten to thirty percent of the NO_x emissions are NO_2 while the remainder is NO. Although NO is the dominant species formed in the cylinder, most of the NO is converted to NO_2 in the atmosphere.

NO is formed by an endothermic reaction in the burned-gas regions of the cylinder. The formation of NO is very sensitive to temperature. The higher flame temperatures accompanying early or rapid combustion greatly increase NO formation. Thus, the amount of NO_x in diesel exhaust is sensitive to injection timing and rate of combustion. The cooling that takes place during the expansion stroke freezes the reaction such that the concentrations that leave the engine are much higher than the equilibrium concentrations would be for the exhaust temperature.

HC emission by diesel engines is large enough to be significant, but is about a factor of 5 lower than spark-ignition engines. The two primary sources of hydrocarbons in diesel exhaust are fuel mixed leaner than the lean combustion limit for ignition during the ignition delay period and undermixing of fuel which leaves the fuel injector nozzle at low velocity, late in the combustion process. High molecular weight hydrocarbons have very low vapor pressures and will condense on carbonaceous particulates and become the SOF of the exhaust particulates. These hydrocarbons may comprise 15 to 45 percent of the total particulate mass.

Carbon monoxide emissions are caused primarily by combustion under fuel-rich conditions. Since diesel engines generally operate with excess air, carbon monoxide emissions are usually so low that they are not a problem. However, with alcohol fumigation it is possible to have increases in carbon monoxide. These increases are caused by poor cylinder-to-cylinder distribution of the injected alcohol. Some cylinders may receive larger amounts of the alcohol than others due to the poor distribution characteristics of diesel engine manifolds. When this happens, the average fuel-to-air ratio in these cylinders is higher and there may be insufficient oxygen to burn all the alcohol and diesel fuel. These cylinders will then produce carbon monoxide even though the overall fuel-to-air ratio for the engine may be correct.

TRANSIENT DYNAMOMETER TESTS

In 1986 the Environmental Protection Agency (EPA) specified a new test procedure for heavy-duty diesel engine emissions measurement. The test requires the engine to be run through a transient operating cycle that simulates city and highway driving of a truck in both New York and Los Angeles. Figure 1 shows how the speed and load for the engine are expected to vary for the cycle. The test requires that the engine be connected to a dynamometer that can be

controlled by a computer. The computer sends signals to the dynamometer controller to regulate the engine's speed. Then, while the speed is changing to follow the transient cycle, a throttle actuator (actually since diesel engines do not have throttles, a more appropriate term is fuel injection pump rack actuator) is used to vary the engine's flywheel torque to match the required torque.

The engine is prepared for the test by cold soaking, that is, remaining inoperative in a 68 to 86 degree Fahrenheit environment for 12 hours or until the oil temperature reaches 75 F. Then the engine is started and immediately controlled according to a twenty minute schedule of speeds and torques that make up the simulation. This part is known as the cold-start or cold-cycle portion of the test. Once the schedule is complete, the engine is shut off and allowed to stand for twenty minutes. The engine is then restarted and immediately controlled according to the same schedule. Again, once the schedule is completed the engine is shut off. This part of the test is termed the hot-start or hot-cycle portion. The results of the test are normally reported as weighted averages of the emission measurements made in each of the two portions of the test. The hot-start results are weighted six times greater than the cold start results.

DILUTION TUNNELS AND CVS SYSTEMS

The Code of Federal Regulations (CFR) [1] specifies an exhaust emissions measurement system design that is to be used for determining whether an engine's emissions meet the applicable EPA standards or not. A schematic diagram of this system is shown in Figure 2. A positive displacement pump, located at the exit of the system, draws ambient air through a filter into a dilution tunnel where it mixes with the exhaust of the engine being tested. Since the entire exhaust flow of the engine is introduced into the tunnel, not just a fraction, the tunnel is termed a "full-flow" dilution tunnel.

The purpose of mixing the exhaust with dilution air in this manner is to simulate the mixing that the exhaust will undergo with the atmosphere when the engine is in actual use. Simulation of atmospheric mixing is considered necessary when measuring particulates because much of the dynamics of particulate formation occur after the exhaust leaves the engine. As the exhaust mixes with the atmosphere and cools, some of the unburned hydrocarbons will adsorb and condense onto the surface of the particulate. The total mass of adsorbed and condensed hydrocarbons is the SOF of the particulates.

FIGURE 1.

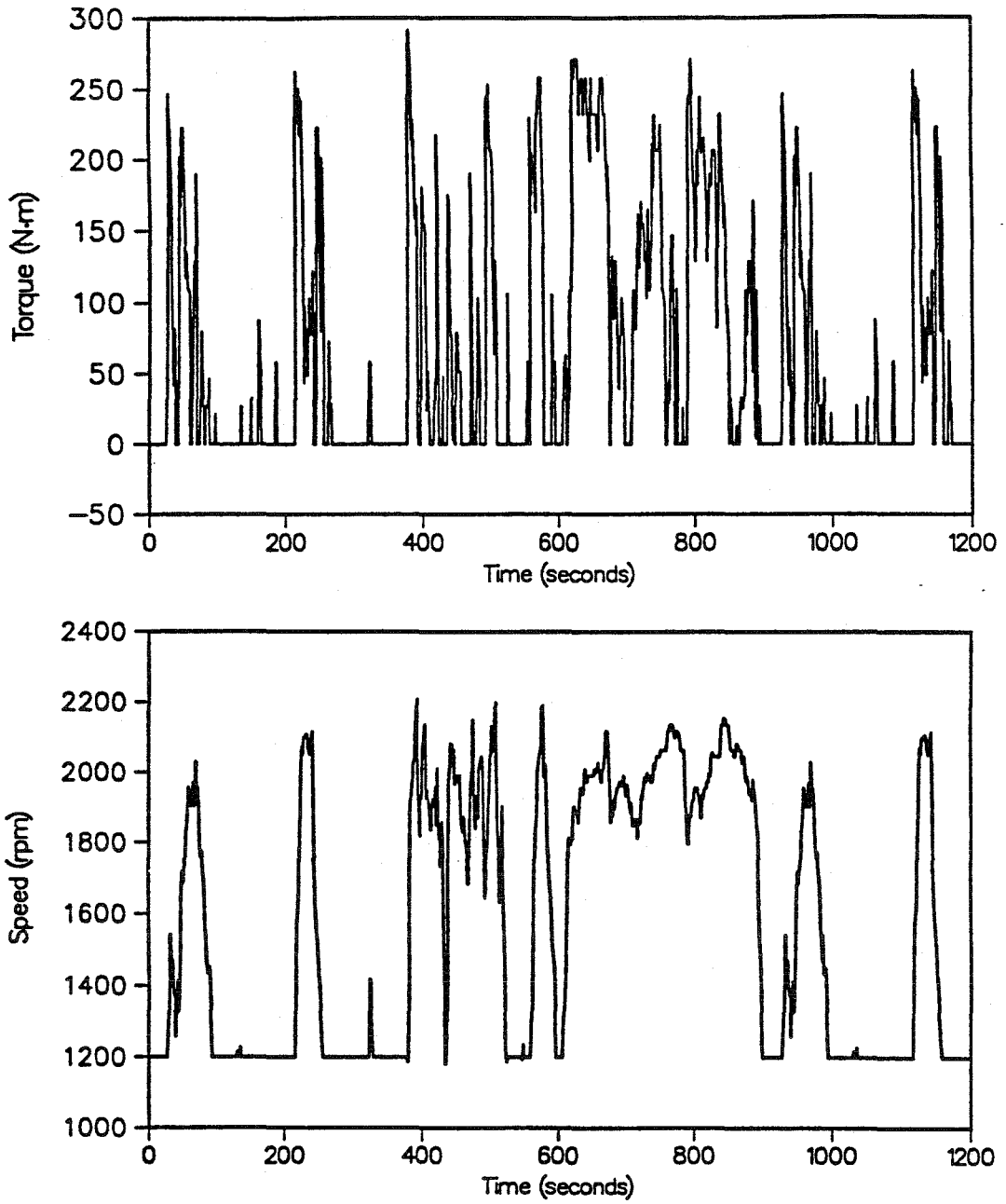

Fig. 1 Reference speed and torque trajectories for the transient cycle

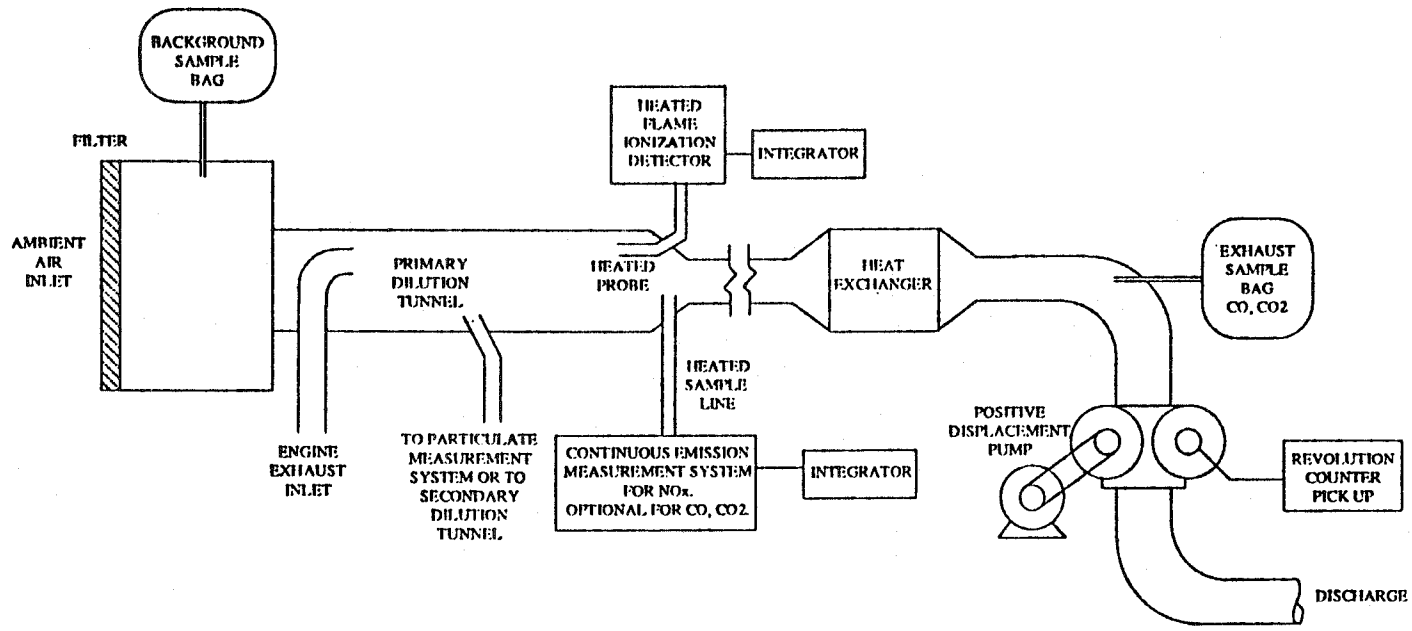

Fig. 2 Schematic diagram of the EPA-specified exhaust emissions measurement system

In addition to the interest in the total amount of mass of the carbonaceous soot plus SOF, sometimes researchers are interested in the quantity and chemical and biological characteristics of the SOF alone. In cases where a researcher wishes to characterize the size and/or shape of the individual particulates, the simulation of atmospheric mixing that the dilution tunnel provides is again important. A large amount of agglomeration among particulates tends to occur both inside and outside the engine. The dynamics of atmospheric mixing will play a role in the extent to which this occurs.

Referring back to Figure 2, the second part of the system which includes the heat exchanger and positive displacement pump, is called a constant volume sampling system or a CVS system. Despite the name, the purpose of the CVS system is to draw a constant mass flow rate of diluted exhaust through the dilution tunnel. When the test engine is running in a transient cycle, the flow rate and temperature of the exhaust entering the dilution tunnel will be highly variable. The positive displacement pump alone will draw a constant volume flow rate through the tunnel. The addition of the large heat exchanger brings the diluted exhaust to almost a constant temperature. Since the pressure in the tunnel does not vary, the fact that the positive displacement pump is drawing a constant volume flow rate of constant-temperature diluted exhaust means that the mass flow rate is also constant. It should be noted, however, that while the volume flow rate is constant downstream of the heat exchanger it is not constant upstream as the density of the mixture varies with its temperature.

A particulate sample is collected by passing a portion of the diluted exhaust through a filter. The best filters to use have been determined by experience to be Teflon-coated fiberglass [3,4]. It is important during a transient test that the flow rate of diluted exhaust passing through the filter is a constant mass fraction of the total diluted exhaust flow rate. If, for example, the flow rate of the engine exhaust increases momentarily and the sampling fraction changes, then the amount and type of particulate emitted by the engine during that period would not be accurately represented on the filter. Since the mass flow rate in the dilution tunnel is being held constant by the CVS system, drawing a constant mass flow rate through the particulate filter is required to maintain a constant mass sampling fraction. Because of the restriction of cooling the exhaust with dilution air only, and because a large amount of the particulate matter would be lost on the heat exchanger surfaces, the particulate sample must be taken upstream of the CVS heat exchanger.

The amount of dilution air required is indirectly specified by the EPA by saying that the temperature of the diluted exhaust at the filter must be maintained at 52 C or less. Further, this cooling should be accomplished only through mixing with dilution air and not by heat transfer with the tunnel or sample line walls since such interaction can result in HC condensation and

particulate deposition. Typical dilution ratios (dilution air to exhaust) range from 5 to 20.

Hydrocarbons also must be measured upstream of the heat exchanger since some might condense on the heat exchanger surfaces. A heated sample probe is used to transfer the sample from the tunnel to the HC analyzer in order to prevent condensation in the sample line. The analyzer is a heated flame ionization detector (HFID). As was the case for the particulates, the mass flow rate through the HFID must be KEPT constant in order to have a constant sampling fraction.

During a transient test, the HC concentration determined by the HFID is continuously integrated to give the total amount of HC for the whole cycle. For a gaseous emission like HC, there is another option besides maintaining a constant mass flow rate through the analyzer. The volume flow rate of the dilution tunnel can be continuously measured and multiplied by the concentration given by the HFID and integrated throughout the cycle.

Since the temperatures of the diluted exhaust should be well above the dew point when it enters the NO_x, CO and CO_2 analyzers, sampling for these gases is also done upstream of the heat exchanger before the mixture is cooled. As was the case for HC, either a constant mass flow can be maintained through these analyzers or measurements of the volume flow rate of the dilution tunnel can be used in integrating the concentrations over the transient cycle.

The EPA also allows the CO and CO_2 measurements to be made via bag sampling downstream of the heat exchanger. As mentioned before, both the volume flow rate and the mass flow rate of diluted exhaust are constant in this region. Therefore, all that is needed is a simple constant-speed pump to fill a Tedlar bag. Then, after the test is complete, the contents of the bag may be fed into CO and CO_2 analyzers. This is a desirable procedure when the response time of the analyzers is too slow for continuous measurement and integration upstream of the heat exchanger.

The exhaust emission measurement system described above would be certifiable by the EPA. However, it is very expensive to build and operate. We have built a system that approximates the EPA specified system in most respects and that should allow comparisons to be made as an engine is modified. This system is described in the next chapter.

CHAPTER 2. EXPERIMENTAL APPARATUS

The emissions measurement system that was developed for this work is presented in this chapter. Aspects of the dilution tunnel and dilution-air supply are discussed first. Then the particulate sampling and aldehyde measurement systems are presented. Finally, the engine test setup is discussed and an overview of the computerized dynamometer controller is given.

DILUTION TUNNEL

A diagram of the dilution tunnel appears in Figure 3. The dilution tunnel is 0.305 meters (12 inches) in diameter. The distance between the introduction of the exhaust and the sample probes is 3.05 meters (10 feet) which corresponds to 10 tunnel diameters. Ten tunnel diameters is generally considered to be adequate for good mixing. In a dilution tunnel, it is desirable to maximize the volume-to-surface area ratio to reduce the opportunity for interactions between the diluted exhaust and the tunnel walls. Such interactions include heat transfer, hydrocarbon (HC) condensation and particulate deposition and re-entrainment. The volume-to-surface ratio is maximized by making the diameter as large as practical.

The exhaust is introduced into the tunnel through a 90 elbow of 6 cm exhaust pipe. The elbow faces downstream. The elbow was used to direct the exhaust into the center of the tunnel cross-sectional area to enhance even mixing of the exhaust with the dilution air.

A 20 cm diameter orifice was placed in the dilution tunnel at the point of entry of the exhaust. The increase in flow velocity and turbulence caused by the orifice is intended to enhance mixing. It should be noted that although the enhancement of mixing is desirable, any type of obstruction such as mixing vanes could allow particulate deposition and re-entrainment. The orifice, when placed at the entrance to the exhaust, does not act as an obstruction to the mixture and consequently should not adversely affect particulate measurements.

The dilution tunnel exits into a high-volume ceiling exhaust fan. The fan is turned on whenever the dilution tunnel is in operation and does an adequate job of keeping the laboratory ventilated.

Sample probes are located at a point 10 feet downstream from the mixing orifice and exhaust inlet. Three stainless steel probes have been installed in the tunnel. A 3/4-inch probe leads to the particulate sampling system, a 1/4-inch probe leads to the hydrocarbon analyzer and a 3/8-inch probe leads to the oxides of nitrogen (NO_x), carbon dioxide and carbon monoxide analyzers.

FIGURE 3.

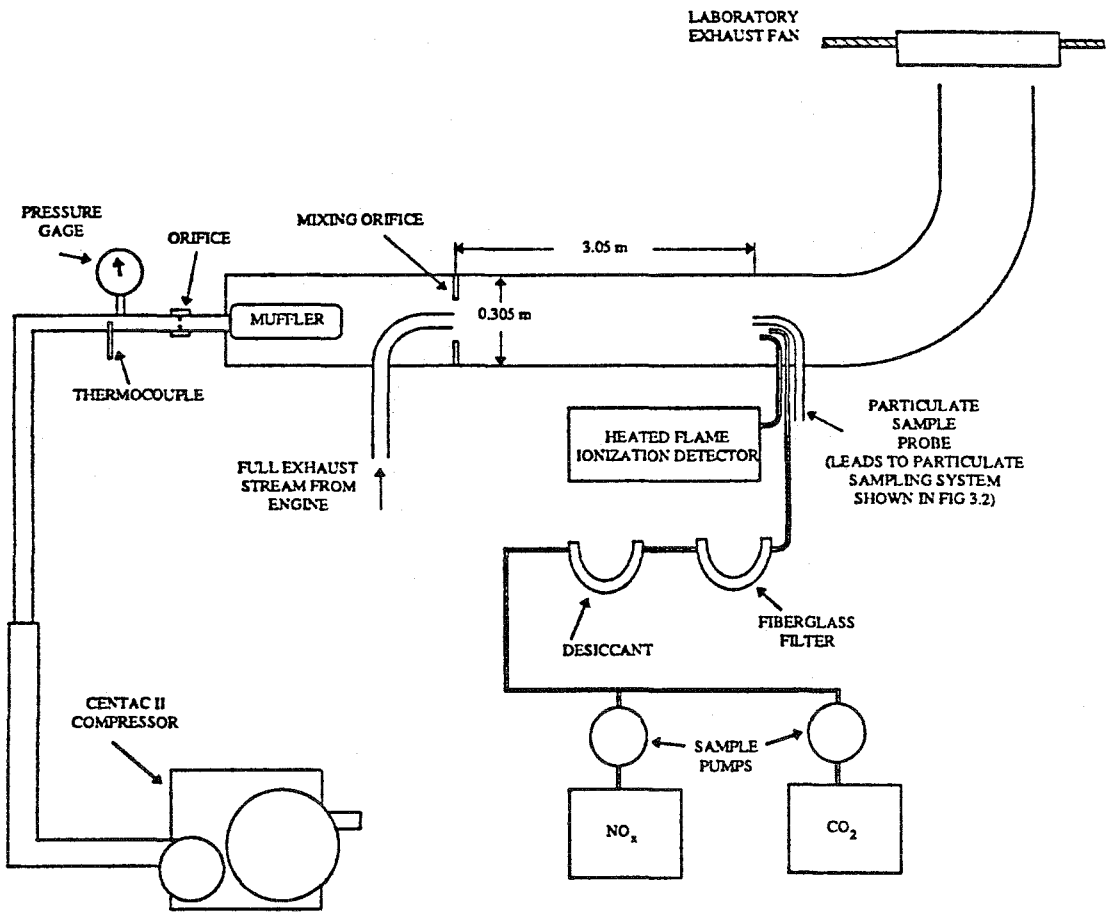

Fig. 3 Schematic diagram of emissions measurement system

DILUTION AIR SYSTEM

Dilution air is provided by an Ingersoll-Rand Centac-II two-stage air compressor located one floor below the engine laboratory. The compressor develops an outlet pressure of 620 kPa (90 psig). Between the compressor and dilution tunnel is 9.8 meters of 15 centimeter (6 inch) diameter schedule 40 pipe followed by 14.44 meters of 5 centimeter (2 inch) diameter schedule 40 pipe. The line loss for the maximum flow rate was measured to be 170 kPa (25 psi). The compressor's controller is able to hold the pressure within 7 kPa (1.0 psi) of the set value.

The dilution air supply system consists of a 5-centimeter (2-inch) pipe line with a ball valve followed by a standard in-line air filter and a smooth-edged orifice. Between the orifice and the pressure regulator is a static pressure probe with a manually readable gage and a thermocouple.

The flow rate of the dilution air in the tunnel is controlled by the dilution-air ball valve shown in Figure 3. The smooth-edged orifice downstream of it was calibrated and the flow rate could be determined from the pressure and temperature upstream of the orifice.

A considerable amount of noise is created by the uncontrolled expansion of the dilution air as it exits the 2-inch-diameter compressed air line to enter the dilution tunnel. An air-exhaust muffler was placed on the end of the compressed air line and the noise was reduced to a tolerable level.

A fairly wide range of dilution ratios (ratio of dilution-air mass flow rate to exhaust mass flow rate) can be achieved with the dilution tunnel. The maximum dilution-air flow rate achievable in the tunnel is approximately 0.91 kg/s. The engine exhaust flow rate varies from about 0.05 kg/s for low-speed, light-load conditions to about 0.11 kg/s for the full-speed, full-load condition. Therefore, dilution ratios as large as 18 can be achieved for light loads while the maximum for the full-speed, full-load case is about 8.

During steady-state operation, the velocity of the diluted exhaust in the tunnel ranges from 7.4 m/s to 12.3 m/s. These high velocities enhance the mixing process by causing highly turbulent flow with Reynolds numbers of 130,000 to 210,000. The disadvantage of the high velocities, however, is that the time for the particulates to interact with the dilution air is less than 1 second. A secondary dilution tunnel could be used to increase the residence time of the diluted exhaust.

The ability of the dilution tunnel to thoroughly mix the exhaust and air was found to be quite good. Both horizontal and vertical sampling traverses were made across the tunnel and uniform measurements were found.

The EPA specifies the temperature of the diluted exhaust to be 52 C or less. The dilution tunnel is capable of meeting this criteria for engine operating conditions of no more than 50% of full load.

PARTICULATE SAMPLING SYSTEM

A schematic diagram of the particulate sampling system appears in Figure 4. The system consists of a ball valve to keep the diluted exhaust out of the system between tests, the particulate filter, a sample pump, and a gas meter with an electronic counter. The electronic counter is capable of displaying the total amount of gas that has passed through the system during a test as well as the instantaneous flow rate. The sample pump has a by-pass line with a valve. This by-pass valve provides a means for manually adjusting the flow rate through the particulate filter. This is necessary since the flow rate will tend to drop as the filter becomes loaded with particulates. An indication of the pressure drop across the particulate filter is given by the vacuum gage in the by-pass line. The vacuum pump shown in Figure 4 is used to check for system leaks.

As mentioned in the previous chapter, to accurately measure particulates during a transient test the flow rate of the particulate sampling system must be a constant fraction of the total diluted exhaust flow rate. However, for this system, such proportional sampling is not possible. The problem is that the diluted exhaust flow rate varies in the dilution tunnel as the exhaust flow rate varies while the volume flow rate of the particulate sampling system is held constant. This produces a variation in the sampling fraction but since this will be a systematic error, it should not diminish the repeatability of the measurements.

While using higher dilution-air flow rates minimizes this non-proportional sampling error, less particulate can be collected from the more highly diluted exhaust. Also, the concentrations of the gaseous emissions will be lower for more highly diluted exhaust. Thus, the random errors associated with measuring the emissions will become larger percentages of the measurements themselves.

Since the volume flow rate is held constant in the particulate sampling train, temperature fluctuations in the diluted exhaust during the transient test cause the mass flow rate in the sample train to vary. These variations are minimized since the sample volume flow rate is adjusted according to the flow rate measured by the gas meter at the end of the sample train. As the sample travels through the line, heat transfer with the surroundings brings the temperature of the sample closer to the ambient temperature, thereby eliminating some of the variation.

FIGURE 4.

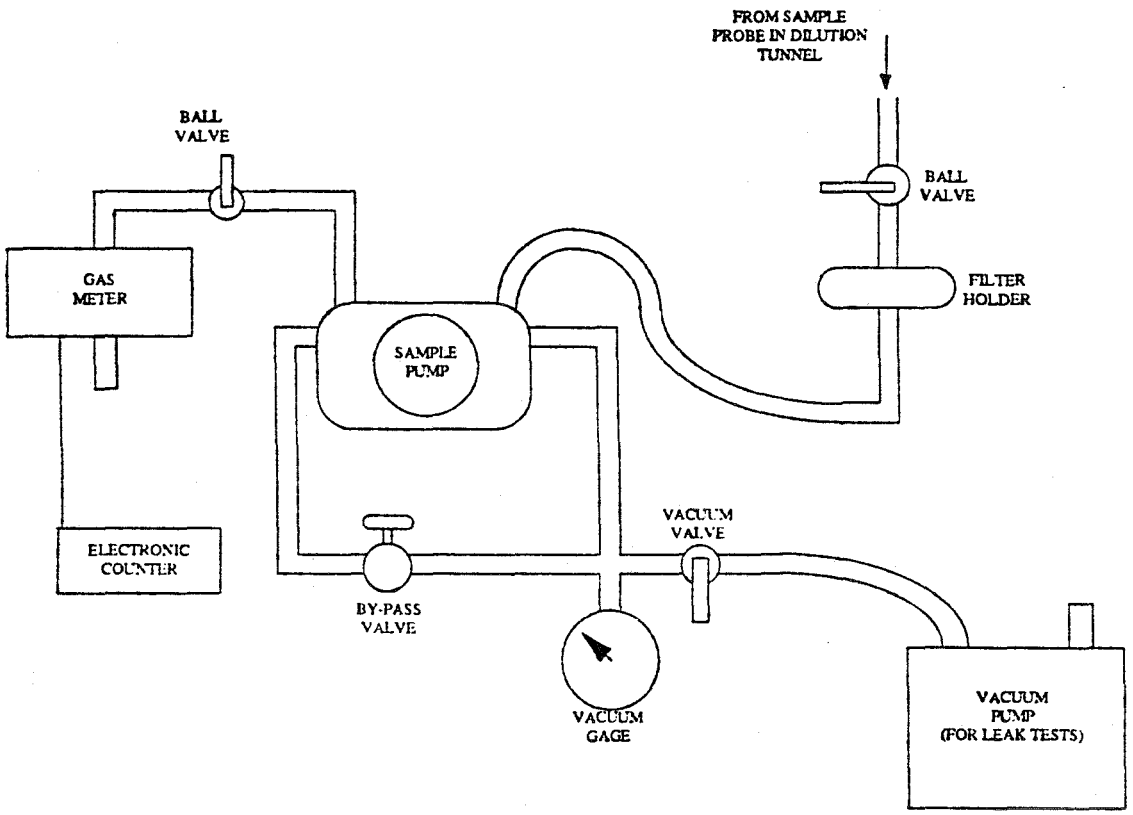

Fig. 4 Schematic diagram of particulate sampling system

To minimize the impact of random errors in particulate filter weighing, it is desirable to obtain a large sample of particulate. Given a time constraint for each test, a large flow rate through the particulate filter must be used. The sample volume flow rate used for the tests in this study was 0.0033 cubic meters per second (7 cfm). This rate was chosen because it was the largest flow that could be maintained through the filter for a 20-minute test with the engine operating in a high particulate-producing condition.

ALDEHYDE MEASUREMENT SYSTEM

Although they are not currently regulated by the EPA, aldehydes are a class of chemical species that are present at low levels in diesel engine exhaust. However, when alcohol fuels are used the aldehyde emissions tend to increase because the aldehydes are easily formed from partially burned alcohol. The simplest aldehyde, formaldehyde, is most easily formed from methanol but is always present in small amounts in diesel exhaust. The next higher level aldehyde is acetaldehyde and is most easily formed from ethanol.

The aldehyde level in the exhaust of the engine was measured using the MBTH method. This technique uses 3-methyl-2-benzothiazolone hydrazone hydrochloride (MBTH) to collect the aldehyde compounds and then an oxidizing solution to form a blue dye that can be measured using a spectrophotometer at 628 nm. The procedure used is identical to that recommended by the American Public Health Association [5] and is briefly described below.

Samples of the diluted exhaust from the dilution tunnel and the dilution air before exhaust is added are bubbled through a 0.05% aqueous MBTH solution. The dilution air represents a blank sample and the aldehydes in the diluted exhaust sample will be compared to this blank. Fifty ml of MBTH solution split between two bubblers in series is used to absorb the aldehyde from the sample and an identical arrangement is used for the blank. The flow rate through the bubblers is about 400 cc/min. After the test, the solution from the two bubblers is combined and distilled water is added to recover 50 ml of solution. Five ml of this solution is placed in a 13 mm x 100 mm pyrex test tube and 1 ml of the oxidizing reagent is added. This oxidizing reagent contains 1.6 g of sulfamic acid and 1.0 g of ferric chloride dissolved in 100 ml of distilled water. After waiting about 20 minutes for the reaction to proceed to completion, the transmissivity of the sample is measured using a Bausch and Lomb Spectronic 20 spectrophotometer. The blank sample is used as a standard for 100% transmittance.

The technique was calibrated using chemically pure acetaldehyde. A calibration was also performed using formaldehyde that showed the response of the technique to acetaldehyde to be

63% that of formaldehyde. However, since ethanol was the alcohol to be used for the tests, it was expected that most of the aldehyde present in the exhaust would be acetaldehyde. The calibration allowed the transmissivity of the sample to be converted to a concentration of aldehyde. Then, knowing the amount of sample, the sampling fraction from the tunnel, and the tunnel flow rate, the total mass of aldehyde emitted by the engine during the test could be determined. This was expressed as a brake specific value by dividing by the total work of the cycle.

ENGINE TEST SETUP

The engine that was used in this study was a John Deere four-cylinder, four-stroke model 4276T turbocharged diesel engine. The basic engine specifications are presented in Table 2. The engine is connected to a General Electric DC dynamometer. A control program is run on a Digital Equipment PRO-380 computer which sends speed commands to the dynamometer controller and torque commands to a linear actuator attached to the fuel governor lever. The control program is described in more detail below.

The volume flow rate of air into the engine was measured using a Meriam Laminar flow element in conjunction with a Baratron pressure transducer to measure the pressure drop across it. The flow rate of diesel fuel was measured using a stopwatch and a Toledo electronic scale.

Table 2. Specifications for John Deere 4276T Engine

Bore	106.5 mm
Displacement	4525.2 cm
Compression Ratio	16.8
Maximum Power	57.1 kW at 2100 rpm
Peak Torque	305.0 Nm at 1300 rpm

Dry bulb and wet bulb thermocouples were located at the inlet to the laminar flow element attached to the intake of the engine. Also, thermocouples were used to measure the engine oil and coolant temperatures. A thermocouple located at the fuel pump was used to measure the temperature of the fuel being supplied to the engine. Lastly, a thermocouple was located about four inches downstream of the turbocharger to measure the exhaust temperature. The barometric pressure was measured with a Datametrics Barocel pressure sensor.

Hydrocarbon measurements were made using a Beckman Model 402 Heated Flame Ionization Detector type hydrocarbon analyzer which includes a 3.5-meter long electrically heated sample line. A Beckman Model 955 chemiluminescent nitrogen oxides analyzer was used to measure the nitrogen oxides emissions. Beckman Model 864 non-dispersive infrared radiation analyzers were used to measure carbon dioxide and carbon monoxide.

A Digital Equipment ADM data acquisition module and a Digital Equipment PRO-380 computer were used to acquire the output of the four analyzers as well as the laminar flow element's pressure transducer. The voltage inputs were converted to units of ppm for the HC, NO_x and CO and percent for the CO_2 and displayed on the screen at the sample rate of 0.5 Hz for the steady-state tests and 1.6 Hz for the transient tests. These readings were also stored on the computer's hard disk for later use.

For the collection of particulates, 110 mm Pallflex T60A20 filters were used. The filters were weighed using a Christian Becker Model EA-1AP mechanical microbalance. The sensitivity of the balance was 0.0001 grams. A plexiglass desiccator was used for drying the filters before weighing. The desiccator measures 35 cm x 94 cm x 71 cm and is subdivided into two chambers each with their own door and each housing about 5 lbs of Drierite desiccant. A Scientific Glass Instruments four-inch, glass filter holder was used in the particulate sample train.

COMPUTERIZED DYNAMOMETER CONTROLLER

The transient engine test described by the EPA's Federal Transient Test Cycle consists of a precise series of engine speeds and loads at which the engine must be operated according to a time schedule [1]. The cycle contains periods of rapid acceleration and deceleration, idling, and steady cruising. The cycle is intended to allow simulation of the actual conditions that an engine will encounter in a vehicle. During the 20 minute test, under both cold and hot cycle operation, the engine/dynamometer pair should be able to follow reference torque and speed trajectories within the specified tolerances.

Figure 5 shows a schematic of the apparatus associated with the dynamometer controller. Independent speed and torque controllers were used. The John Deere 4276T engine was coupled to a General Electric TLC-2544 DC electric dynamometer. Speed control was provided by a General Electric Siltron Dynamometer Controller. A Digital Equipment Corporation (DEC) PRO-380 computer with a data acquisition module was used to supply reference speed and torque trajectories and to accomplish the closed loop torque control. A step-motor driven Jasta Inc. PULSE POWER PP-125 linear actuator was used to regulate the throttle position. Dynamometer and actuator specifications are given in Table 3.

Table 3. Dynamometer and Actuator Specifications

Dynamometer (GE TLC-2544):	
Excitation voltage	250.0 V
Maximum current	410.0 A
Maximum absorbed power	111.8 kW
Maximum delivered power	89.5 kW
Actuator:	
Maximum thrust	28.1 N
Maximum linear velocity	38.1 cm/sec
Stroke	7.6 cm

A simplified model was developed for the throttle-torque system. In order to minimize its complexity, only the dominant phenomena were modeled. The system was first modeled in the continuous time domain and transformed to the Laplace domain. Parameters of the linearized model were found by off-line parameter identification techniques. Discrete-time domain equations were obtained using the Z-transform. Smith's method was employed and parameters of the controller were obtained by a closed loop pole assignment technique [6]. The development of the dynamometer controller has been documented in Reference 7.

The procedures used to operate the equipment described above will be discussed in the next chapter.

FIGURE 5.

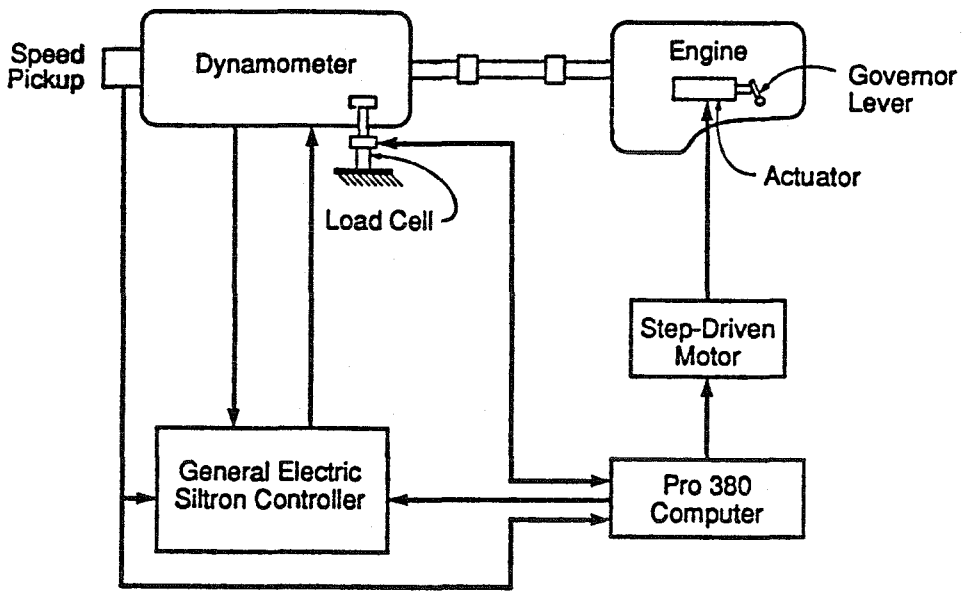

Fig. 5 Apparatus for the torque-speed control loops

CHAPTER 3. EXPERIMENTAL PROCEDURE

In this study, both steady-state and transient tests of the dilution tunnel were conducted to investigate the repeatability of the measurement technique. The procedures used to conduct these tests are described in this chapter.

STEADY STATE TEST PROCEDURES

The steady-state tests were twenty minutes in length. This length was chosen so that the particulate filters could accumulate a sample of at least 10 mg. The measurement error of the microbalance used to weigh the filters was 0.3 mg. Thus, for a sample of at least this size, the error in each weighing is no greater than 3%. The gaseous emissions were also sampled for twenty minutes since it was desired to measure all of the emission species for the same period of time.

The procedure followed to conduct the steady-state tests in this study started with placing the particulate filters to be used into the desiccator. The purpose of the desiccator was to remove most of the moisture from the filters so the mass of the filter material alone could be determined. After 48 hours, the filters were removed from the desiccator and weighted with the Christian Becker microbalance. The approximate length of time between removing a filter from the desiccator and obtaining a reading of the microbalance was 45 seconds. It was assumed that in that amount of time, a negligible amount of moisture would be reabsorbed from the atmosphere. The relative humidity in the room containing the microbalance was consistently between 53% and 56%.

Once the tare weights of the filters were determined, the engine tests could be conducted. Before starting the engine, the gaseous emission analyzers were calibrated, a particulate filter was placed in the filter holder shown in Figure 4, and the data acquisition program was started. Regarding instrument calibration, the instruments were calibrated just prior to the tests, and the calibration was checked at the end of the nine-hour schedule. An all subsequent days, the instrument calibration was checked every three hours and recalibrations were done as necessary.

When the engine was started, the speed and load were immediately set at 1400 rpm and 50% rated load. The flow rate of the engine's intake air was viewed on the screen of the data acquisition computer and used to set the dilution air orifice pressure needed for a dilution ratio of 10.

Five minutes after the engine was started, the first measurement began. The data acquisition system automatically recorded the emission analyzer readings and engine inlet-air flow rate. The particulate sampling system was started by opening the sample line valve shown in Figure 4 and then immediately starting the sample pump. The electronic counter shown in the same figure displayed the sample flow rate. During the test, the sample pump bypass valve was adjusted as necessary to maintain the flow rate within 3% of 7 cfm.

At the beginning of the test the weight of the diesel fuel supply tank was recorded. Within five minutes after the beginning of the test, the ambient, wet bulb, fuel, oil and exhaust temperatures were recorded as well as the atmospheric pressure. At the end of the test, the particulate sample pump was turned off, the sample-line valve was immediately closed, and the weight of the diesel fuel supply tank was recorded. Ten minute intervals were allowed between each test to facilitate calibration checks, desiccant replacements and particulate filter changes. When the test schedule called for a speed or load change, the change was made in the middle of the ten minute interval such that the next test began five minutes after the change.

After the last test was finished, the particulate filters were replaced into the desiccator for 48 hours. At the end of that period, they were removed and weighed. The difference in the final weight and the tare weight was considered to be the mass of the particulate sample. These data were then combined with the sample flow rate and total tunnel flow rates to determine the total mass of particulates emitted by the engine during the twenty minute test. At all times the filters were handled using steel forceps.

The HC and NO_x data for each two-second data-acquisition sampling interval were combined with the total tunnel flow rate for the same interval and totaled over the twenty minute period to determine the total mass of each emission emitted during the test. The change in calibration of the analyzers was recorded prior to each calibration of the analyzers and after the completion of all of the tests each day. This information was then used to correct the data for errors due to calibration drift. To do so, a linear change was assumed between the initial and final calibrations. A typical calibration drift was 2% of the span-gas concentration.

While the particulate, HC and NO_x data are reported as the total masses emitted by the engine during the twenty-minute test, the CO_2 results were normalized to brake specific CO_2 ($BSCO_2$) values by dividing by the total engine work of the test. Brake specific fuel consumption (BSFC) values were calculated from the change in the fuel supply tank weight and the total engine work of the test. Finally, the average equivalence ratio of each test was calculated from the fuel consumption and average inlet-air flow rate.

TRANSIENT TEST PROCEDURES

For the transient tests, the procedure specified in the Code of Federal Regulations (CFR) [1] was followed as closely as possible. The procedure specified for the Federal Test Procedure (FTP) transient test is to cold-soak the engine by shutting it off for a minimum of 12 hours or until the oil temperature reaches 75 F. In this study, the cold-soak periods were at least 21 hours in length resulting in an oil temperature of 84 F. Then the engine is started and immediately controlled according to the twenty-minute FTP schedule of speeds and loads. However, in this work, due to the nature of the control program, an extra minute of idle (1200 rpm, 0% load) condition occurred between the starting of the engine and the beginning of the transient control. Another one minute of idle operation occurred at the end of the test.

Then, as specified in the CFR, the engine remained off for twenty minutes. Following this period, known as the "hot-soak," the engine was restarted for the hot-start test, and following the one minute idle period, controlled according to the same schedule of speeds and loads. Once the schedule was completed and the extra idle period finished, the engine was shut off and the FTP test was complete. However, several hot-start tests were run consecutively thereafter. Between each of the tests a twenty minute hot-soak period was scheduled.

The sample valve was opened at the same time as the engine was started for each test, and the sample pump was turned on immediately thereafter. The sample flow rate was maintained within 3% of 7 cfm for the whole test via the sample pump by-pass valve. The pump was turned off at the completion of the transient schedule, before the one-minute idle period, and the sample line valve closed. The particulate filters were dried and weighed in the same manner as described above for the steady-state tests.

The total mass of the particulate emitted by the engine during each test was determined from the sample mass, the average sample flow rate and the average total dilution tunnel flow rate. The total masses were then normalized to brake specific particulate values by dividing by the total engine work of the test.

The gaseous emission analyzers were calibrated prior to the cold-start test and before each hot start test. A correction was applied to the gaseous emission data by using the average of the initial and final calibrations for each test. The gaseous emissions data were recorded from the beginning of the transient test schedule to the end, not including the two extra one-minute idle periods.

In order to properly determine the total amount of each emission produced during the test, the delay times between the exhaust leaving the engine and being measured by the analyzers was taken into account. The delay time for each analyzer was determined by introducing a calibration gas at the sample probe and measuring the length of time required for the analyzer to reach 90% of its final reading. Then, assuming a typical exhaust flow rate and corresponding tunnel flow rate the length of time required for the exhaust to travel from the engine exit, through the exhaust pipe, and down the tunnel to the sample probes was calculated and added to the sample line/instrument delay.

The total amount of each of the gaseous species for a given test was determined by multiplying the concentration data provided by the emission analyzers by the total tunnel flow rate for each data acquisition-sampling interval, taking into account the delay times. The results were then normalized to brake-specific values by dividing by the total engine work of the test.

CHAPTER 4. DISCUSSION OF RESULTS

This chapter provides a discussion of the effect of alcohol fumigation on diesel exhaust emissions. Initially, some results of steady state testing will be presented that provide information about the amount of alcohol required to reduce emissions at steady state and the effect of changing the formulation of the injected alcohol. Then the strategy used for injecting the alcohol during transient cycles and the current status of the transient testing will be discussed. As mentioned above, the primary focus of this study was to investigate transient cycle behavior so most of the steady state data presented was obtained during an earlier study on this engine. However, it does answer the central questions of operating cost and alcohol formulation.

EFFECT OF PERCENT TORQUE REPLACEMENT

Figures 6 and 7 show the effect on the gaseous emissions of varying the fraction of the engine's torque that is replaced by alcohol fumigation. The fraction of the brake torque provided by the ethanol is varied from 10% to 30%. The two figures show the emissions at 100% and 50% of full load at 2100 rpm. The unburned hydrocarbon (HC) and carbon monoxide (CO) emissions were essentially independent of proof except for a slight increase at the lowest proofs. The oxides of nitrogen (NO_x) emission increased consistently as the proof was increased. The HC and CO increased substantially above their diesel-only values and NO_x decreased substantially below its diesel value as the torque replacement was increased from 10% to 30%. For the range of torque replacements studied here, there was no evidence of a decrease in the impact of the fumigation on the emissions from the engine. It would appear that increasing the fraction of the fuel provided by the ethanol would simply continue to increase the HC and CO emissions and decrease the NO_x emission. Since the main objective of this study was to investigate the use of alcohol fumigation as an emission control technique, the rapid rise in HC and CO emissions was considered to be undesirable and would probably disqualify the use of higher torque replacements.

Based on Figures 6 and 7, the optimum proof for the injected alcohol is about 80 proof. Proof levels higher than this do not give as much reduction in NO_x emissions but the HC and CO emissions rise rapidly at proof levels lower than 80 proof.

FIGURE 6.

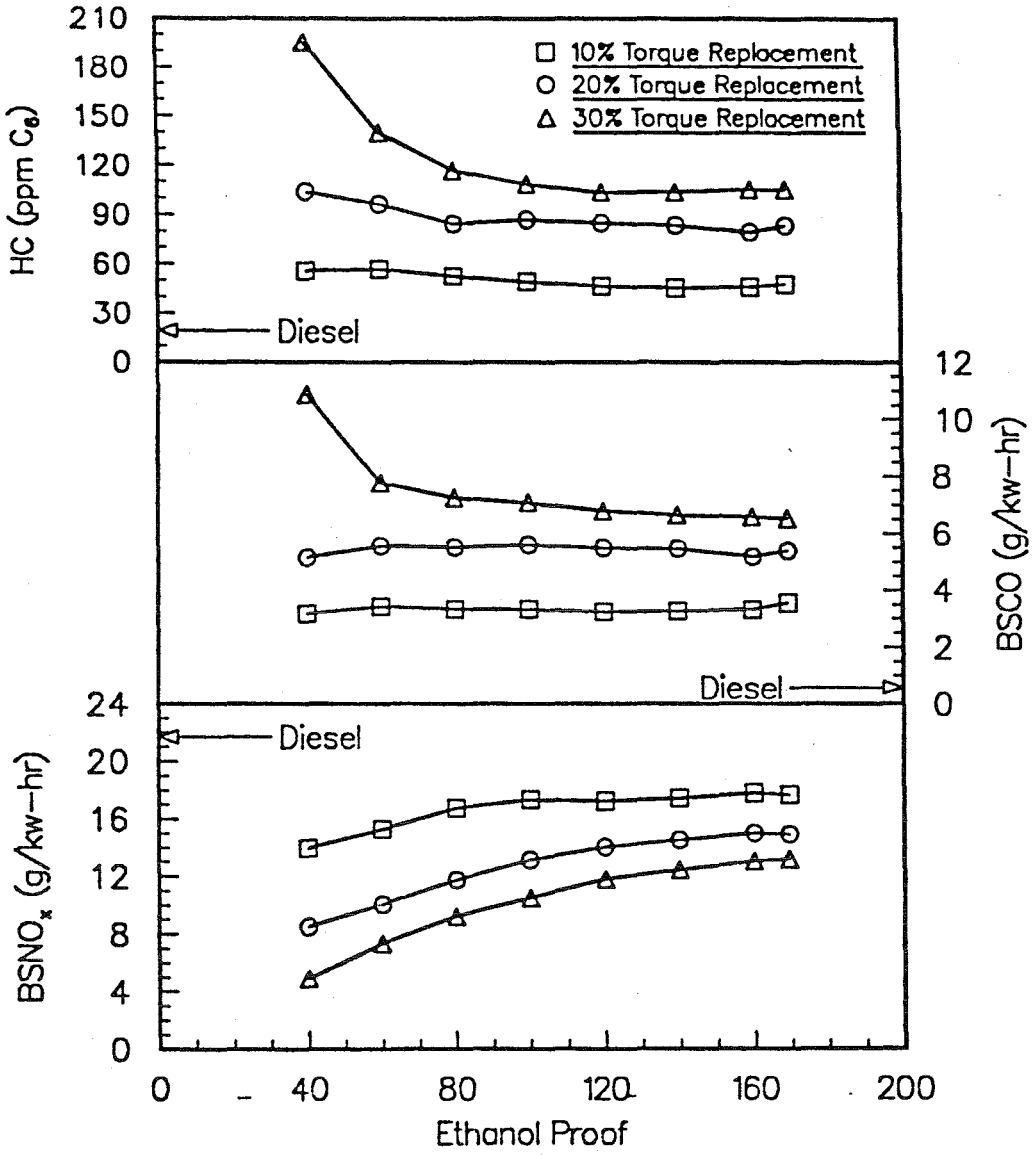

Fig. 6 Effect of torque replacement by ethanol on gaseous emissions at 2100 rpm and full load

FIGURE 7.

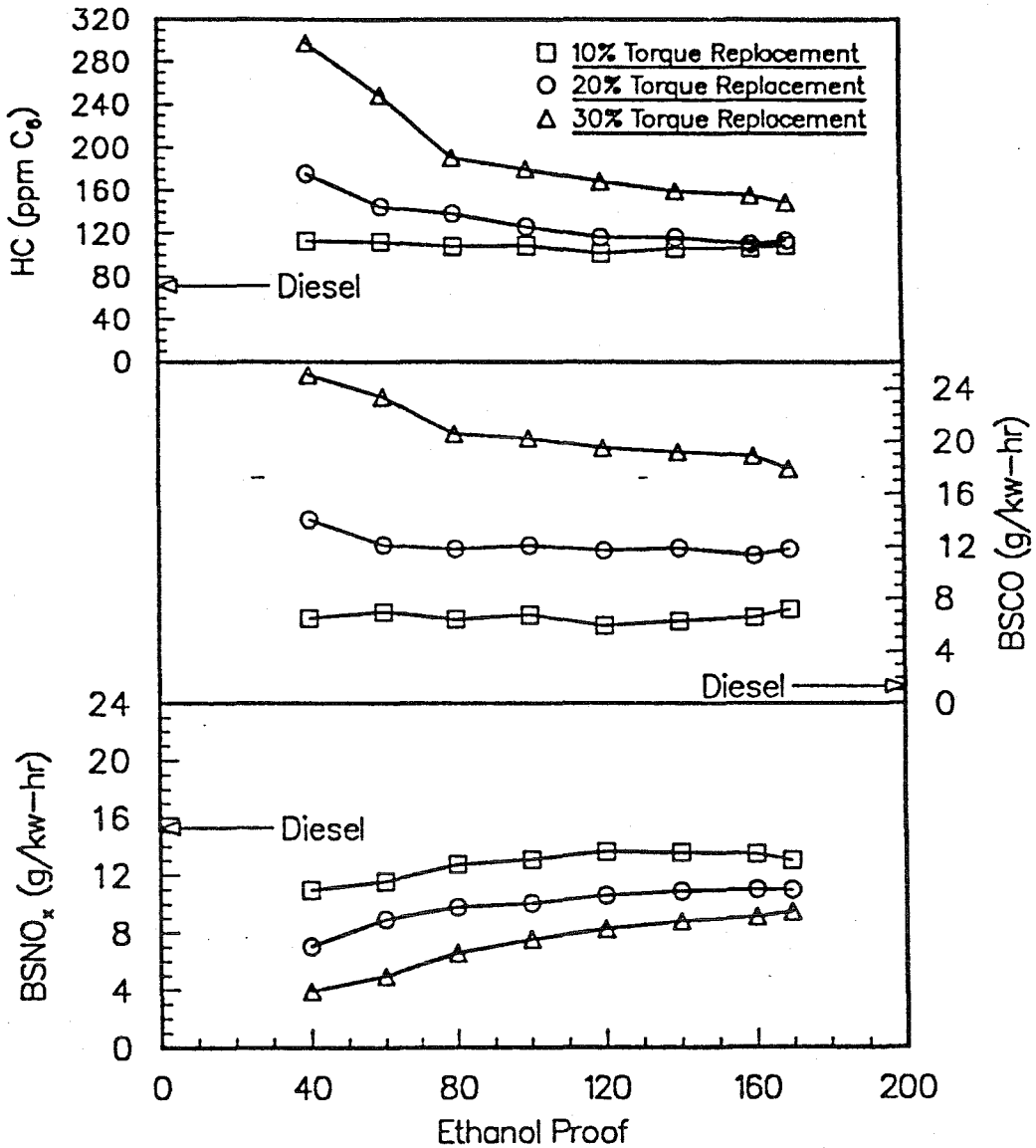

Fig. 7 Effect of torque replacement by ethanol on gaseous emissions at 2100 rpm and half load

COMPARISON OF ETHANOL AND METHANOL

Figures 8 and 9 show a comparison of the gaseous emissions from ethanol and methanol for two selected operating conditions. Figure 8 shows results from tests at 1500 rpm, at full load and 20% torque replacement, and Figure 9 shows results at 2100 rpm, at full load and 30% torque replacement. The comparisons are made using the equivalent amount of methanol in water that matches a certain proof of ethanol, both in terms of lower heating value and enthalpy of vaporization. When compared on this basis, there is very little discernable difference between the emissions from the two alcohols. The differences are within the range of uncertainty of the data. These measurements were taken by alternating between the two alcohols for each successive proof. The differences between the two fuels were so small that comparisons could not be made between data taken on separate days since the day-to-day variations were larger than the differences between the fuels.

EFFECT ON SMOKE EMISSIONS

Figure 10 shows the smoke emissions from the engine measured in Bosch Smoke Units. The figure shows the effect of varying the torque replacement at full load for 1500 and 2100 rpm. The 1500 rpm smoke levels were higher and although the change was on the same order as the variation from one proof level to the next, the smoke appeared to decrease with increasing torque replacement. The smoke readings corresponding to the 30% torque replacement were equal to or less than the values for the 10% and 20% torque replacements. The smoke levels at 2100 rpm were considerably less than at 1500 rpm. The turbocharger boost pressure level was considerably higher at the higher speed and the fuel-to-air ratio was lower. The smoke emission at this speed was very close to the detection limit of the instrument and it was not possible to determine whether the ethanol fumigation decreased the smoke emissions.

Although smoke measurements were taken at each steady state operating condition, the smoke levels were always so low that it was not possible to draw general conclusions about how the smoke levels were affected by the ethanol fumigation.

TRANSIENT CYCLE TESTING

STATUS

Development of the alcohol injection system and the aldehyde measurement system required more time than was anticipated. For this reason, testing of the fumigation system under transient conditions was not able to proceed to an optimum strategy. However, some results are available

FIGURE 8.

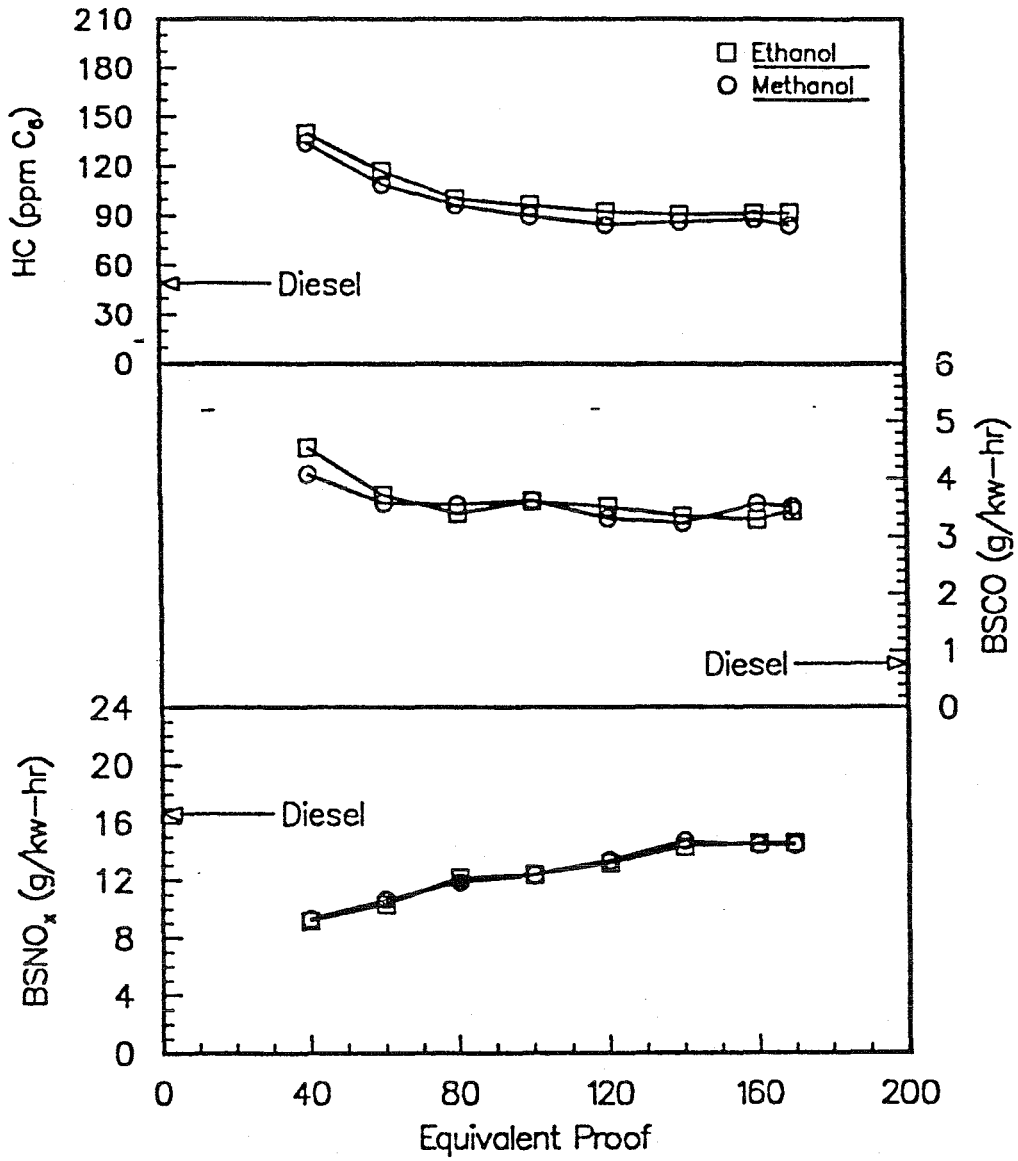

Fig. 8 Comparison of gaseous emissions from ethanol and methanol fumigation at 1500 rpm, full load and 20% torque replacement

FIGURE 9.

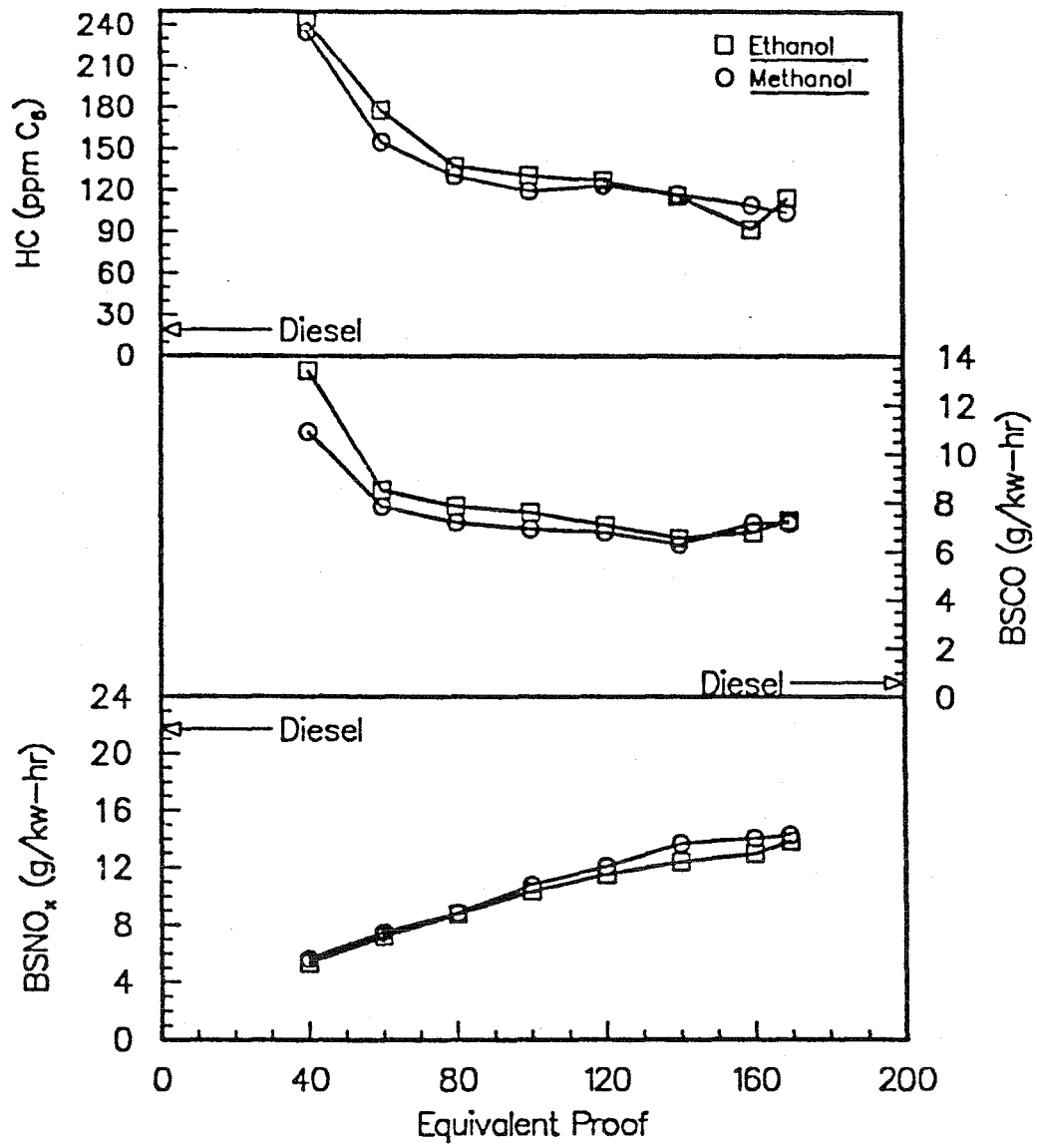

Fig. 9 Comparison of gaseous emissions from ethanol and methanol fumigation at 2100 rpm, full load and 30% torque replacement

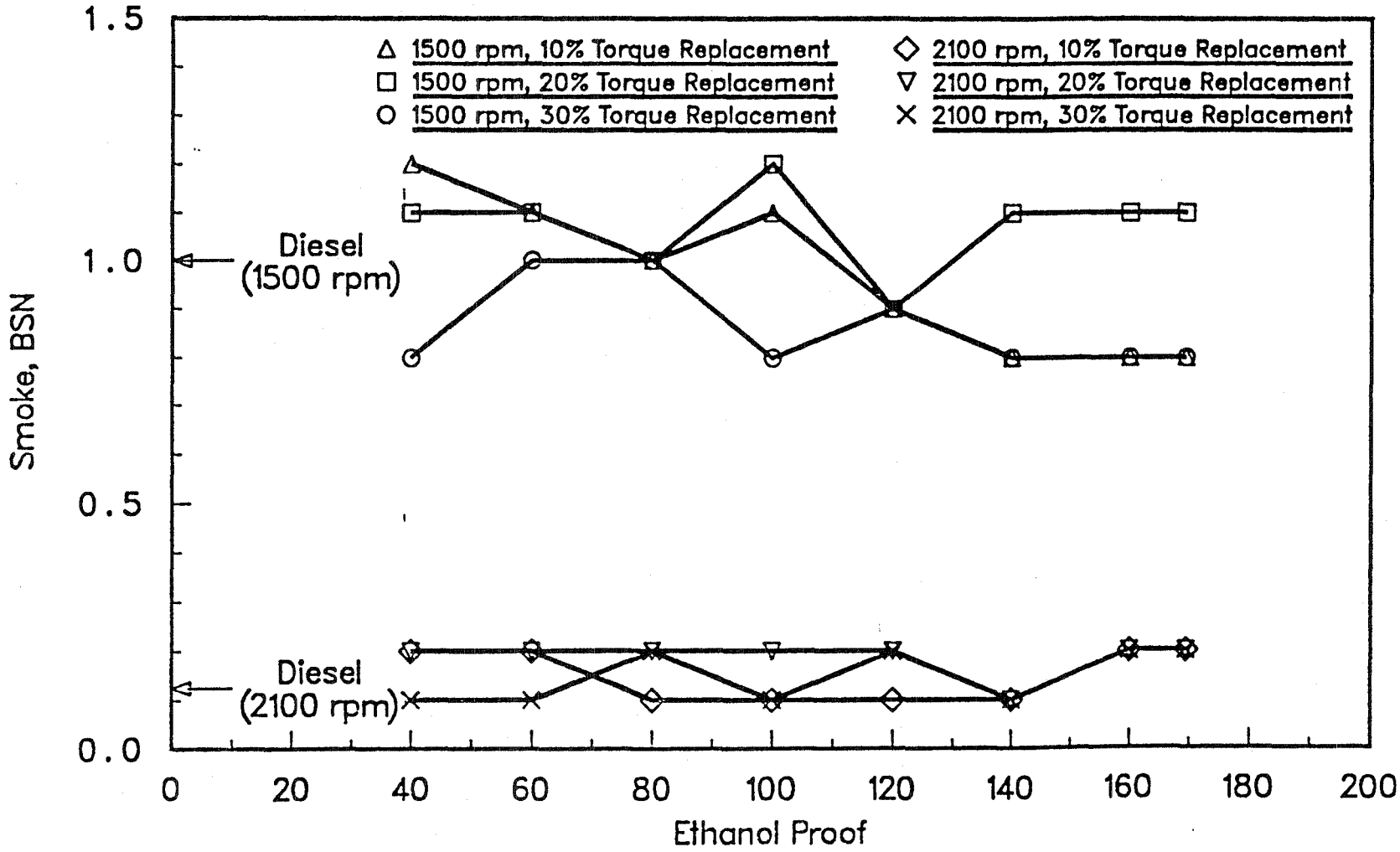

FIGURE 10.

Fig. 10 Effect of torque replacement by ethanol on smoke emission at full load for 1500 rpm and 2100 rpm

and they do allow some general conclusions to be drawn. In order to understand the different conditions used for the tests, it is first necessary to explain the strategy used to inject the alcohol.

As mentioned above, a fumigation system was obtained from Midwest Power Concepts for use in this study. This system uses the pneumatic pressure provided for actuating the diesel injection pump rack in buses to control the amount of alcohol injected. This pressure is regulated by the position of the accelerator pedal of the bus driver. Since we did not have a pneumatic throttle actuator on our engine, a simple voltage-to-pneumatic convertor was installed to allow our control computer to issue a voltage command to the convertor and have the appropriate pressure signal sent to the alcohol injector. In order to determine what pressure to send to the injector, the computer used the following rule:

$$P = K1\ [K2(rpm) + K3(torque) + K4(d(rpm)/dt) + K5(d(torque)/dt) + K6(rpm \times torque)]$$

It can be seen that K2, K3, K4, K5, and K6 are weighting factors on the speed, the torque, the speed derivative, the torque derivative and the power. These quantities have previously been normalized so only the relative magnitudes of the weighting factors are important. K1 is an overall multiplier that determines the total pressure (or total amount of alcohol). By changing the values of K1 through K6, it is possible to change the strategy of how the alcohol is injected.

This rule allows a high degree of flexibility in setting a control strategy for injecting alcohol. If K3, K4, K5 and K6 were set equal to zero and K2 set equal to 1.0, then alcohol will be injected proportional to the engine speed. The maximum amount of alcohol injected is determined by selecting an appropriate value of K1. It was anticipated that the most promising strategy for controlling particulate emissions would be to inject alcohol proportional to the engine torque and the derivative of the torque. The torque was chosen because it is under conditions of high torque that the in-cylinder fuel to air ratio is highest and this tends to produce the highest smoke level. Rapid increases in the torque, producing large values of the torque derivative, can also produce momentarily high values of the in-cylinder fuel to air ratio due to turbocharger lag. Engine power (K6) was also expected to be an important parameter since it includes a speed effect that will demand more alcohol when the number of engine cycles per second increases.

Table 4 shows a summary of the transient test results to date. The first series of tests, identified as the 100 series under the test number column, shows the variation in emissions as the total amount of alcohol was increased. The alcohol quantity was varied by changing K1 while the weighting factors were held constant with K5, the torque derivative factor weighted twice as high as the power factor. All other weighting factors were set equal to zero. At K1=0.2 and K1=0.4 the pressure developed by the alcohol injector was below the boost pressure of the engine so little

or no alcohol was injected. As K1 was raised above 0.6, the ratio of alcohol to diesel increased to 1.12% and finally as high as 21.57% for K1=1.0. This is a mass ratio that includes the mass of water injected with the alcohol. The Brake Specific NO_x ($BSNO_x$), or the amount of NO_x emitted divided by the total work done during the cycle, decreased from 37.0 g/bkW-hr to 32.3 which was 12.7%.

The particulates, although somewhat irregular, apparently decreased from 2.15 g/bkW-hr to 1.54 g/bkW-hr which was 28.3%. The particulate measurements have considerable uncertainty and a high priority for future work will be to resolve this problem. A considerable increase in unburned hydrocarbons (HC) and aldehydes is also apparent. The HC emissions show a definite increase but the level of emissions may be off since the level is considerably lower than later measurements. Carbon monoxide measurements are also difficult to interpret since there is no definite trend. The problems with the CO and HC measurements are most likely due to the extremely low levels of these species in the dilution tunnel. The lowest range on our CO analyzer is 1000 ppm but the level of CO in the tunnel is 10-30 ppm. Similarly, the level of HC in the tunnel is 5-20 ppmC6 and this is low enough that background levels of HC in the air supplied by the Centac compressor become significant. Since the NO_x, particulates and aldehyde measurements are considered to be the most reliable, they will be the focus of discussion from now on.

The 200 series of tests show the effect of switching the weighting of the torque derivative and power so that the power is now weighted twice as much as the torque derivative. A reduction in the $BSNO_x$ from 34.9 g/bkW-hr to 29.7 was observed. No reduction and possibly a slight increase was observed in particulate. Aldehyde levels increased from 0.76 g/bkW-hr to 1.06.

The effect of timing advance was investigated with the 300 series of tests. In this series of tests the pressure signal sent to the alcohol injector was advanced slightly relative to the speed and torque demands sent to the engine. This simulates the effect of an anticipatory control system that could anticipate the speed and load changes before they occur. This might be implemented in practice using a time delay between the operators accelerator pedal and the engine's rack control. Timing advances of 0.4 seconds, 1.0 seconds and 1.4 seconds were tested at two alcohol flow rates. There did not seem to be any effect on NO_x emissions and a small, if any, effect on particulates. Aldehydes also did not seem to be affected by the timing advance.

The 400 series of tests was performed to investigate using the engine torque and the torque derivative as the controlling parameters for the injector. Figure 11 shows the variation in the NO_x level in the dilution tunnel during the 20 minutes of the transient cycle for runs 401 and 402. These tests are for diesel fuel-only, without alcohol and for alcohol injection with equal weighting

TABLE 4. CALCULATED TEST RESULTS

Test #	K1	K2	K3	K4	K5	K6	delT (sec)	Add/Dsl (%)	BSCO	BSNO	BSHC	Part	Alde
									|---------- g/bhp-hr ----------|				
101	0.200	0	0	0	2	1	0.0	0.23	7.19	37.00	1.03	2.15	0.38
103	0.400	0	0	0	2	1	0.0	0.38	6.20	36.10	0.93	2.31	0.58
104	0.600	0	0	0	2	1	0.0	1.12	6.11	37.00	1.08	2.36	0.61
106	0.700	0	0	0	2	1	0.0	3.83	6.65	35.40	1.34	2.13	0.68
105	0.800	0	0	0	2	1	0.0	9.44	7.88	34.40	1.74	1.97	0.78
107	0.900	0	0	0	2	1	0.0	17.89	10.63	32.90	2.77	2.11	0.92
102	1.000	0	0	0	2	1	0.0	21.57	6.84	32.30	3.11	1.54	0.97
203	0.400	0	0	0	1	2	0.0	0.30	10.97	34.90	7.82	2.11	0.76
204	0.500	0	0	0	1	2	0.0	1.00	9.62	33.60	7.99	2.09	0.72
205	0.550	0	0	0	1	2	0.0	3.85	9.41	33.30	8.28	2.12	0.88
201	0.600	0	0	0	1	2	0.0	11.10	11.06	32.20	8.83	2.30	0.87
202	0.700	0	0	0	1	2	0.0	25.10	11.14	29.70	9.68	2.20	1.06
301	0.000	0	0	0	0	0	0.0	0.00	9.47	35.90	6.17	2.44	0.56
302	0.700	0	0	0	2	1	0.4	1.91	4.65	36.00	6.38	2.20	0.72
303	0.800	0	0	0	2	1	0.4	4.85	6.69	35.30	6.65	2.25	0.77
304	0.800	0	0	0	2	1	1.0	4.89	5.15	36.00	7.22	2.30	0.82
305	0.800	0	0	0	2	1	1.4	4.70	5.82	34.40	7.10	2.22	0.87
306	0.900	0	0	0	2	1	0.4	6.91	5.04	34.60	7.40	2.22	0.95
307	0.900	0	0	0	2	1	1.0	7.38	3.96	35.70	7.56	2.24	0.92
308	0.900	0	0	0	2	1	1.4	7.97	3.79	35.40	7.50	2.16	0.95
401	0.000	0	0	0	0	0	0.0	0.00	12.07	37.00	7.81	2.35	0.74
407	0.650	0	1	0	1	0	0.0	8.10	11.98	34.40	8.33	2.31	1.02
403	0.700	0	1	0	1	0	0.0	12.00	9.54	33.90	8.76	2.29	0.94
402	0.800	0	1	0	1	0	0.0	21.40	8.86	32.00	9.54	2.23	1.09
404	0.700	0	1	0	2	0	0.0	3.10	11.12	35.80	7.81	2.25	0.87
408	0.800	0	1	0	2	0	0.0	6.50	12.02	35.00	8.24	2.27	0.97
405	0.900	0	1	0	2	0	0.0	12.30	11.07	34.00	8.96	2.29	1.03
406	0.600	0	2	0	1	0	0.0	8.60	11.05	34.10	8.29	2.31	0.95
410	0.625	0	2	0	1	0	0.0	10.70	14.98	33.90	8.49	2.01	1.01
409	0.700	0	2	0	1	0	0.0	21.70	16.55	31.50	9.36	2.34	1.22
501	0.000	0	0	0	0	0	0.0	0.00	6.75	36.30	8.18	2.62	0.67
502	0.950	0	0	0	1	0	0.0	6.70	19.08	36.10	9.36	2.39	0.96
503	0.600	0	1	0	0	0	0.0	7.90	14.25	32.50	8.48	2.14	0.93
504	0.600	0	0	0	0	1	0.0	7.40	14.68	34.00	8.28	2.30	0.89
505	0.000	0	0	0	0	0	0.0	0.00	12.27	35.90	7.70	2.21	0.83
							set maximum torque down by 10%						
506	0.000	0	0	0	0	0	0.0	0.00	6.91	35.20	6.60	1.99	0.72
507	1.000	0	0	0	1	0	0.0	7.00	5.36	34.60	8.01	2.06	0.91
508	0.600	0	1	0	0	0	0.0	3.80	4.05	34.60	7.57	1.95	0.82
509	0.650	0	1	0	0	0	0.0	9.40	3.07	34.10	8.10	2.11	0.99
510	0.650	0	0	0	0	1	0.0	8.80	3.41	33.40	7.94	2.04	0.95
511	0.000	0	0	0	0	0	0.0	0.00	1.74	35.10	7.35	1.99	0.83
512	0.650	0	0	0	0	1	0.4	8.80	2.44	33.90	8.11	2.06	0.96
513	0.650	0	0	0	0	1	1.0	7.70	1.11	34.70	8.18	2.05	0.92
514	0.650	0	0	0	0	1	1.4	7.60	1.31	34.90	8.34	2.13	0.85
515	0.000	0	0	0	0	0	0.0	0.00	0.71	36.50	7.90	2.05	0.76

on the torque and the torque derivative. The large variations in the NO_x emissions of the engine at different times in the cycle are readily apparent. Figure 12 shows the same data as Figure 11 but with the time scale expanded to more easily show the differences between the two tests. Run number 402, with alcohol, shows a noticeable reduction in NO_x during the period of sustained power occurring between 620 seconds and 670 seconds. The cumulative total for the cycle was a reduction from 37.0 g/bkW-hr to 32.0.

Figure 13 shows the instantaneous hydrocarbon emissions for the same tests as Figures 11 and 12. The dotted line corresponding to the case with alcohol injection is consistently higher than for the diesel-only case. The points where alcohol is injected are clearly indicated as spikes in the hydrocarbon emission. Figure 14 shows the same data on an expanded time scale. The alcohol injected during the sustained high power region between 620 and 670 seconds referred to above gives a high hydrocarbon emission during a period that would otherwise be quite low. It is clear that the reduction in NO_x emissions is obtained at the cost of a rise in HC emissions.

The particulate emissions did not seem to be affected by the changes in alcohol flow rate for the 400 series of tests. As expected, aldehyde emissions increased as the alcohol flow increased.

The first portion of the 500 series of tests was performed to investigate the effect of weighting single terms in the pressure equations. Test number 502 used K5, the torque derivative term to control alcohol injection, test number 503 used the speed derivative and test number 504 used the power. The best reduction in NO_x and particulate was obtained when the alcohol injection was proportional to the torque.

The second portion of 500 series tests involved a change to the diesel engine. Although the alcohol injector has the potential to inject alcohol equal to that required to produce 30-40% of the total engine torque, since the diesel engine can provide power very quickly, it may not be utilizing the alcohol to meet load increases. To address this question, we adjusted the torque screw on the engine's injection pump to reduce by 10% the maximum torque of the engine. After doing so, the engine was still able to meet the torque specifications of the EPA transient test cycle although it was relying on the energy in the alcohol to do so.

At the reduced torque condition there is a definite reduction in particulate emissions. Even for relatively small amounts of alcohol, there is a reduction in particulate on the order of 10%. NO_x emissions are reduced also compared to the higher torque case although the further reduction due to the alcohol injection is not so apparent.

To finish the 500 series of tests, the effect of timing advance was again investigated with the power used to control the alcohol flow rate. Again, no significant effect of advance was shown on particulate or NO_x.

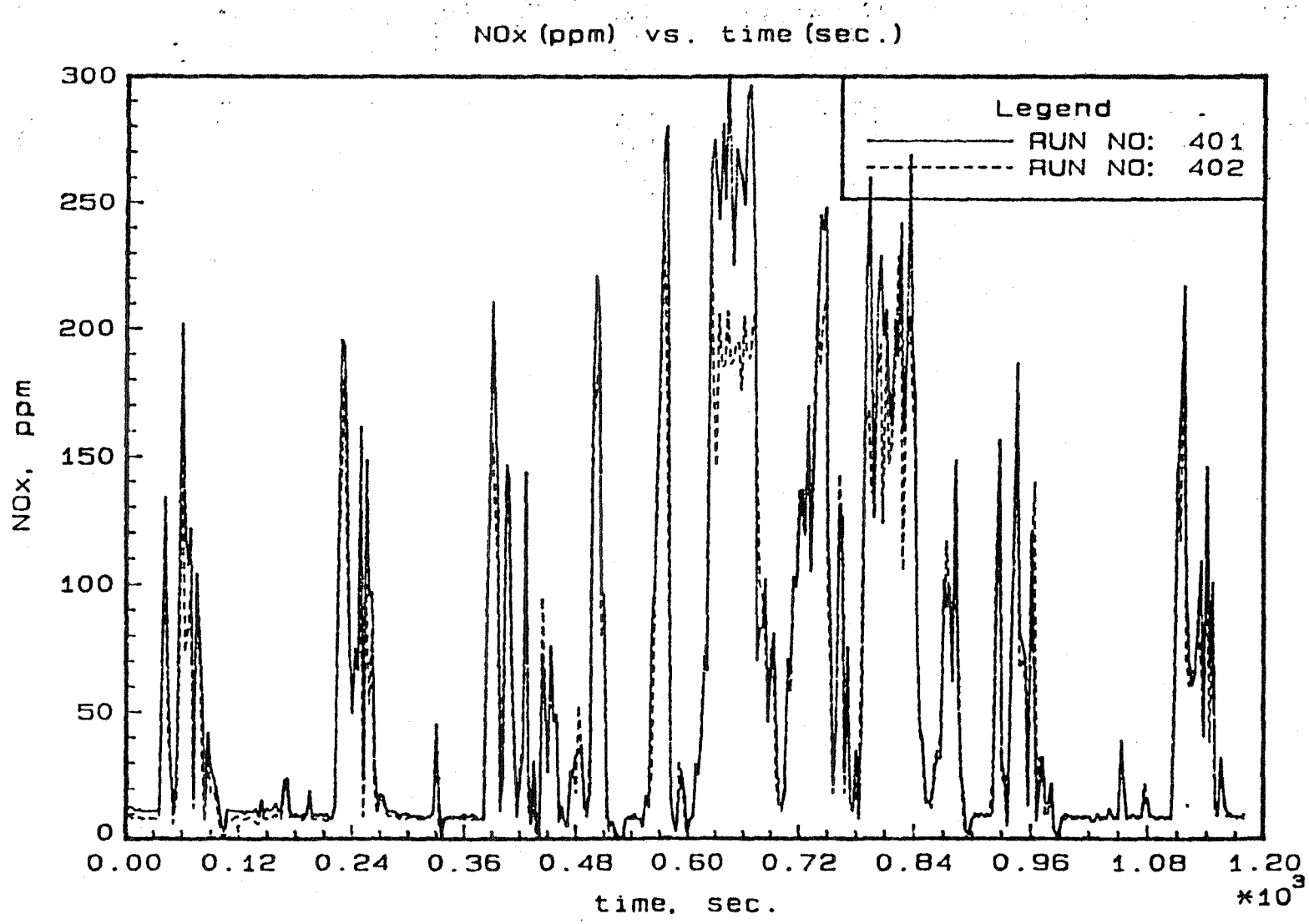

Fig. 11 NOx emissions for the transient cycle

FIGURE 11.

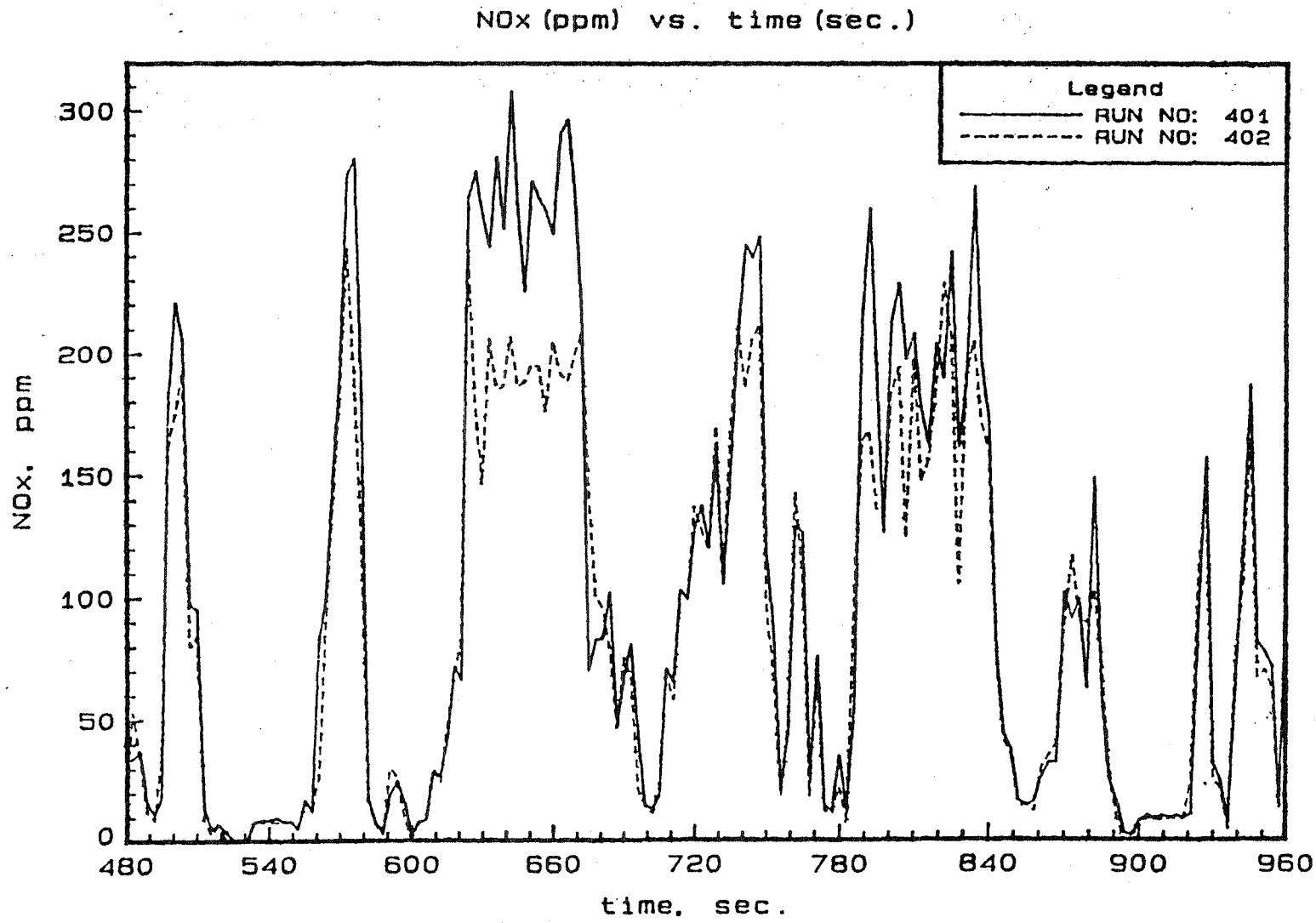

Fig. 12 NOx emissions for a portion of the transient cycle

FIGURE 12

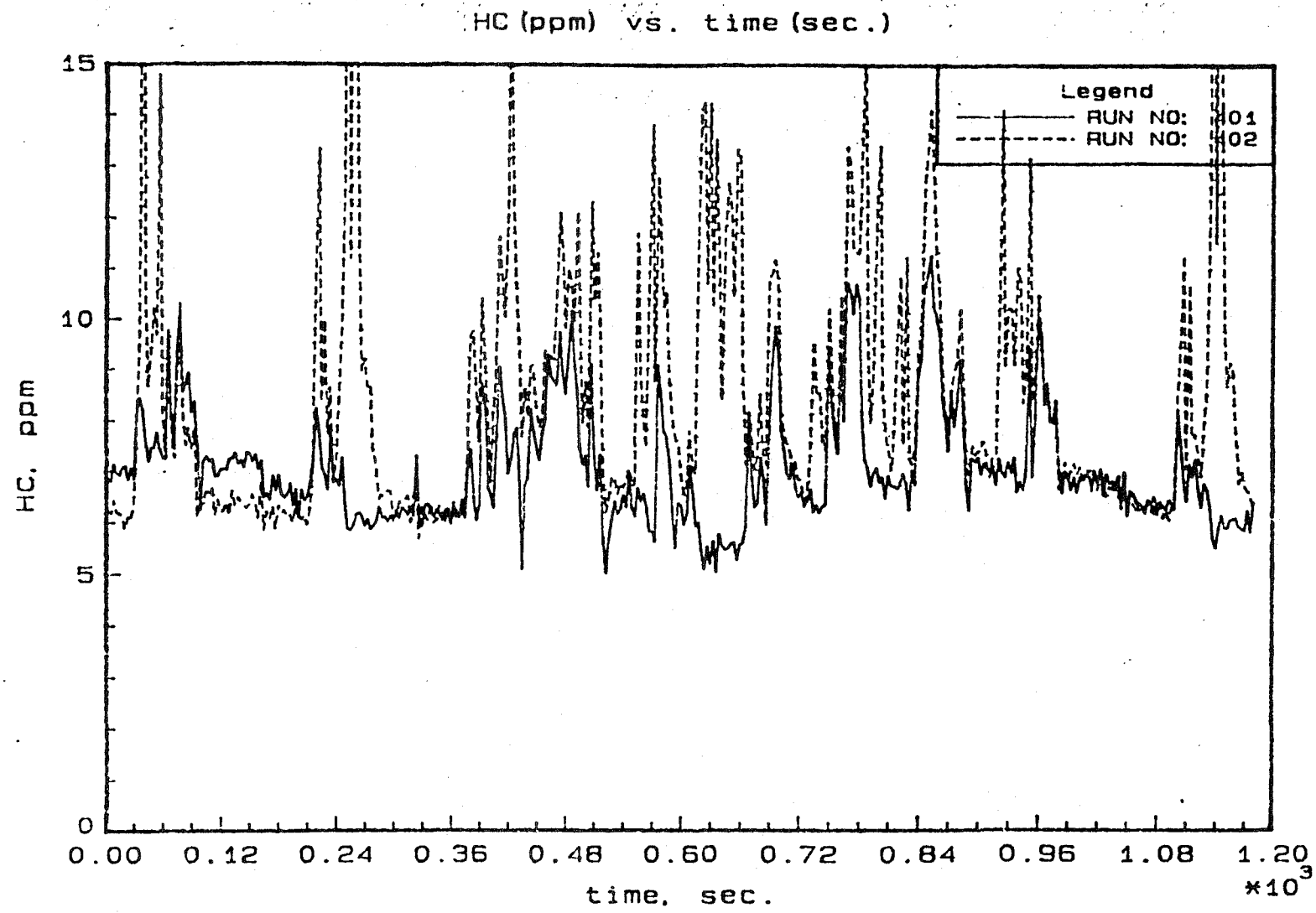

Fig. 13 Unburned hydrocarbon emissions for the transient cycle

FIGURE 13.

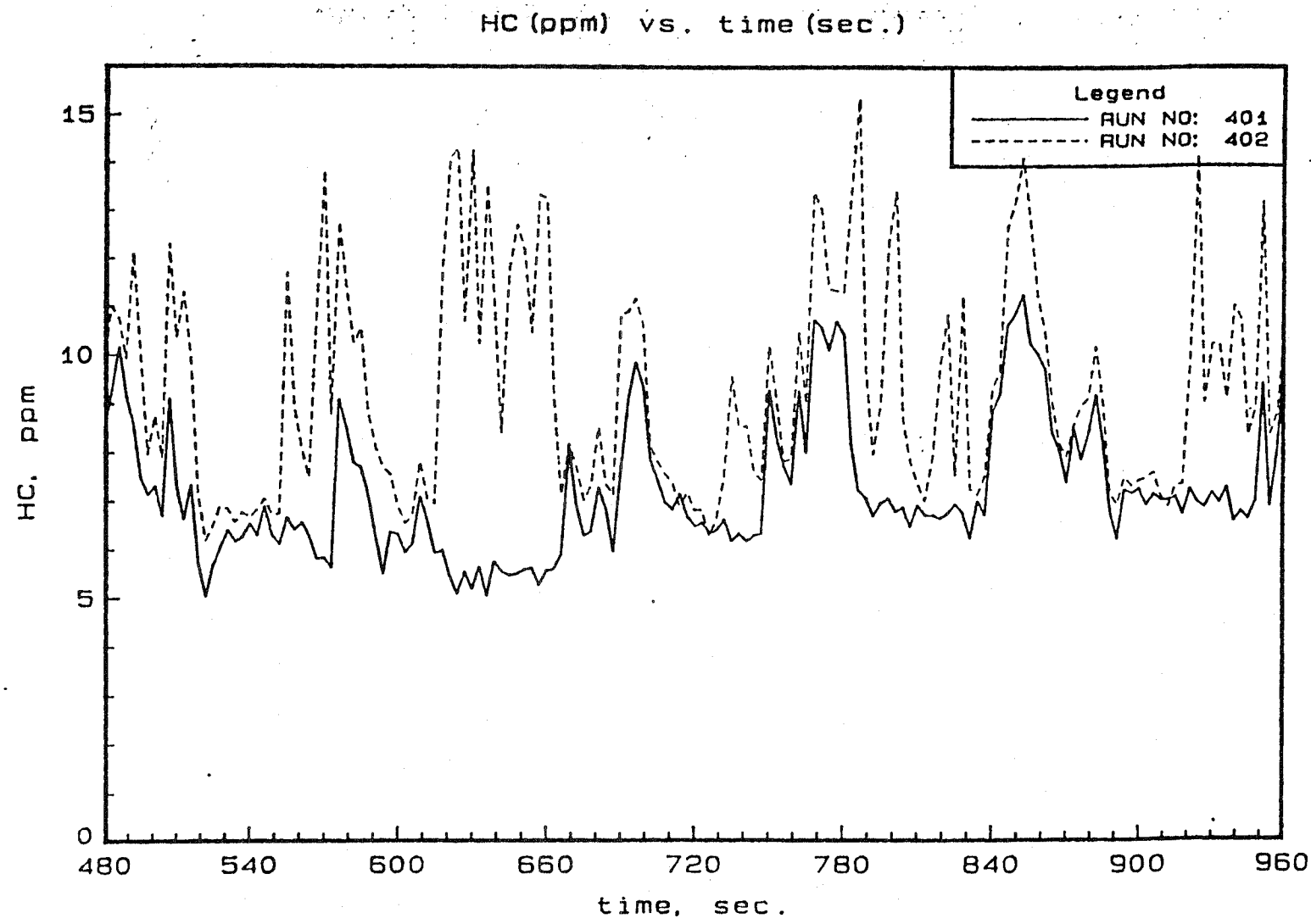

Fig. 14 Unburned hydrocarbon emissions for a portion of the transient cycle

FIGURE 14.

CHAPTER 5. CONCLUSION

One of the objectives of this project was to provide students with exposure to problems and technology in the transportation area. We were very successful in this regard. Four graduate students, Bennett Murray, Taner Tuken, Qiqing Jiang and Pradheepram Ottikkutti, worked on different aspects of this project. Three undergraduates also involved were Eric Hensley, William Carr and Darren Herum.

The optimum formulation for the alcohol additive that is injected into the engine appears to be about 80 proof ethanol. Higher proofs than this do not give as much reduction in NO_x emissions and lower proofs give increases in CO and HC emissions. There does not appear to be any difference between methanol and ethanol in their emissions reduction potential although a higher concentration of methanol would be required to give the same effect as 80 proof ethanol. The choice of alcohol should be made based on the cost of the alcohol.

The cost of operating an engine with alcohol fumigation depends on the amount of emission reduction desired. The steady state data indicated that the engine could be operated with up to 30% of its torque supplied by the fumigated alcohol. However, to accomplish this would require very large flow rates of the alcohol-water mixture since the 80 proof mixture is only about 33% ethanol and ethanol has only about 65% of the energy content of diesel fuel. The transient tests, although not conclusive, indicate that if the alcohol is injected only on demand, then the amount of alcohol required to achieve significant reductions in emissions may be much smaller. Typically, the mass of the alcohol-water mixture consumed during the transient test is about 5-20% of the diesel mass and this is only about 33% ethanol. Thus, it may be possible to reduce emissions using an amount of ethanol equal to 2-6% of the diesel fuel consumed. Reductions of 12.7% in oxides of nitrogen emission and 28.3% in particulates have been observed under transient conditions.

There is a definite increase in the amount of aldehydes produced by the engine when operated on alcohol. Although we did not determine which aldehyde species were present, since the alcohol used was ethanol, we expect that the principal aldehyde was acetaldehyde. The aldehyde was observed to increase by as much as 2.5 times its diesel-only value. However, since the aldehyde emissions from diesel engines have not generally been considered to be a problem, this increase does not necessarily mean that higher aldehyde emissions would preclude alcohol fumigation.

CHAPTER 6. REFERENCES

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