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Fumigation of Alcohol in a Light Duty Automotive Diesel Engine

Entezaam M. H. Broukhiyan and Samuel S. Lestz
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NOMENCLATURE

A,B,C	Constants for Antoine Vapor Pressure Equation
API	American Petroleum Institute
A/F	Air-Fuel Ratio
B(a)P	Benzo(a)Pyrene
BHP_c	Brake Horsepower Corrected
BHP_t	Brake Horsepower Test
BMEP	Brake Mean Effective Pressure
BSEC	Brake Specific Energy Consumption
C_p	Constant Pressure Specific Heat
DNA	Deoxyribonucleic Acid
dP/dt	Time Rate of Pressure Change
FID	Flame Ionization Detector
k	Reaction Rate Constant
K_s	Severe Knock Event
LTP	Lower Trip Point of Comparator
MON	Motor Octane Number
P	Pressure
Φ,PHI	Equivalence Ratio [(F/A) act./(F/A) stoich.]
PNA	Polynuclear Aromatics
PPM	Parts Per Million
R	Pressure Regulator
RON	Research Octane Number
RPM	Revolution Per Minute
SI	Spark Ignition
SOF	Soluble Organic Fraction

NOMENCLATURE (Continued)

SSU	Saybold Seconds Universal
T	Temperature
TDC	Top Dead Center
UTP	Upper Trip Point of Comparator

CHAPTER I

INTRODUCTION

1.1 Background

As world population steadily increases, so do the energy requirements of every economic sector, ranging from transportation to home heating and stationary power plants for industry. Traditionally, petroleum and its derivatives have been used in many energy applications; particularly transportation accounts for a large proportion of the petroleum used. After the OPEC imposed oil embargo in 1973, research and development work on alternative fuels became very intense. One result was a renewed interest in alcohols, which have been considered as potential automotive fuels for almost 60 years. Ethanol and methanol are of particular interest because they can be produced from renewable resources. Much early research work was devoted to the production and use of ethanol (C_2H_5OH). Oxygen comprises about 35% by mass of the ethanol molecule, providing 14% of the total oxygen required for stoichiometric combustion. Ethanol has a latent heat of approximately twice that of gasoline; it burns cleanly and produces almost no soot. Until about 1929, practically all of the United States production of ethanol was by fermentation. In recent years, most non-beverage ethanol has been produced synthetically from natural gas and petroleum. Current U.S. annual ethanol production is approximately 300 million gallons (1)*; much of this production involves hydration of ethylene, a petroleum derivative (2).

*Number in parentheses designate reference list entries.

Methanol (CH_3OH) has wide flammability limits, good lean combustions characteristics, and high flame velocity relative to gasoline. Oxygen comprises 50% by mass of the methanol molecule, providing 25% of the total oxygen required for stoichiometric combustion. Methanol's latent heat of vaporization is almost four times that of gasoline. Methanol has been traditionally produced by the destructive distillation of wood or synthetically from carbon monoxide and hydrogen. Methanol may also be derived from coal by-products; the merit of this approach lies in the fact that the United States has vast coal reserves. The current annual U.S. methanol production is approximately one billion gallons (1). An investigation by Wagner et al. shows that U.S. methanol production represents about one percent of the current gasoline consumption and the ethanol produced could only fill one-half of one percent of the gasoline demand (2). Since most of this alcohol is petroleum derived, it is evident that the U.S. cannot yet depend on alcohol as a renewable motor fuel.

There has been much research and literature devoted to the use of alcohols in spark-ignited (SI) engines, both in the neat form and in alcohol/gasoline blends. Ethanol and methanol have relatively high motor octane numbers, 107 and 106 respectively; consequently, they are excellent fuels for SI engines (3). Methanol has long been used as a racing fuel; all thirty-three cars in the 1978 Indianapolis 500 race ran on one hundred percent methanol (4). Other benefits of burning alcohols in SI engines include increased thermal efficiency and reduced exhaust emissions (5,6).

The increased need for petroleum fuel conservation in the 1970's also focused attention on the Diesel engine which is more efficient than the equivalent SI engine. The Diesel engine's high compression ratio, unthrottled intake air, and heterogeneous combustion process all combine to increase its thermal efficiency. Rising gasoline prices and government fuel economy regulations prompted special interest in Diesel applications to the light-duty market. Production of the Oldsmobile 5.7 liter V-8 Diesel engine was on the basis of this idea (7).

Comparison of the gaseous exhaust emissions of equivalent SI and Diesel engines also favors the Diesel engine; reduced carbon monoxide (CO) and unburned hydrocarbons (HC) result from the leaner-than-stoichiometric combustion process. Additionally, the Diesel engine's lower combustion temperatures contribute to reduced production of oxides of nitrogen (NO_x) (8).

Conversely, examination of particulate emissions reveals a disadvantage of Diesel engines: the production of 50 to 100 times more particulate matter than an equivalent SI engine (9). In 1977, the U.S. Environmental Protection Agency (EPA) notified the Department of Energy (DOE) that Diesel engine particulate emissions had been tested and were shown to be mutagenic and potentially carcinogenic (10). Investigation of the potential health effects of Diesel engine particulate emissions was initiated and is ongoing.

Since there are increasing numbers of light-duty Diesel engines in use, it follows that the suitability of alcohol fuels in Diesel engines should be evaluated. Further, the clean, sootless burning characteristics of ethanol and methanol suggest a possible role in reducing the Diesel engine's particulate problem. However, alcohols are not easily

burned in Diesel engines; the high octane ratings of ethanol and methanol indicate their poor autoignition tendencies. This is verified by cetane numbers, ranging from zero to five, that can be determined only by extrapolation (11,12). Holmer used additives to increase the cetane number of methanol to 35, sufficiently high to permit Diesel operation with a compression ratio of 15:1. This practice was not economical because large amounts of additives were required (up to 20% by volume) (13).

Havemann et al. attempted to use alcohol/fuel oil blends; however, the blends were unstable and separated in the presence of trace amounts of water (14). Moses et al. was able to operate unmodified Diesel engines on alcohol/fuel oil emulsions containing up to 20% ethanol or methanol (15). Pischinger reported that direct injection of methanol into the combustion chamber permitted the substitution of a large amount of methanol for fuel oil without sacrificing reliable ignition or combustion efficiency (16).

In order to reduce Diesel engine smoke, Alperstein et al. employed fumigation to introduce alternative fuels into the engine, that is, the alternative fuel was sprayed into the intake air manifold of the engine. It was shown that fumigation aided the Diesel combustion by providing better air utilization due to premixing (17).

1.2 Objectives

The purpose of this work is to study a way to utilize alcohol (ethanol and methanol) in a light-duty Diesel engine. The effect of fumigation on performance, smoke, emissions, and the biological activity of the exhaust soot will be investigated.

1.2.1 Specific Objectives

The specific objectives of this research are:

1) Establish a baseline test matrix for different engine speeds and rack settings.

2) Obtain, for each condition in the test matrix, thermal efficiency, power output, smoke, and gaseous emissions.

3) For each test point by fumigation, substitute methanol and ethanol for the fuel oil such that the total energy input remains constant. For each point, the percentage of alcohol substituted will be calculated on an energy basis. Alcohol substitution will be limited by the occurrence of severe knock or severe combustion degradation.

4) Determine the biological activity of the exhaust soot using the Ames *Salmonella typhimurium* assay.

CHAPTER II

THEORETICAL CONSIDERATIONS

2.1 Introduction

The Diesel engine because of its good fuel economy and the recent fuel shortages is becoming more and more popular, and society feels its importance growing day by day. But in contrast to its good fuel economy and ability to use a wide range of fuels, Diesel engines produce more soot than SI engines do. The EPA has made an announcement to the effect that the particulate emissions from Diesel engines may be carcinogenic and harmful (10). Now many intense studies are being done on the biological activity of the soot formed in the Diesel engine. Because of the oil embargo of 1973 and the ensuing fuel shortages, intense work was started to find alternative fuels for passenger automobiles. Hydrogen, the alcohols, vegetable oils, and distillates from coal and shale are among the non-petroleum fuels receiving attention. Alcohol substitution is of interest in the U.S. because a relatively large capacity for its production exists. Due to its high octane number, much work has been done on the use of alcohol in SI engines. Because of its growing popularity for light-duty service, many researchers have begun to investigate how to best burn alcohol in a Diesel engine. Alcohol with its high octane number and low cetane number is a good fuel for SI engines and a poor fuel for Diesel engines. Therefore, appropriate methods for introducing alcohol into a Diesel engine need to be developed.

In this work in order to study the effects of ethanol and methanol substitution, it was first necessary to establish a baseline of operating conditions using a control No. 2D fuel oil. The baseline operating conditions were organized in the form of a matrix, each cell having a rate of energy input determined by engine speed and injection-pump rack setting. The properties of methanol, ethanol and the control No. 2D fuel oil are presented in Table 2.1. Ethanol and methanol due to the presence of a hydroxyl group are polar in contrast to the non-polarity of most hydrocarbon fuels. This chapter will discuss Diesel engine combustion, alcohol chemistry and combustion, and fumigation which is one way of introducing alcohol into a Diesel engine.

2.2 Diesel Combustion

In a SI engine, a homogeneous premixed charge of fuel and air enters the cylinders where it is ignited and burning is usually characterized by the propagation of a well-defined flame front. In a Diesel engine only air is inducted and compressed. Fuel is then injected as a finely atomized spray into the hot compressed air where it simultaneously autoignites at many different points. This is called heterogeneous combustion and results in different emission products than those from homogeneous SI engine combustion. The stoichiometric A/F ratio for Diesel fuel oil, like gasoline, is about 15 to 1. Since there is no intake throttling in a Diesel engine, the amount of air inducted does not change per cycle and power output is controlled by the amount of fuel injected into the cylinder. Diesel engines operate over a wide range of A/F ratios, 20 to 1 at maximum power to 100 to 1 at idle which is always fuel lean (11). Because of the way fuel is

Table 2.1

Selected Liquid Fuel Properties
(Adapted from Obert (3), and Houser (18))

Fuel	Methanol	Ethanol	Diesel Fuel
Physical Properties			
Specific Gravity at 68°F	0.796	0.795	0.846+
Liquid Density (lbm/gal)	6.61	6.60	7.05
Boiling Temperature, (°F)@latm	149	172	376-627+
Freezing Temperature, (°F)@latm	-144	-170	-
Specific Heat, (Btu/lbm°F)@60°F	0.60	0.648	0.52
Heat of Vaporization, (Btu/lbm)	502.	396.	110.++
Viscosity at 68°F (cp)	0.595	-	0.785
Water (H_2O) Solubility	∿∞	∿∞	∿0

+ Indicates property value is from laboratory analysis of No. 2D fuel oil.
++ Indicates property value is for dodecane.

Table 2.1 (continued)

Selected Liquid Fuel Properties
(Adapted from Obert (3), and Houser (18))

Fuel	Methanol	Ethanol	Diesel Fuel
<u>Chemical Properties</u>			
Formula	CH_3OH	C_2H_5OH	-
Molecular Weight	32.04	46.06	-
Composition by Weight			
% Carbon	37.5	52.5	86-87
% Hydrogen	12.6	13.1	11-13.5
% Oxygen	49.9	34.7	∿0
Combustion Equation	$CH_3OH+1.5O_2+5.66N_2$	$C_2H_5OH+3O_2+11.3N_2$	$C_nH_{2n}+1.5nO_2+5.66nN_2$
	$\rightarrow CO_2+2H_2O+5.66N_2$	$\rightarrow 2CO_2+3H_2O+11.3N_2$	$\rightarrow nCO_2+nH_2O+5.66nN_2$
Stoichiometric Air-Fuel Ratio (lbm/lbm)	6.4	9.0	∿15.0
Moles Product Per Mole Charge	1.061	1.065	1.062++

\+ Indicates property value is from laboratory analysis of No. 2D fuel oil.
++ Indicates property value is for dodecane.

Table 2.1 (continued)

Selected Liquid Fuel Properties
(Adapted from Obert (3), and Houser (18))

Fuel	Methanol	Ethanol	Diesel Fuel
Constant Pressure Heating Value at 77°F, $-\Delta H$ (Btu/lbm)			
HHV } liquid fuel	9770	12780	-
LHV } liquid fuel	8644	11604	19197+
Btu/ft^3 (Stoich. Mix.)	89.4	92.9	96.9++
<u>Combustion Properties</u>			
Octane Ratings:			
Research	106	107	-
Pump (RON+MON)/2	99	98	-
Cetane Number	0-5	0-5	47.5++
Flash Point (°F)	52	54-55	158.++
Autoignition Temperature (°F)	867-878	738-964	∿230

+ Indicates property value is from laboratory analysis of No. 2D fuel oil.
++ Indicates property value is for dodecane.

injected into the cylinder and the ensuing spontaneous combustion, there are many different locations in the combustion chamber where the A/F ratio varies from lean to reach (19). This causes many different types of reactions which yield many different kinds of emissions. But note that overall, the A/F ratio is always lean.

2.2.1 Oxides of Nitrogen

A better understanding of the combustion and emission formation processes in Diesel engines was an important goal of this research. Oxides of nitrogen emissions by Diesel engines are very important because of the potential health effects that are associated with them and also because they participate in the photochemical reactions that result in smog formation. McConnell (20) in a brief discussion in his paper writes, "Nitric oxide (NO) has a high affinity for hemoglobin and produces an oxygen deficiency in the blood, but no human deaths have ever resulted from exposure to the gas at concentrations below 25 parts per million (PPM). Nitrogen dioxide (NO_2) produces nitric acid in the lungs causing severe irritation and subsequent lung oedema." Usually any combustion temperature above 1800 K produces sufficient NO to be of concern (21). For heterogeneous combustion, Kesten (22) and Bracco (23) have shown that the amount of NO formed from droplet diffusion flames can be related to the droplet size. A large droplet will produce more NO than that obtained from a group of smaller droplets of the same mass as the larger droplet. McCannell (20) has shown that the temperature which effects the formation of oxides of nitrogen in a Diesel engine is not the bulk gas temperature but a temperature somewhere between the flame temperature and the gas temperature as fixed by

the pressure and volume. Landen (24) showed in a precombustion chamber Diesel engine that the formation of NO_x depends strongly on inlet air temperature (NO_x formation increases as inlet temperature increases). The formation of NO_x as a function of A/F ratio in a Diesel engine increases as A/F ratio increases, reaches a maximum and then decreases. In light-duty automotive engines, NO_x is controlled by retarding injection timing and by using exhaust gas recirculation (EGR). French et al. (25) used EGR in an indirect injection Mercedes engine and showed a significant reduction in the NO_x level (1.66 gm/mile as received, reduced to 0.471 gm/mile). In another test using EGR, they (25) achieved 0.32 gm/mile NO_x by heavily retarding the injection at full load which reduced performance, increased HC and CO emissions and driving of the vehicle was severely affected due to misfire at steady speeds below 40 mph. McConnell (20) showed that a precombustion chamber Diesel engine (indirect injection Diesel engine) produces less oxides of nitrogen than a direct injection Diesel engine.

2.2.2 Soot Formation

The high level of soot formation during the combustion process is a severe problem associated with Diesel engines. The Diesel engine, because of its heterogeneous combustion, produces more soot than a SI engine. Springer et al. (9) reported soot formation in a Diesel engine to be 50 to 100 times more than that of a comparable SI engine. Heterogeneous combustion produces a diffusion flame and across any plane through a diffusion flame there is a wide variation of the fuel-oxidizer ratio from very fuel-rich to very fuel-lean. Thus in a diffusion flame there is always a zone very close to the flame that is

at a high temperature which has a very high carbon to oxygen ratio. This characteristic of a diffusion flame is the reason that they always have some luminosity and form soot relatively easily (21). In all flames, the lower the pressure the lower the tendency to soot. Uyehara (26) in a review of existing data indicates the temperature range of interest for soot formation in the flame is approximately 2000 K to 2400 K. The peak concentration of soot in the flame occurs near 2100 K. At both ends of the range, i.e. 2000 K and 2400 K, the soot concentration is negligible. He also mentions that fuels with high H/C ratios produce less exhaust soot than fuels with low H/C ratios. Usually the required conditions to form soot in a flame are at least two: 1) rich fuel-oxygen mixture and 2) a temperature of at least 2000 K. In addition to these two factors, Frisch et al. (27) showed fuel properties have significant effect on the total particulate emissions and its soluble organic fraction (SOF) (see Section 2.2.3). They showed higher distillation temperature, lower API gravity and higher aromatic content cause higher levels of particulate matter emissions. Formation of soot in a Diesel engine has an effect on measuring NO_x because of the physical and chemical adsorption of NO_x by the soot (28). Therefore, a shorter sample line and also a high flow rate which reduce the retention time for NO_x minimize the error in the measuring of NO_x.

2.2.3 Health Effect of Particulates

Diesel particulates are basically carbonaceous as compared with particulates from SI engines which are largely sulfate, and have higher molecular weight organic substances absorbed onto the chain type agglomerates/aggregates. These higher molecular weight organic substances are soluble when extracted with a solvent and are referred

to as the "soluble organic fraction" (SOF) of Diesel particulate matter (29). EPA has recently shown that the total SOF as well as some of the subfractions of the SOF are mutagenic and possibly carcinogenic (30,31) when introduced into an Ames Microbial Mutagenicity Bioassay Test System (see Appendix A for further information on Ames test). The Ames test is being used to isolate those fractions and individual components of the SOF that are most biologically active (27). That portion of the SOF which has the greatest mutagenic activity is that which contained compounds such as substituted polynuclear aromatics (PNA's), phenols, ethers and ketones (30,31). Mutagens which are direct acting were found; also it was found that metabolic activation increases the mutagenic response to the fractions isolated suggesting the presence of pro-mutagens in the SOF. Pitts et al. (32) and Wang et al. (33) studied this aspect of pro-mutagens and found that the PNA benzo(a)pyrene (B(a)P) needs to be activated metabolically or chemically before it becomes a direct mutagen. Exposure of B(a)P to as little as one PPM NO_2 for 8 hours in the ambient atmosphere serves the function of converting the pro-mutagen B(a)P into the direct mutagen. Diesel particulates also have the potential for chronic lung disease (i.e. emphysema) as a result of long term exposure to these particles (34). In addition, Diesel particulates increase the total mass of suspended particulates in the ambient atmosphere which then contributes to decreased visibility and the promotion of haze and smog (27).

2.3 Alcohol Properties

Familiarity with the physical properties and the combustion of the alcohols helps to better understand the problems encountered upon their utilization as a Diesel engine fuel.

2.3.1 Stoichiometry

Oxygen in ethanol makes up 35 percent of the molecule's mass and supplies almost 14 percent of the total oxygen required for the stoichiometrically correct combustion reaction which occurs at an A/F ratio of almost 9 to 1. Oxygen in methanol makes up 50 percent of the molecule's mass and 25 percent of the total oxygen which is required for the stoichiometrically correct combustion that occurs at an A/F ratio of almost 6.4 to 1.

2.3.2 Heat of Combustion

The heat of combustion of ethanol is less than two thirds that of Diesel fuel oil and the heat of combustion of methanol is less than one-half that of Diesel fuel oil. The heat of combustion that is usually used to compare different liquid fuels is the lower heating value at constant pressure. Table 2.1 gives the heat of combustion for ethanol and methanol.

2.3.3 Latent Heat of Vaporization

Ethanol and methanol have a high latent heat of vaporization due to the presence of the hydroxyl group (-OH) in their molecules. The latent heat of vaporization of 396 Btu/lbm at 68°F and 506 Btu/lbm at 68°F (35) of ethanol and methanol respectively are over three and four times that of dodecane, a pure hydrocarbon representative of a typical

Diesel fuel oil. This high latent heat of vaporization can lead to problems in the induction system. The occurrence of condensation because of the lack of sufficient heat in the induction system and intake manifold can lead to fuel maldistribution among the cylinders. This higher latent heat of vaporization produces internal cooling which has an effect on combustion in SI and Diesel engines.

2.3.4 Vapor Pressure

Ethanol and methanol have relatively low vapor pressure and since they are pure substances have unique boiling temperatures. Higher vapor pressure of ethanol and methanol with respect to gasoline cause difficulties in driveability in cold weather and may induce vapor lock in hot weather. Their vapor pressures can be calculated by the Antoine equation:

$$\log_{10}P = A - \frac{B}{C+T} \qquad (2.1)$$

where:

P = Vapor pressure in mm Hg

T = Temperature in °C

and A, B, C are constants found in table 2.2 (36).

2.3.5 Solubility

Ethanol and methanol are completely soluble in water and likewise water is soluble in ethanol and methanol. Alcohols are considered derivatives of water; where a hydrogen is replaced by a hydrocarbon radical. The strong solvent power of alcohol is due to the existence of the hydroxyl group and only when the C/H ratio reaches 12 or above do alcohols behave like hydrocarbon solvents (37). It is noted that ethanol and methanol have a lower solubility in hydrocarbons of the paraffin series than those of the aromatic series and this

Table 2.2

Constants for the Antoine
Vapor Pressure Equation (36)

	Compound: Methanol (CH_3OH)		
Temperature Range	A	B	C
-14 to 65°C	7.89750	1474.08	229.13
64 to 110°C	7.97328	1515.14	232.85

	Compound: Ethanol (C_2H_5OH)		
Temperature Range	A	B	C
-2 to 100°C	8.32109	1718.10	237.52

solubility decreases with an increase in the molecular weight of the hydrocarbon molecule (14) (38). Aromatics perform poorly as Diesel fuels and therefore they are not present in a large enough amount to cause significant solubility of ethanol and methanol in Diesel fuel oil. As Havemann et al. (14) showed a mere trace of water causes phase separation between Diesel fuel and alcohol. Overall, solubility of anhydrous ethanol in hydrocarbons is higher than methanol with the water tolerance of the ethanol blends being roughly five times greater than similar methanol blends (38) (39).

2.3.6 Material Compatability

Ethanol and methanol are quite different from the other hydrocarbon fuels. They do not lubricate the same as Diesel fuel and have a tendency to corrode some metals. They are corrosive due to the presence of the reactive polar hydroxyl group. If water and salts are present in the fuel, the corrosive tendencies are compounded. Alkali metals react with the alcohols by replacing the hydrogen on the hydroxyl group to form a metal alkoxide and hydrogen gas. Aluminum and magnesium may also form alkoxides but require a catalytic action (37) which can be brought about by impurities in the alcohol.

2.4 Alcohol Combustion

2.4.1 Dissociation

Dissociation of ethanol occurs above 800°C (1472°F); the products of the decomposition are ethylene, acetaldehyde, water and hydrogen (37). The dissociation of methanol produces hydrogen and carbon monoxide at low temperature and low pressure, $\sim$ 100°C at 1.0 atm (12).

2.4.2 Flame Speed and Flammability Limits

The laminar flame speed of ethanol is higher than that of gasoline probably because of the existence of hydrogen and ethylene which both have a high flame velocity, particularly hydrogen which has a flame speed significantly higher than all petroleum fuels. The flame speed of methanol is higher than isooctane particularly for lean mixtures (40). The flammability limits of ethanol falls in a range between that of gasoline and methanol. The relatively wide flammability limits for ethanol and methanol permit alcohol fueled engines to run fairly lean and thereby improve overall efficiency (35).

2.4.3 Flame Temperature

The increase in the number of moles of products during combustion, the internal cooling effect of an inducted charge and the endothermic dissociation of ethanol and methanol combine to produce a flame temperature for ethanol and methanol that is lower than that of gasoline. The flame temperature of methanol because of its higher latent heat of vaporization, is expected to be lower than the ethanol flame temperature. The combustion of methanol in air has been established to have a flame temperature of 3490°F which is 450°F below that of isooctane (12).

2.4.4 Luminosity

Ethanol and methanol have low luminosity because of their structure. Methanol, the lightest alcohol, does not soot at all but the tendency to soot increases with molecular weight of primary alcohols and the luminosity of a flame is related to the amount of carbon formation (41). Then one expects low luminosity for methanol

and ethanol. An engine burning methanol rather than gasoline was shown to have 10% less heat lost to the coolant (12) due to the lower radiation and convection losses brought about by the lower flame temperature and reduced luminosity.

2.4.5 Octane Ratings

The high octane ratings (see Table 2.1) of the lower alcohols compared to gasoline permits the use of a higher compression ratio (CR) in a SI engine. Using neat ethanol in a SI engine and a CR of 7.2 to 1 did not change the power output significantly but when a CR of 11 to 1 was used, there was a 19 percent increase in maximum power output and a 20 percent increase in maximum torque (38).

2.4.6 Cetane Rating

The properties of alcohols (ethanol and methanol) which make them good SI engine fuels (high octane number), results in poor combustion when they are used as compression ignition (CI) engine fuels. A good CI engine fuel is characterized by its ignitability by autoignition wherever a stoichiometrically correct air-fuel mixture is present in the combustion chamber (3). The quicker a fuel ignites in a CI engine the better that fuel is from the standpoint of Diesel knock. The cetane number, which is the knock rating for CI engine fuels, is based upon the autoignition characteristics of the fuel in relation to a blend of two primary reference fuels (3). Diesel fuels used in CI engines have a cetane number of around 50. Ethanol and methanol fuel have a cetane number ranging from zero to five (11). One reason the alcohols have a low cetane number can be attributed to their high latent heats of vaporization which lowers the mixture temperature and increases the

ignition delay. Therefore, using straight alcohols as CI engine fuels is impossible. By heating the fuel before or during induction and then through the use of a very high CR and spark assisted ignition, it may be possible to burn neat alcohol in a Diesel-like engine. Also there are some ways to increase the cetane number of the alcohols and make them useful as CI engine fuels. For example there are some compounds, which are typically nitrates, that are called ignition accelerators or improvers. However, they are expensive and their toxicities in the raw fuel or as a component of the engine exhaust are not known (11).

2.5 Emissions

Because of their low cetane numbers, neat ethanol and methanol have not been tested extensively in Diesel engines. Therefore, emission data for pure alcohol fueled Diesel engines are scant. However, there are a few cases which have been reported whereby adding an ignition improver or by using a high CR and spark plug, neat alcohol has been burned in a Diesel-like engine. Adelman et al. (42) used a CFR swirl precombustion Diesel engine with spark ignition and ran it with straight alcohols (methanol and ethanol). His study showed lower NO_x emissions and no particulate formation with alcohols but higher unburned fuel and CO emissions.

Most neat alcohol studies have been done in SI engines. Here the formation of CO is strongly dependent on the A/F ratio. Leaner A/F ratios give lower CO emission because of more complete burning of the carbon with oxygen to form carbon dioxide (CO_2). It was found that lean operation with neat ethanol in a SI engine gave a slight reduction in CO and slight increase in engine efficiency compared to burning

gasoline (38). The HC emissions are primarily made up of unburned fuel, caused by incomplete combustion due to flame quenching. When using neat alcohol in SI engines, a reduction in HC emissions is possible because of its ability to burn leaner. Also their leaner burning and cooling effect reduces NO_x emissions (38). Another study using neat methanol fueled SI engines showed, in comparison to gasoline, a reduction in CO and NO_x emission but higher unburned hydrocarbons (43).

Studies have been done using alcohol as a supplementary fuel in a Diesel engine and the emissions were reported. Shipinski et al. (44) used a direct injection Diesel engine and injected methanol upstream of the intake valve. They showed reduced smoke output, a slight reduction in NO_x emissions, little effect on CO, and higher unburned hydrocarbons. Pischinger (16) used a direct injection Diesel engine and injected methanol directly into the cylinder. This resulted in virtually eliminating smoke, reducing NO_x emissions, equal or lower CO and HC emissions and reduced aldehyde emissions. Fumigation of methanol and ethanol (mixing the alcohol with the intake air) in a Direct injection Diesel engine produced an increase in HC emissions, a relatively fixed amount of CO except at overall fuel-rich operation where there was an increase, and either an increase or a decrease in the amount of NO_x emissions (45).

The formation of NO is based on the Zeldovich mechanism (21),

$$O + N_2 \rightleftarrows NO + N \qquad k = 1.4 \times 10^{14} \exp(-78500/RT) \qquad \text{(a)}$$

$$N + O_2 \rightleftarrows NO + O \qquad k = 6.4 \times 10^{9} \exp(-6280/RT). \qquad \text{(b)}$$

It forms in the higher temperature region of a flame. Since reaction

(a) requires very high activation energy compared to reaction (b), it controls the system. This makes the formation of NO highly temperature dependent and therefore, the presence of any diluent which reduces the flame temperature will cause a reduction in NO formation. Thus, any reduction of NO caused by the introduction of alcohol could be due to its acting as a diluent.

The formation of unburned hydrocarbons when adding alcohol is a result of two competing factors. Addition of alcohols (ethanol and methanol) that produce hydrogen and ethylene during the combustion reaction, results in a higher flame speed. This reduces the time for heat losses to the cylinder wall which in turn reduces the wall quench and therefore lower HC emissions (46). On the other hand, addition of alcohol can cool the charge and result in temperature reductions which increase flame and wall quench and hence an increase in HC emissions. These two effects compete with each other in alcohol fueled engines and in general it seems that with increasing amounts of alcohol, the temperature reduction effect becomes the predominant factor and HC emissions ultimately increase.

The formation of carbon monoxide (CO) is primarily due to incomplete combustion. It is formed by the combustion of carbon at high temperature in a region of low oxygen concentration.

The air pollution standards for exhaust emissions are based on the toxicity in the environment and their tendency to undergo photochemical reactions to form smog. The burning of alcohol in engines produces unburned hydrocarbons that are much more toxic and reactive than those from gasoline or fuel oil (12). Also soot formation from Diesel engines is probably more harmful as well (47).

2.6 Alcohol Fumigation

Because of their low cetane numbers, ethanol and methanol are usually used as a supplementary fuel in Diesel engines. Fumigation is one way of introducing alcohol into a Diesel engine. Fumigation has been defined simply as the introduction of an alternate fuel into the intake air upstream of the intake manifold by spraying or carbureting. This method of introduction of a supplementary fuel was done by Alperstein et al. (17) and termed fumigation. In this method, a portion of the fuel which is introduced into the engine is mixed with the intake air thereby producing better air utilization. Havemann et al. (14) fumigated both prechamber and direct injection Diesel engines. They found that the amount of alcohol (ethyl alcohol) which could be introduced into the engine was limited by misfiring for the direct injection Diesel engine and by knocking in the precombustion chamber Diesel engine. The fumigation method has been chosen in this study because of the flexibility of the system; i.e. the ability to heat the alcohol fuel or air separately or together, to mix the alternative fuel with air at many different A/F ratios, and to vary the time for vaporization.

CHAPTER III

EXPERIMENTAL EQUIPMENT AND PROCEDURES

3.1 Introduction

In order to study a Diesel engine using fumigated alcohol, an experimental set-up was designed and built. This chapter describes this equipment and the procedures followed to obtain the data for the study.

3.2 Dynamometer and Engine

3.2.1 Dynamometer

The engine was connected by a driveshaft to a Westinghouse type SK cradled dynamometer. This dynamometer has a balance beam resolution of 0.3 lbf and its power equation is:

$$BHP_t = \frac{\text{Scale Units X RPM}}{1000} \tag{3.1}$$

The dynamometer acts as a motor to drive the engine for starting and to measure its friction horsepower. It acts as a generator to load the engine and when operating in the generator mode its output is dissipated as heat from a bank of resistors.

3.2.2 Engine

The engine which was used in this study was a 1978 Oldsmobile 350 CID (5.7 ℓ) V-8 automotive Diesel engine. Table 3.1 lists the engine specifications. The engine was of the indirect injection type utilizing a swirl precombustion chamber cast from stainless steel

Table 3.1

Engine Specifications (7)

Type/Configuration	Prechamber Diesel/V-8
Bore (in.)	4.047
Stroke (in.)	3.385
Connecting Rod (Center to Center) (in.)	5.8855
Displacement (in.^{3})	350
Compression Ratio	22.5:1
Rated BHP @ RPM	120 @ 3600
Rated Torque ft-lbf @ RPM	220 @ 1600
Intake Valve Diameter (in.)	1.875
Exhaust Valve Diameter (in.)	1.625

ground to size and pressed into the cylinder head. This engine was designed to obtain good fuel economy with acceptable emissions and noise levels.

3.2.2.1 Precombustion Chamber Pressure and Rate of Pressure Change

To measure pressure versus time a Kristal type 6031, acceleration-compensated, quartz crystal piezoelectric pressure transducer was used in the precombustion chamber of cylinder number one. The pressure transducer was housed in an adapter which was the identical shape of the glow plug and had the same volume. The output of the quartz transducer, which was proportional to the rate of pressure change with time, was fed to a charge amplifier to provide a voltage proportional to pressure or to a current amplifier to provide a time rate of pressure change signal. The charge amplifier was set such that the proportionality constant was 100 psi per 1.0 V output.

3.2.2.2 Knock Quantification System

In an effort to determine the maximum allowable substitution of alcohol for Diesel fuel before reaching knock limited operation, Houser et al. (18) developed a knock quantification system. A study by Barton et al. (48) determined through a frequency analysis that a rate of pressure (dP/dt) signal was a responsive indicator of knocking combustion in SI engines. Similarly, a dP/dt signal was used in this study for quantification of Diesel knock.

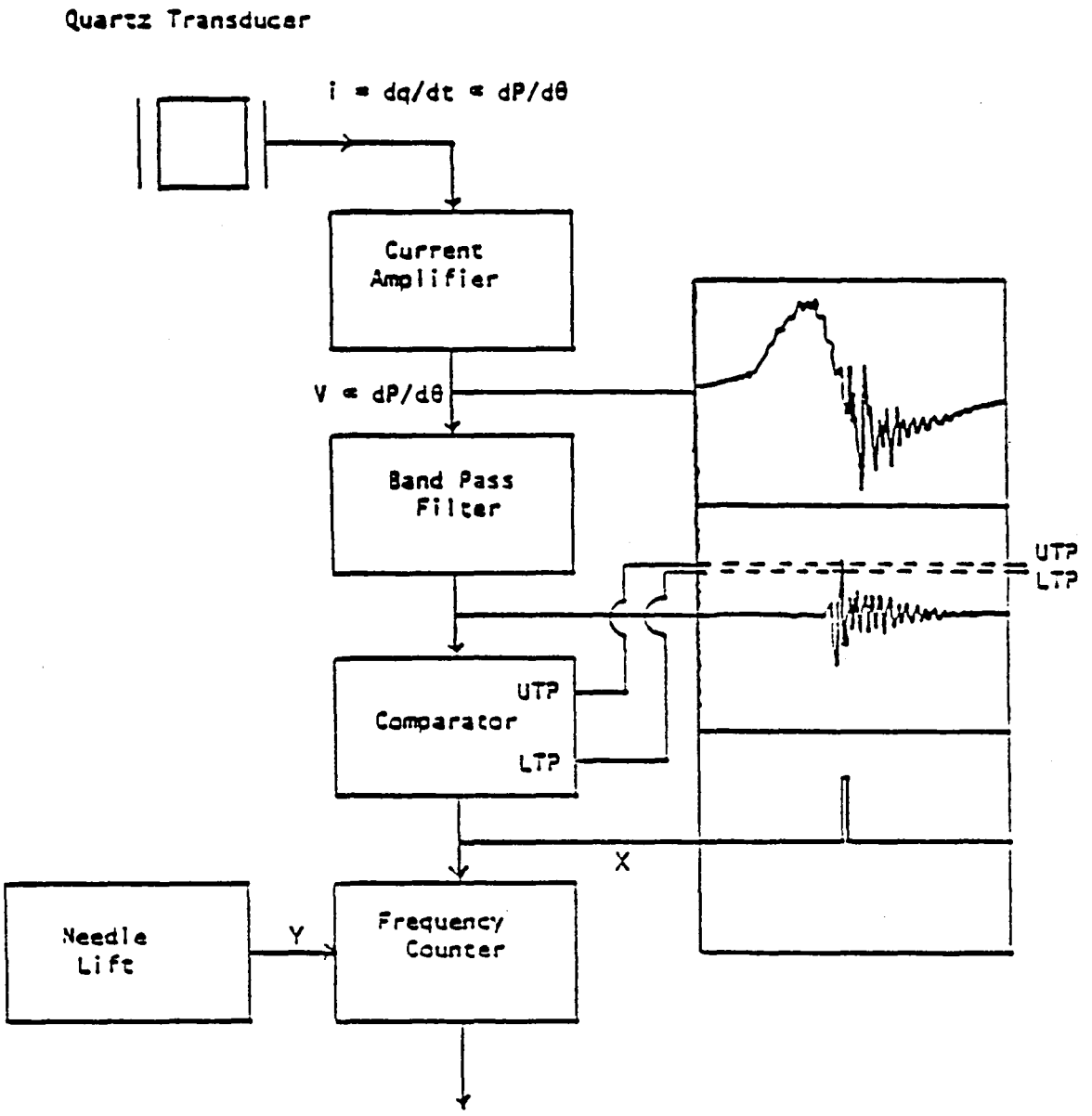
The output signal of a quartz pressure transducer was input to a current amplifier which provided a voltage signal proportional to dP/dt. This voltage signal was passed through an electronic bandpass filter (Khron-Hite Model 330M) which attenuated signals outside a

1000-10,000 Hz bandwidth. Rates of pressure change associated with the compression and expansion strokes of the engine as well as high frequency noise were thus effectively eliminated.

The magnitude of the filtered pressure signal was used as the basis for comparing combustion knock under different engine operating conditions. A non-inverting comparator with hysterisis was used to measure the magnitude of this signal. Two reference points, an upper trip point (UTP) and lower trip point (LTP) were incorporated in the comparator design. When the magnitude of the filtered pressure signal exceed the UTP, the comparator output shifted to a high state condition. The comparator output remained at this high state until the filtered dP/dt signal dropped below the LTP. The comparator thus provided a square wave pulse for each filtered dP/dt spike which exceeded a preselected level.

The wave pulses from the comparator were input to a digital frequency counter (Beckman Counting Unit 6380, Input Module 683, and Function Unit 678); a needle lift signal was also input to the frequency counter to form a time base to determine the ratio of comparator output pulses (corresponding to severe knock events) to combustion cycles. A schematic diagram of the knock quantification system and characteristic output signals are presented in Fig. 3.1.

The knock quantification signal was intended to provide information regarding the relative increase in knock severity as alcohol was substituted for Diesel fuel. Obert (3) has indicated that the maximum desirable rate of pressure rise in CI engines is limited to 50 psi/°CA. In accordance with this guideline, severe knock events were measured as points at which the filtered dP/dt signal exceeded this level.



$$(K_s)_f = \frac{X}{Y} = \frac{K_s}{\text{Combustion Stroke}}$$

Fig. 3.1 Knock Quantification System

Although the knock quantification system could not provide specific details concerning the mechanisms of combustion knock or the absolute levels of dP/dt in the combustion chamber, it did permit comparison of knock intensity for baseline and alcohol fumigated operating conditions.

3.2.2.3 Fuel Injection System and Needle Lift Instrumentation

The engine fuel injection system used a Roosa-Master Model DB2 injection pump of the rotary opposed plunger, distributor type produced by the Hartford Division of Stanadyne Incorporated. A hydraulic piston in the injection pump provided precise automatic control of injection timing according to the load and speed of the engine. This injection pump provided a nominal pressure of 1800 psig at the injection nozzles which were also made by Stanadyne. The injection nozzles were of the fixed-orifice type with two 0.017 in. diameter holes which gave a 30 deg. spray angle.

For experimental purposes, the number one cylinder injection nozzle was modified to provide needle lift by using a Kaman Model KD 2400 proximity indicator, a power supply, and signal conditioner.

3.2.2.4 Analogue Data Manipulation Capabilities

To generate a timing mark which would serve as a reference point for the pressure and needle-lift traces a magnet pick-up was mounted to sense a tab on the dynamometer shaft. To display these signals (timing mark, needle-lift, pressure and time rate of pressure change) a Nicolet series 2090 Model 206 Explorer digital Oscilloscope incorporating a Model 111 magnetic disk memory was used. Output terminals from the oscilloscope allowed interfacing with a Hewlett-Packard Model 7044A

X-Y plotter. This oscilloscope was also interfaced with an Apple II microprocessor so that information stored on its disk memory could be massaged and used in various calculational programs.

3.3 Fuel Systems

3.3.1 Diesel Fuel System

The pump used to inject diesel fuel into the engine was described in Sec. 3.2.2.3. On this pump, fuel flow rate could be changed by varying its rotary fuel metering valve. In the production automobile this valve is connected directly to the accelerator pedal. The amount of fuel which was introduced into the engine was measured with calibrated rotameters to cover the range of interest. The fuel return line from the injection nozzles rejoined the fuel circuit downstream of the rotameters. Fluctuations in the rotameter readings at low flow rates caused by the electric fuel-transfer pump were eliminated by pressurizing the fuel tank with nitrogen. The needle-lift instrumentation which was used on the number one cylinder allowed injection timing, injection duration and fuel delivery rate to be determined. Figure 3.2 shows Diesel fuel system.

3.3.2 Alcohol Fuel System

The alcohol fuel system consisted of an air-atomizing nozzle (spraying system series 1/4J) through which alcohol was sprayed into a steel cylindrical variable-length mixing channel. Eight separate steel tubes joined the mixing channel to the engine so that the same amount of charge was distributed to each of the eight cylinders (see Fig. 3.3). To mix alcohol completely with the air required good alcohol atomization and the correct spray pattern. Information giving the proper air

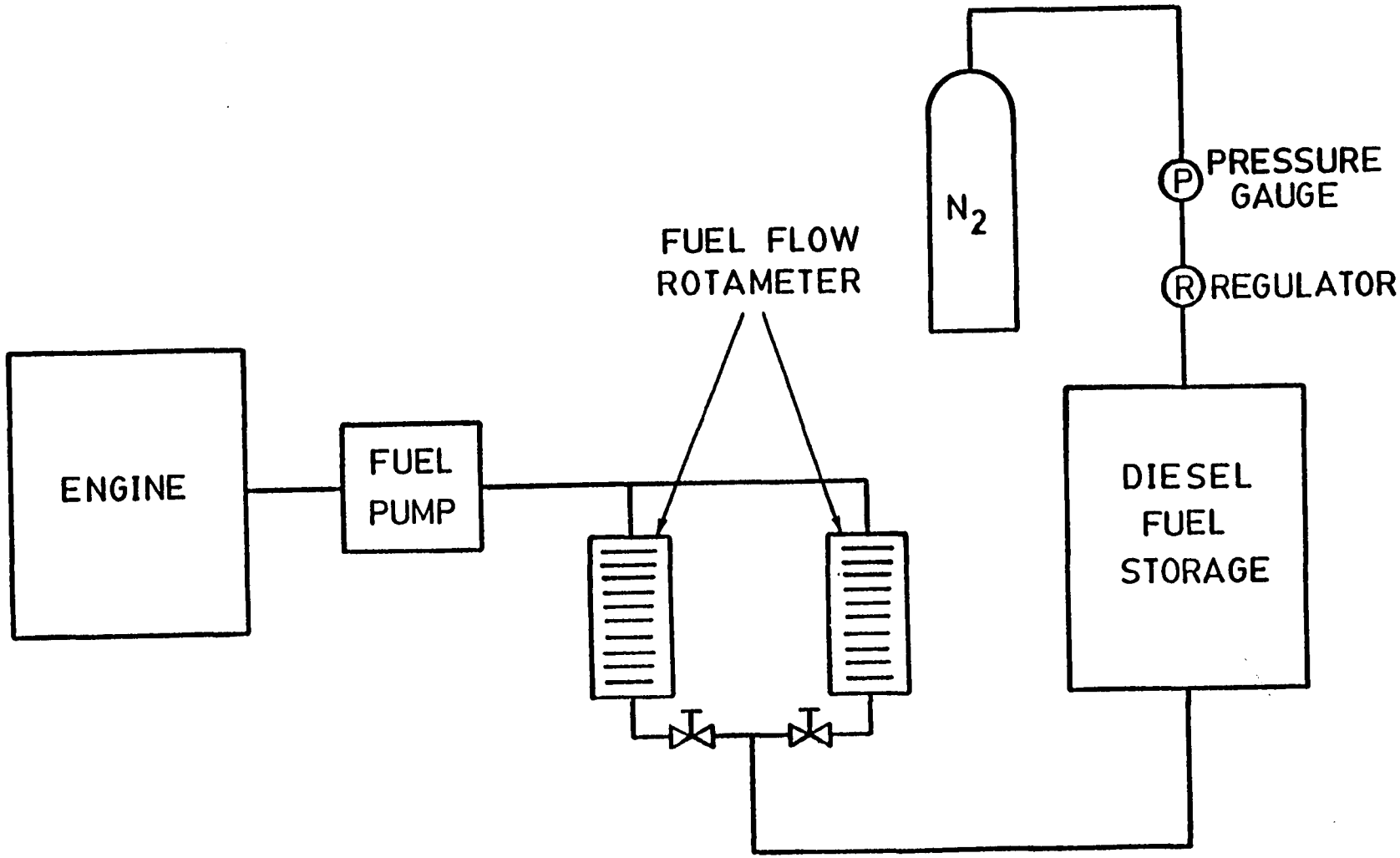


Fig. 3.2 Diesel Fuel Supply System

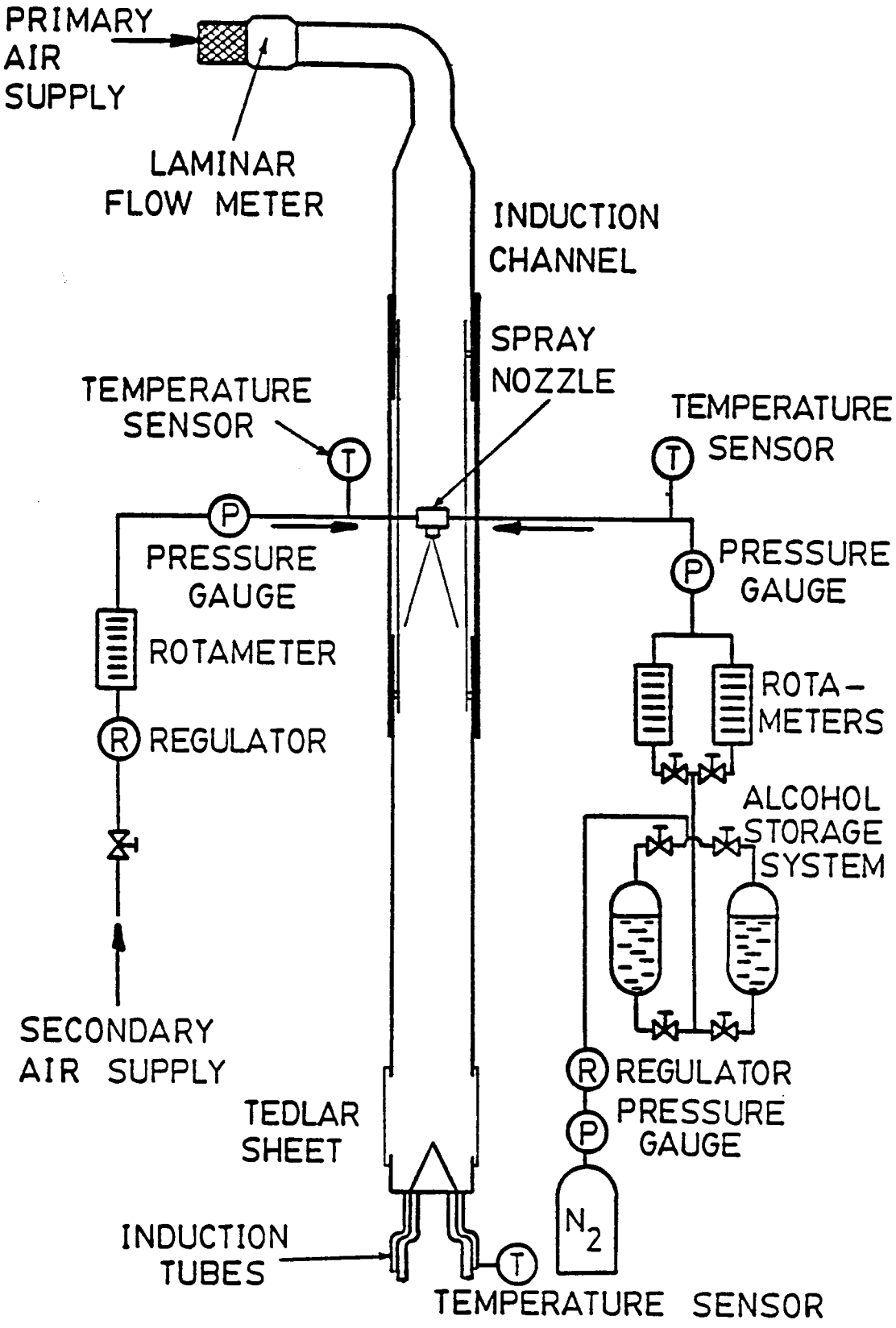


Fig. 3.3 Alcohol Fuel System

pressure to insure good atomization for various alcohol flows was provided by the nozzle manufacturer (Spraying System Co.). These recommendations were followed throughout the testing program.

The source of secondary air used by the atomizing nozzle was 60 psig shop-air which was controlled by a shut-off valve, a pressure regulator, a gauge, a calibrated rotameter, and a thermocouple. Two five-gallon, high-pressure stainless steel cylindrical tanks (Fig. 3.4) contained the alcohol supply. These tanks were kept under a nitrogen pressure in order to: 1) reduce the possibility of contacting air and absorbing water from it and 2) provide the required pressure for the different alcohol flow rates which were measured by two calibrated rotameters to cover the range of interest, a pressure gauge, and a thermocouple.

3.4 Air Inlet System

The primary air supply for the engine was taken from the well-ventilated room and its temperature, relative humidity and barometric pressure were measured. The primary air inlet was located upstream of the mixing channel for the alcohol fuel introduction system. The primary air flow rate was measured with a Meriam 50 MC2-4F laminar flow element. The pressure difference across this element was indicated by either a ten-inch vertical or a three-inch inclined Ellison Manometer.

3.5 Smoke Opacity

To measure the smoke opacity an end-of-the-line United States Public Health Service Full-Flow Light Extinction Smokemeter was used. The instrument was mounted over the open end of the 3.5 in. I.D. exhaust pipe such that the exhaust passed through the 10 in. O.D. ring.

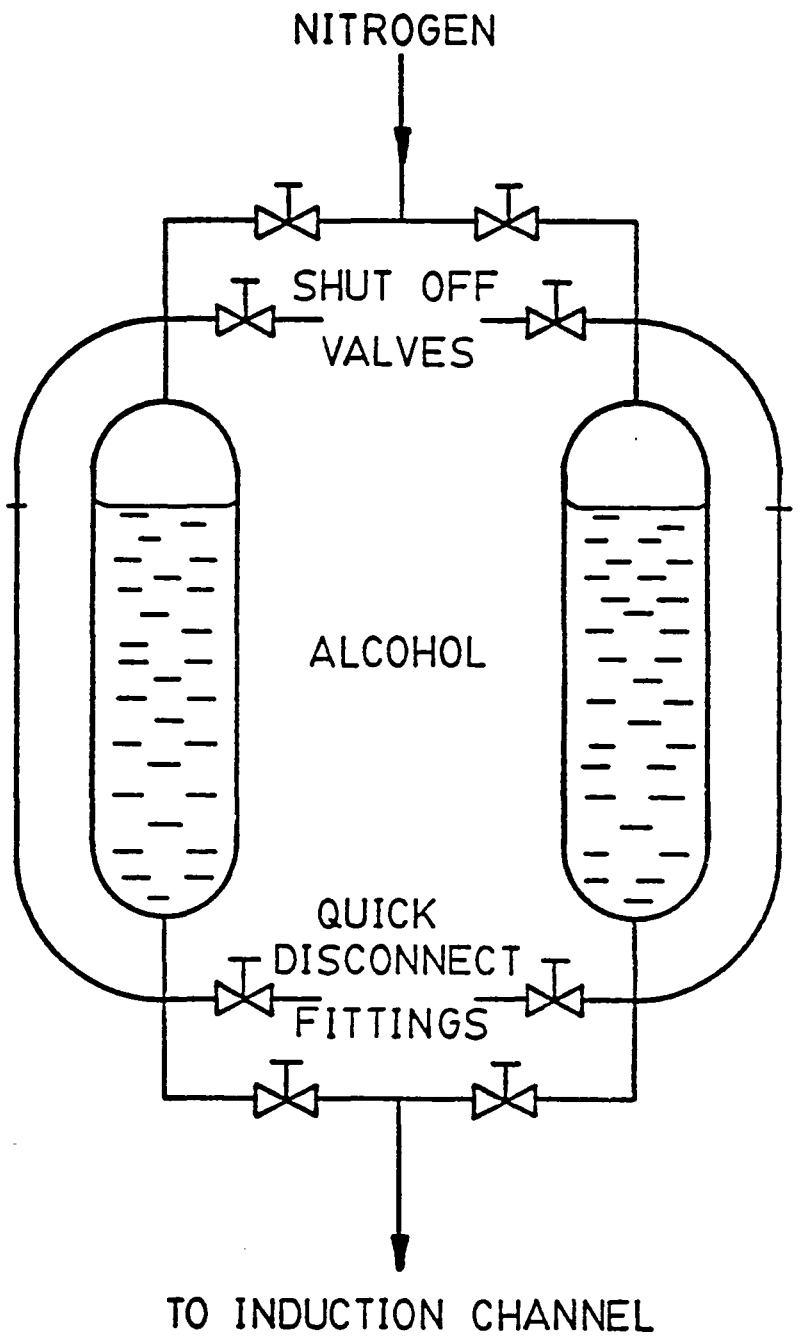


Fig. 3.4 Alcohol Storage System

This smokemeter consisted of a photovoltaic cell, lenses and a light source. To keep the photovoltaic cell and lenses clean, they were swept with a small stream of compressed air. Because of the sensitivity of the smokemeter to heat, calibration before each reading was done and this gave results which were repeatable. Calibration of the smokemeter was simply done with an appropriate shaped section of 4 in. O.D. pipe which slipped over the exhaust pipe and caused the exhaust to flow outside of the 10 in. O.D. ring, this then gave a clear optical path for calibration. Figure 3.5 shows a diagram of this smoke opacity measurement system.

3.6 Diesel Particulate Collection

Since Diesel engines produce more particulate than spark ignition (SI) engines, and since there may be a health hazard associated with the particulate, research in this area is a part of this work. The Environmental Protection Agency (EPA) has defined Diesel exhaust particulate (excluding water) to be material that can be collected at a temperature not to exceed 125°F on a glass-fiber filter (49). The experimental set-up to collect particulate and the procedures followed are now described.

The sample was drawn using a 0.372 in. I.D. stainless steel probe which was placed at the center of the 2.5 in. I.D. main exhaust pipe at a point more than ten diameters downstream from the exhaust mixing tank. Simple calculations showed that turbulent flow existed in the main exhaust pipe. Therefore, the relatively small sample probe would not create any large disturbance in the exhaust flow. After being drawn through the probe, the exhaust sample went through a heat

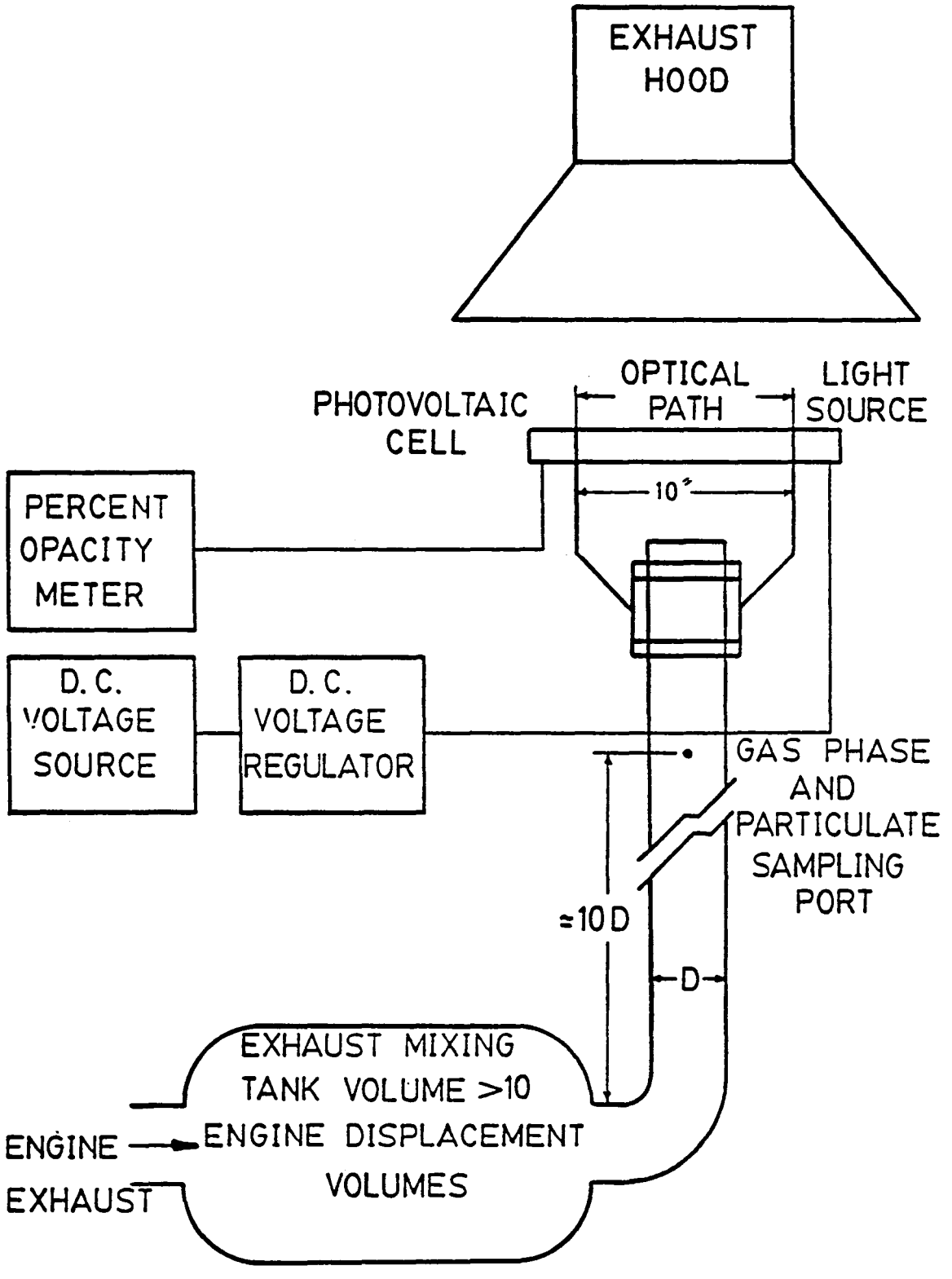


Fig. 3.5 Smoke Opacity Measurement System

exchanger before being trapped on a 142 mm "Pallflex" filter. Isokinetic sampling was done to prevent any mass discrimination of the exhaust particles by the probe. A simple scheme to do this was developed wherein the flow rate of the sample was adjusted so that no difference between the static pressure in the sample line and in the exhaust pipe would exist. The entire sampling system was made of stainless steel. Throughout this study, 142 mm "Pallflex" (Pallflex Products Corporation, Putnam, Connecticut) teflon-coated, glass-fiber filters were used. Figure 3.6 shows the particulate sampling system.

3.7 Gaseous Exhaust Emission Analysis

The exhaust gas emission analysis system is shown in Figure 3.7. The entire system was movable as a unit and during a test run, the system was connected to the sample probe in the exhaust pipe of the engine. The system consisted of: A Beckman Model 109 flame ionization total hydrocarbon detector (FID) which was spanned on 530 PPM methane in nitrogen. A Beckman IR15 non-dispersive infrared (NDIR) carbon dioxide (CO_2) analyzer which was spanned on 14.9 percent CO_2 in nitrogen. A Beckman model 864 infrared carbon monoxide (CO) analyzer which was spanned on 4003 and on 418 PPM CO in nitrogen. A Beckman model 741 oxygen analyzer with a quick response amperometric sensor which was spanned on air for 21 percent oxygen. A Beckman model 955 heated oxides of nitrogen analyzer which was spanned on 550 PPM NO (less than 5 PPM NO_2) in nitrogen. Before passing through the instruments, the exhaust gas sample first passed through a particle

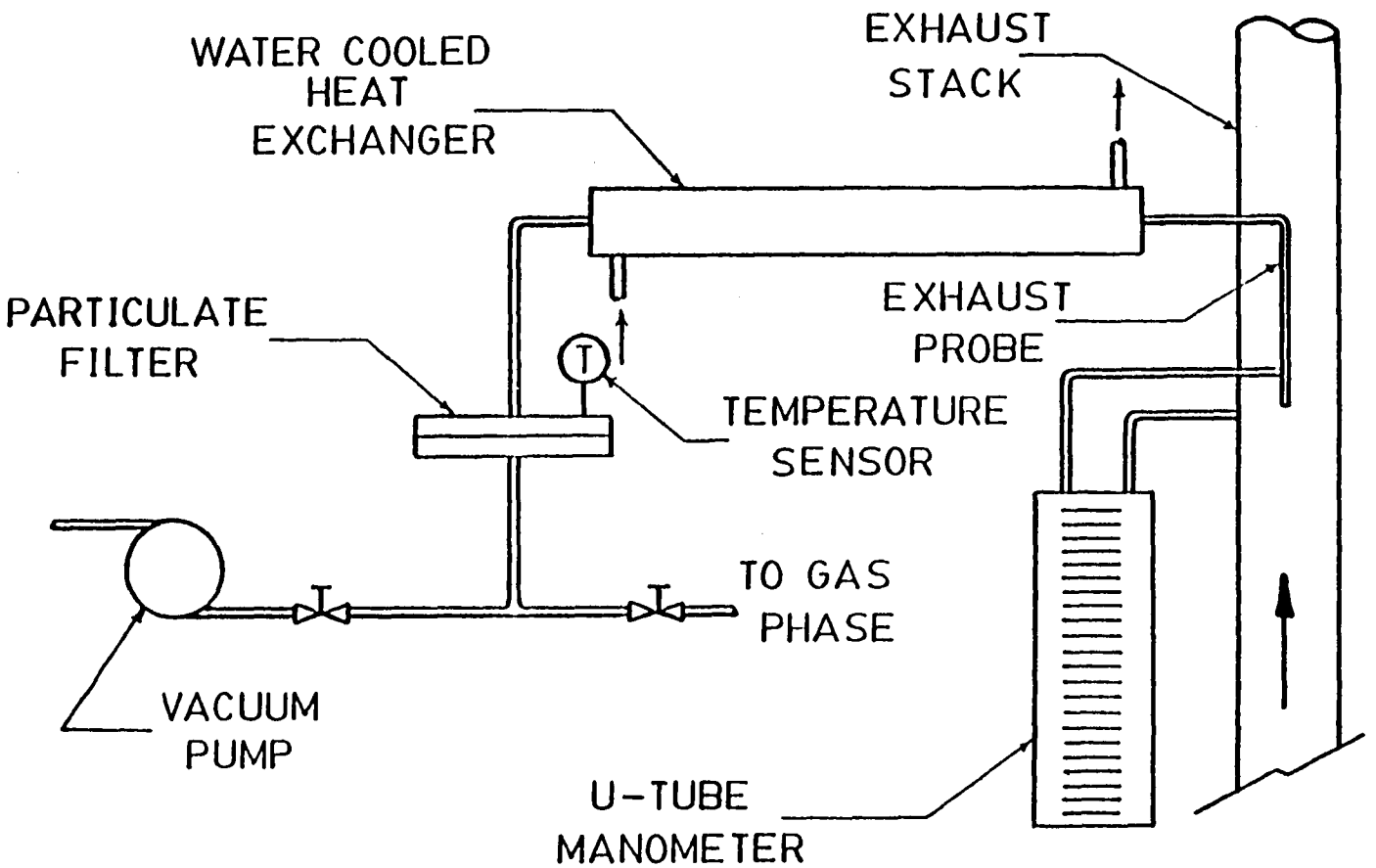


Fig. 3.6 Diesel Exhaust Particulate Matter and Exhaust Gas Sampling System

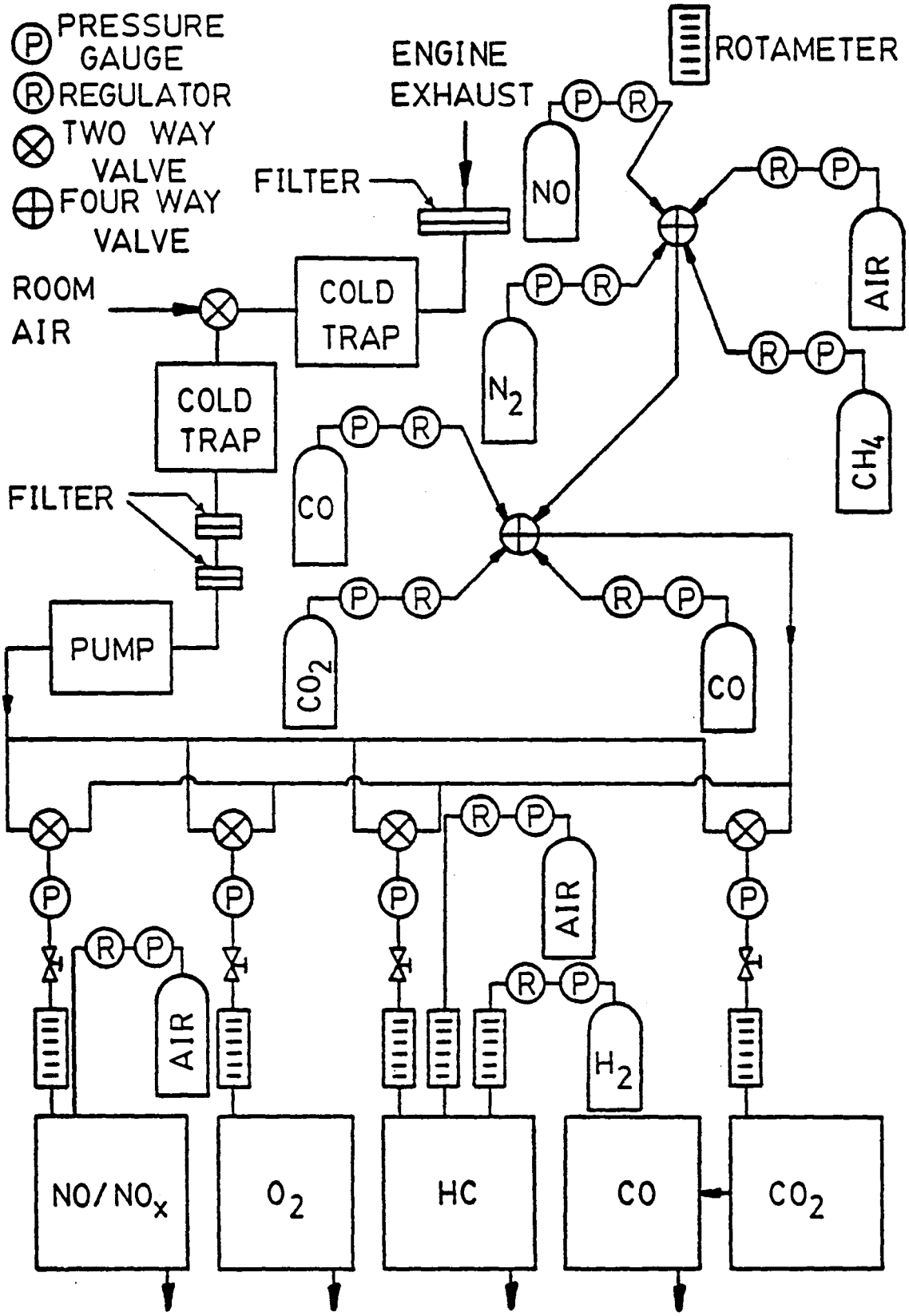


Fig. 3.7 Exhaust Gas Sampling Instrument Train

filter and a cold trap to remove any particulate and water from it. A feature of this system was that each instrument could be calibrated while taking data with any or all of the other instruments.

3.8 Temperature Measurements

Chromel-alumel thermocouples were used to measure all temperatures. The output of each thermocouple was indicated in degrees Fahrenheit by a Leeds and Northrup Manual potentiometer calibrated for chromel-alumel thermocouples with an ice bath for the reference point. Also a Leeds and Northrup Speedomax model H recorder was used to record the temperature from twelve of the thermocouples.

3.9 Experimental Procedure

Since it was decided to determine the effect of substituting alcohol for fuel oil while keeping the total fuel energy constant, it was first necessary to determine the fuel energy supplied for any condition in the test matrix. This was done by running the engine at each test condition on only the baseline fuel. Also some engine operating parameters were fixed to eliminate them as a cause for any observed variations. The cooling water was maintained at 180$\pm$5°F and the oil temperature at 230$\pm$5°F. There was no control on the inlet air temperature, pressure, or relative humidity but they were recorded to allow engine performance to be corrected to standard conditions. The fuel injection timing used followed the program that was built into the pump by the manufacturer. No attempt was made to optimize the injection timing for alcohol.

Engine operating conditions throughout the test were fixed by RPM and rack setting (2000, 1720, 1500 RPM - Full, 3/4, 1/2, 1/4 Rack). For each rack setting the amount of fuel flow required by the engine was determined by the total fuel energy required.

The maximum rated output of the engine was 120 brake horsepower at 3600 RPM. With the engine running under steady state conditions at 3600 RPM and 120 corrected horsepower, the rotary fuel metering valve on the injection pump was locked in place. Then the dynamometer load was increased until the engine speed dropped to 2000 RPM. The fuel flow rate at this condition was defined as full rack at 2000 RPM. Further dynamometer load increases permitted the full rack 1720 and 1500 RPM conditions to be defined. The nominal fractional rack settings were then defined by simply multiplying the full rack fuel flow rate at any speed by the appropriate fraction.

The main purpose of this study was to investigate the effect of alcohol addition on performance and exhaust emissions of a light-duty Diesel engine. Starting at the baseline condition shown in each cell of the test matrix (see Sec. 4.2), alcohol was substituted in increasing amounts for the baseline fuel until the engine either started to miss or its operation became knock limited. All tests with alcohol were done without heating the inlet air and the sliding section of the induction system was set as high as possible to provide maximum time for vaporization. The data obtained from each test were loaded into an Apple II microprocessor for reduction to get the parameters of interest.

CHAPTER IV

PRESENTATION OF EXPERIMENTAL RESULTS

4.1 Introduction

To study the effect of alcohol fumigation on the performance and emissions of a V-8, 5.7ℓ Oldsmobile Diesel engine, the equipment discussed in Chapter III was used. All the raw data collected during a test were reduced using an Apple II microprocessor and are presented in Appendix B. The purpose of this chapter is to present and describe the results of the present study. Therefore no 2000 RPM methanol results are given because they were presented in a former study by Houser (18).

4.2 Baseline Testing

In order to have a basis of comparison for the alcohol tests, it was first necessary to establish a baseline using No. 2D fuel oil. As described in Chapter III, all full-rack fuel flows for the different engine speeds (2000, 1720, 1500 RPM) were determined and from these, fuel flows for the different nominal rack settings (1/4, 1/2 and 3/4 rack) at each speed were calculated. The engine was run at each rack setting and speed to establish the baseline test matrix. Energy input rate for each cell of the test matrix was based on the lower heating value of No. 2D fuel oil. The baseline test matrix data appear in Table 4.1. The repeatability of these data were within ± percent. Inspection of these data show that the full rack energy is not partitioned exactly at all the fractional rack settings. This is the reason the fractional rack settings are referred to as being

Table 4.1 - Baseline Data

Rack \ RPM	1500	1720	2000
1/4	12.8* 19.3 0.687 13212. 2813.	14.5 19.1 0.691 13259. 3208.	12.5 14.1 0.818 15710. 3265.
1/2	39.20 59.1 0.427 8205. 5360.	39.65 52.1 0.453 8702. 5751.	40.4 45.7 0.488 9366. 6302.
3/4	51.9 78.34 0.447 8588. 7433.	62.1 81.6 0.458 8797. 9098.	65.7 74.3 0.454 8704. 9535.
Full	57.2 86.3 0.492 9439. 8996.	68.2 89.7 0.475 9113. 10360.	77.5 87.7 0.457 8783. 11348.

* Data in each block is tabulated as follows:

bhp
bmep in PSI
bsfc in lbm fuel/bhp-hr
bsec (brake specific energy consumption) in btu/bhp-hr
Total fuel energy input in btu/min
corrected to standard Atmospheric Conditions;

T=540°R, P = 29.38 in. Hg

nominal settings. The properties of the Diesel fuel oil, the engine lubricating oil, methanol and ethanol used in this study are presented in Tables 4.2 and 4.3.

4.3 The Effect of Alcohol Fumigation on the Occurrence of Knock

Since alcohols (methanol and ethanol) have low cetane numbers ($0 < CN < 5$) and therefore are not good Diesel engine fuel, the occurrence of severe knock is probable when they are used as a fumigant. The technique used for quantifying knock was developed by Houser et al. (18) and was described very briefly in Chapter III. The knock quantifying system was set to count those rate-of-pressure change peaks during a combustion event that exceeded 50 psi/°CA. An average count greater than 1.5 for 1000 combustion events was arbitrarily designated as severe knock. As seen in Fig. 4.1, the last point of the 3/4 and full rack tests at 1720 RPM for methanol fumigation and also as Fig. 4.2 shows the last point for all 3/4 and full rack tests was knock limited, i.e. severe knock occurred. For methanol fumigation, no severe knock occurred for 1500 RPM at all rack settings and for 1720 RPM at the 1/4 and 1/2 rack settings. Ethanol fumigation did not show severe knock at the 1/4 and 1/2 rack settings for all speeds.

4.4 The Effect of Alcohol Fumigation on Engine Efficiency

Since the total fuel energy was fixed as the amount of alcohol fumigated was increased, the thermal efficiency curves also represent power curves, see Figs. 4.3 and 4.4. Methanol substitution at the higher rack settings (3/4 and full rack) resulted in a slight thermal efficiency increases. However, operation at 1720 RPM became knock

Table 4.2

Baseline Fuel and Lubricating Oil Specifications

PROPERTIES OF BASELINE TEST FUEL

Fuel Type	MIL F 46162 A Grade 2
Properties	
Gravity, °API	35.9
Cetane, Calculated	47.5
Viscosity, Kinematic @ 100°F	2.47
Flash Point, °F	158.
Pour Point, °F	-10.
Cloud Point, °F	0
Net Heat of Combustion, Btu/lbm	19197.
Arometics, %	36.5
Distillation	
IBP, °F	376
50% point, °F	490
EP, °F	627
Recovery, %	99.0

PROPERTIES OF TEST ENGINE LUBRICATING OIL

Oil Type	Shell Rotella T Premium Multipurpose HD
Saybolt Viscosity @ 100°F, SSU	560.0
" " @ 210°F, SSU	67.0
Viscosity Index	98.0
Pour Point, °F	5.0
Sulfate Residue, % wt.	1.0
Neut. No. (TBNE)	7.0
Quality Level	
Meets	MIL-L-2104C
Exceeds	MIL-L-46152 MIL-L-2104B
API Classification	CD,SE

Table 4.3 - Selected Alcohol Properties

	Methanol	Ethanol
Molecular Weight	32.04	46.06
Research Octane No.	106	107
Cetane No.	0-5	0-5
Flash Point, °F	52	60
Autoignition Temp., °F	867-878	738-964
Flammability Limits (% by volume in air)	6.7-36	4.3-19
Higher heating value, Btu/lbm	9770	12780
Lower heating value, Btu/lbm	8644	11604
Latent heat, Btu/lbm	502	396
Specific gravity	0.792	0.794
Boiling Temp. @ 1 atm, °F	149	172
Vapor Pressure @ 100°F, Psia	4.55	2.25

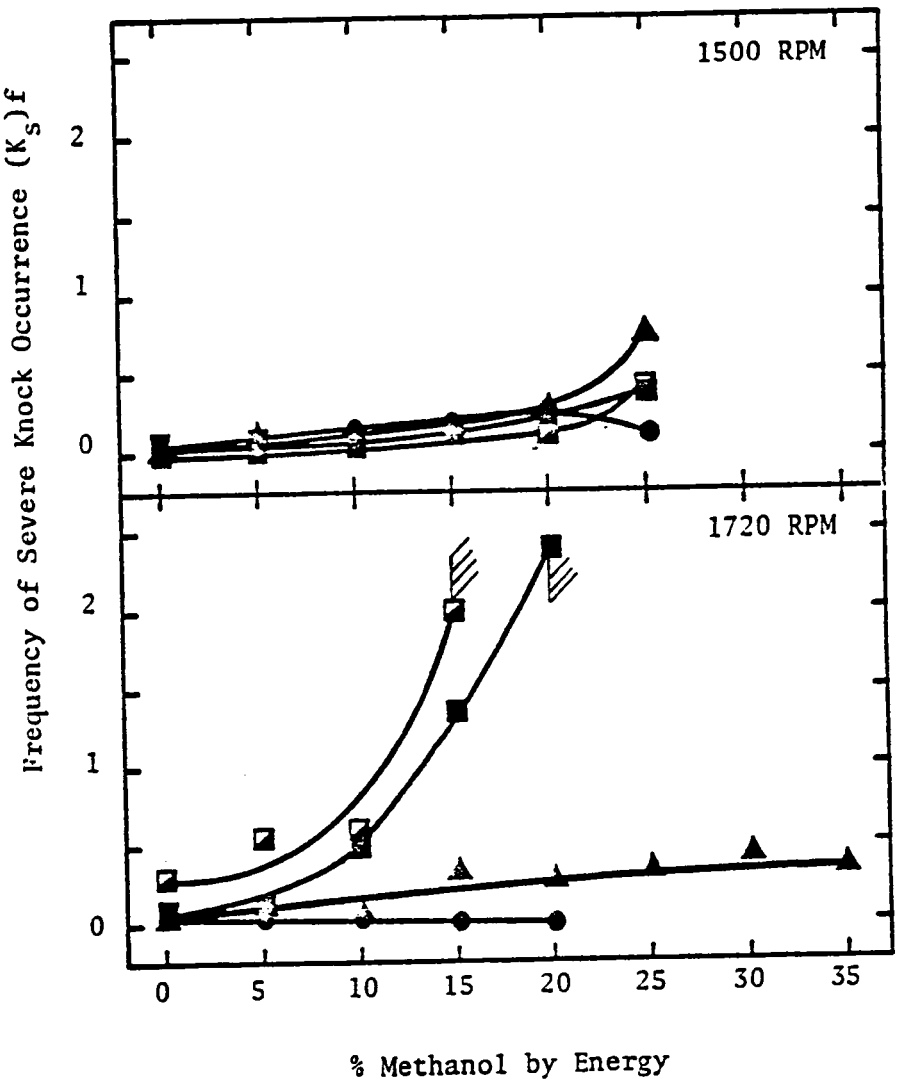


Fig. 4.1 Frequency of Severe Knock Occurrence as a Function of Fumigated Methanol

● - 1/4 Rack

▲ - 1/2 Rack

■ - 3/4 Rack

◪ - Full Rack

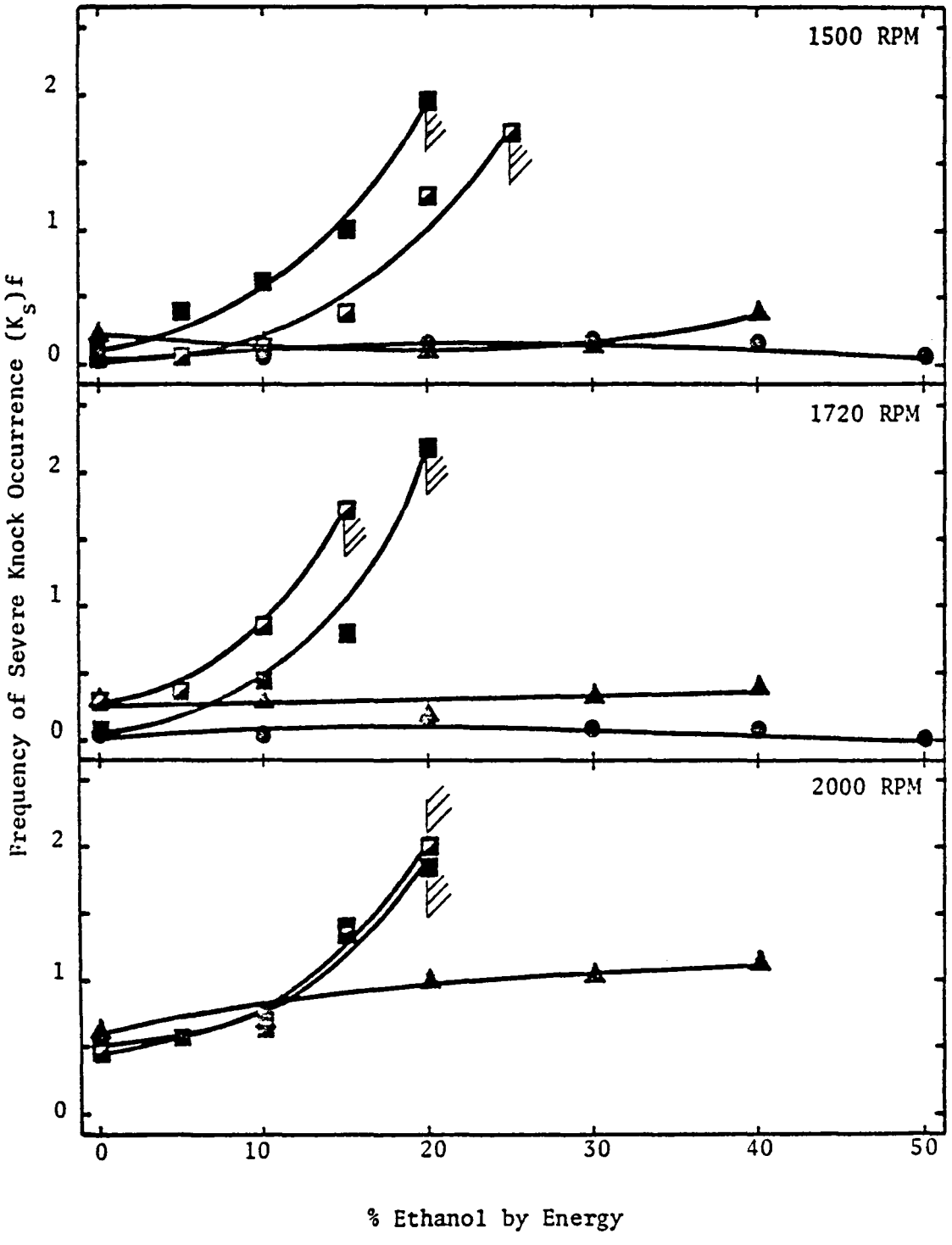


Fig. 4.2 Frequency of Severe Knock Occurrence as a Function of Fumigated ethanol

● - 1/4 Rack

▲ - 1/2 Rack

■ - 3/4 Rack

◪ - Full Rack

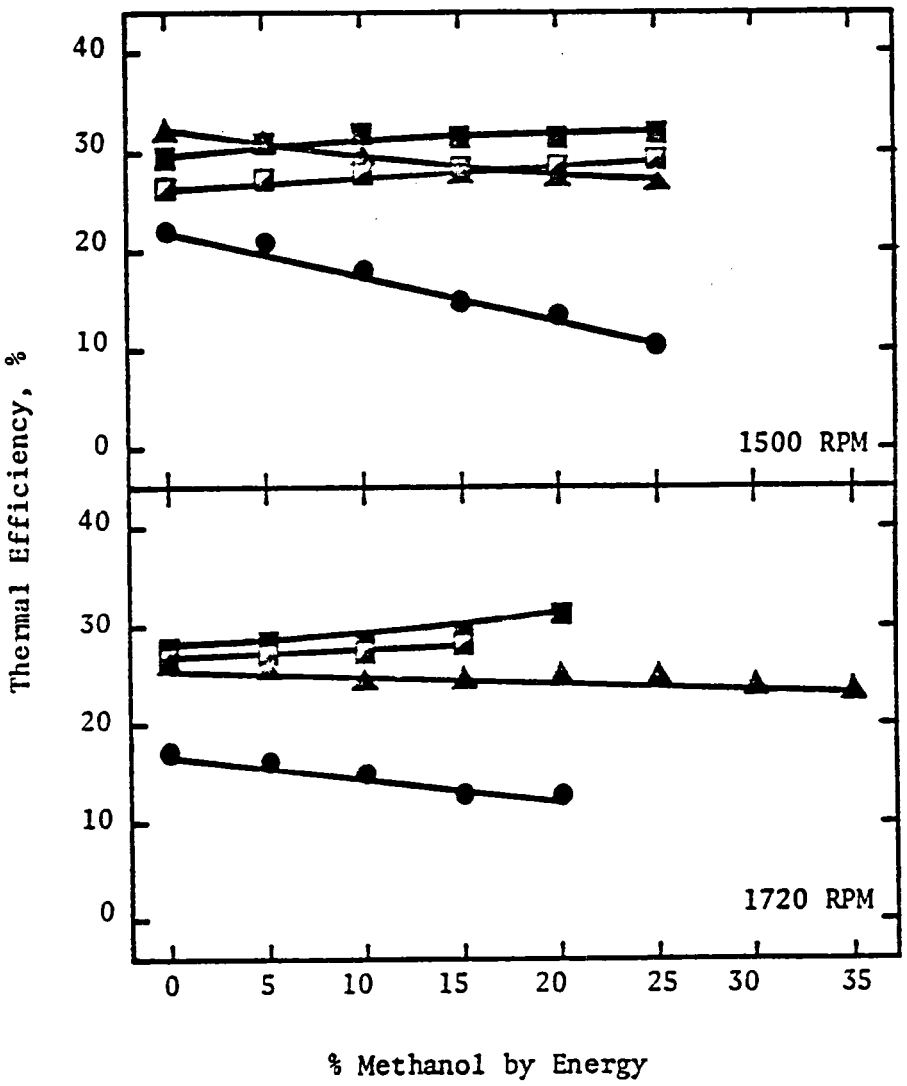


Fig. 4.3 Thermal Efficiency as a Function of Fumigated Methanol

●- 1/4 Rack

▲- 1/2 Rack

■- 3/4 Rack

◪- Full Rack

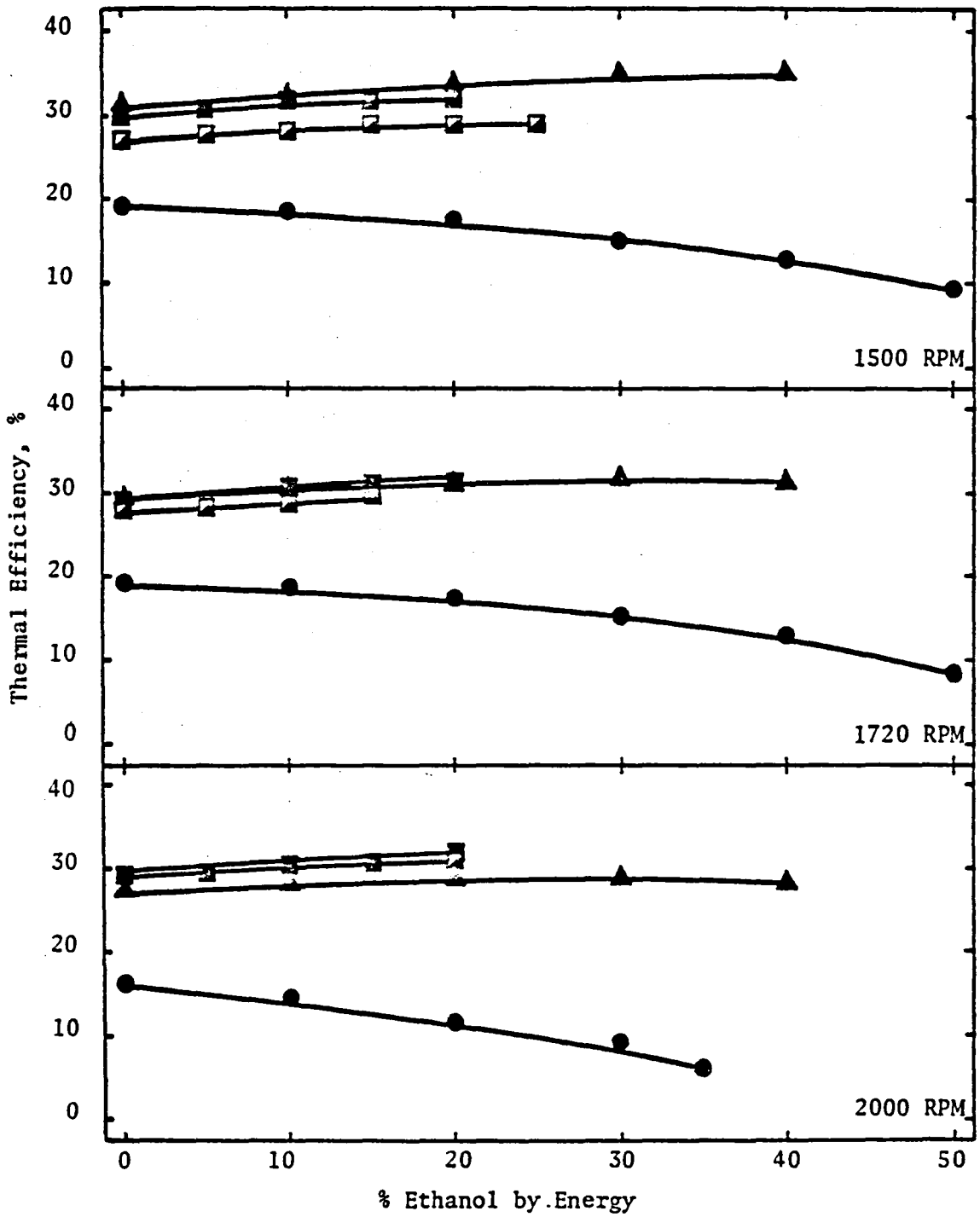


Fig. 4.4 Thermal Efficiency as a Function of Fumigated Ethanol

● - 1/4 Rack

▲ - 1/2 Rack

■ - 3/4 Rack

◪ - Full Rack

limited. At the 1/4 and 1/2 rack settings, thermal efficiency decreased for 1500 and 1720 RPM. Ethanol substitution showed the same trend at higher rack settings as methanol, a slight efficiency increase at 3/4 and full rack but became knock limited here for all speeds. The 1/2 rack results also showed a thermal efficiency increase with the substitution of ethanol but here operation was limited by engine roughness as the amount of ethanol fumigated approached 40%. Thermal efficiency dropped at 1/4 rack for all speeds and the substitution of ethanol was limited because of the high power loss (about 55% of the baseline value) that eventually would lead to misfire.

4.5 The Effect of Alcohol Fumigation on Air-Fuel and Equivalence Ratios

Figures 4.5 through 4.8 show the effect of methanol and ethanol fumigation on the measured A/F ratio and the equivalence ratio (Φ). Measured A/F ratios were determined from the measured mass of air and fuel at each test condition. Figures 4.5 and 4.6 indicate that the measured A/F ratio decreases as the amount of alcohol (methanol and ethanol substituted for the Diesel fuel oil increased). The equivalence ratio was determined from the measured A/F ratio and the stoichiometric A/F ratio based upon the total fuel supplied to the engine at any condition. Figures 4.7 and 4.8 generally show that the equivalence ratio remains almost constant for each speed and rack setting.

4.6 The Effect of Alcohol Fumigation on Engine Wear

Checks for cylinder wear were made at two different times during this study; 1) after methanol fumigation for 250 hours of engine

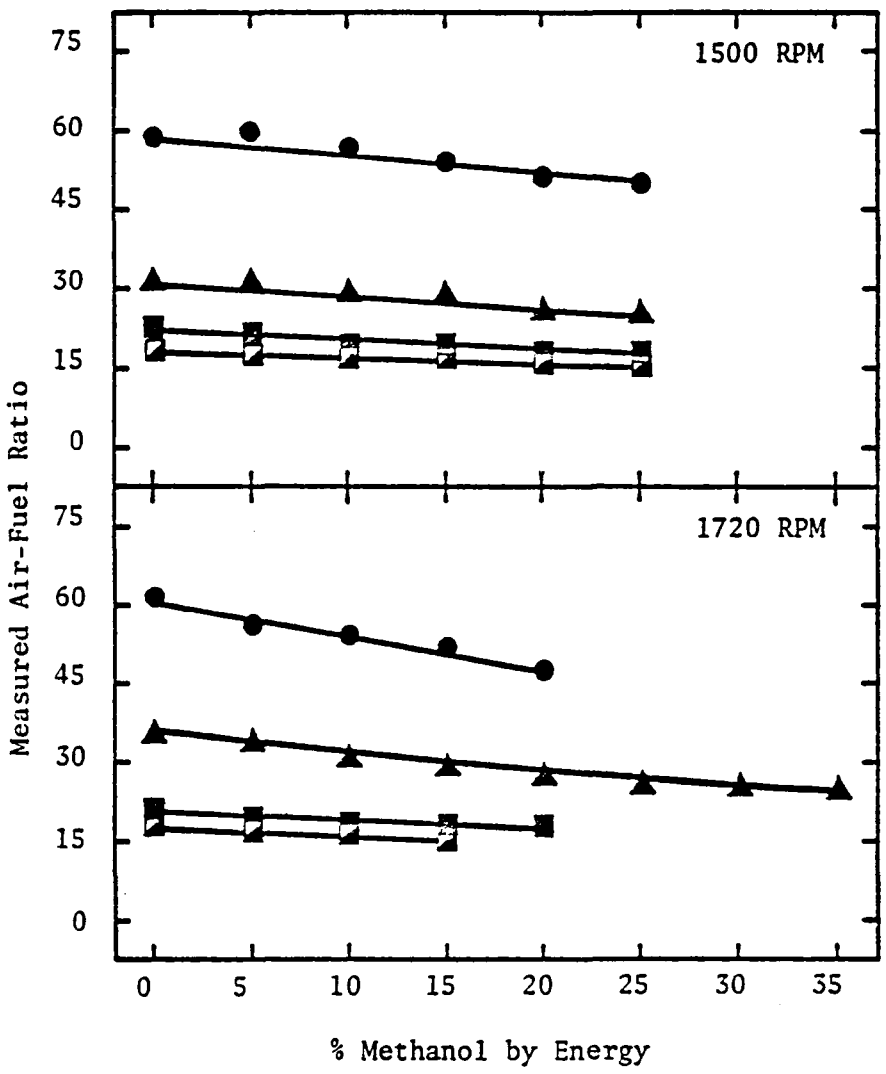


Fig. 4.5 Measured Air-Fuel Ratio as a Function of Fumigated Methanol

● - 1/4 Rack

▲ - 1/2 Rack

■ - 3/4 Rack

◪ - Full Rack

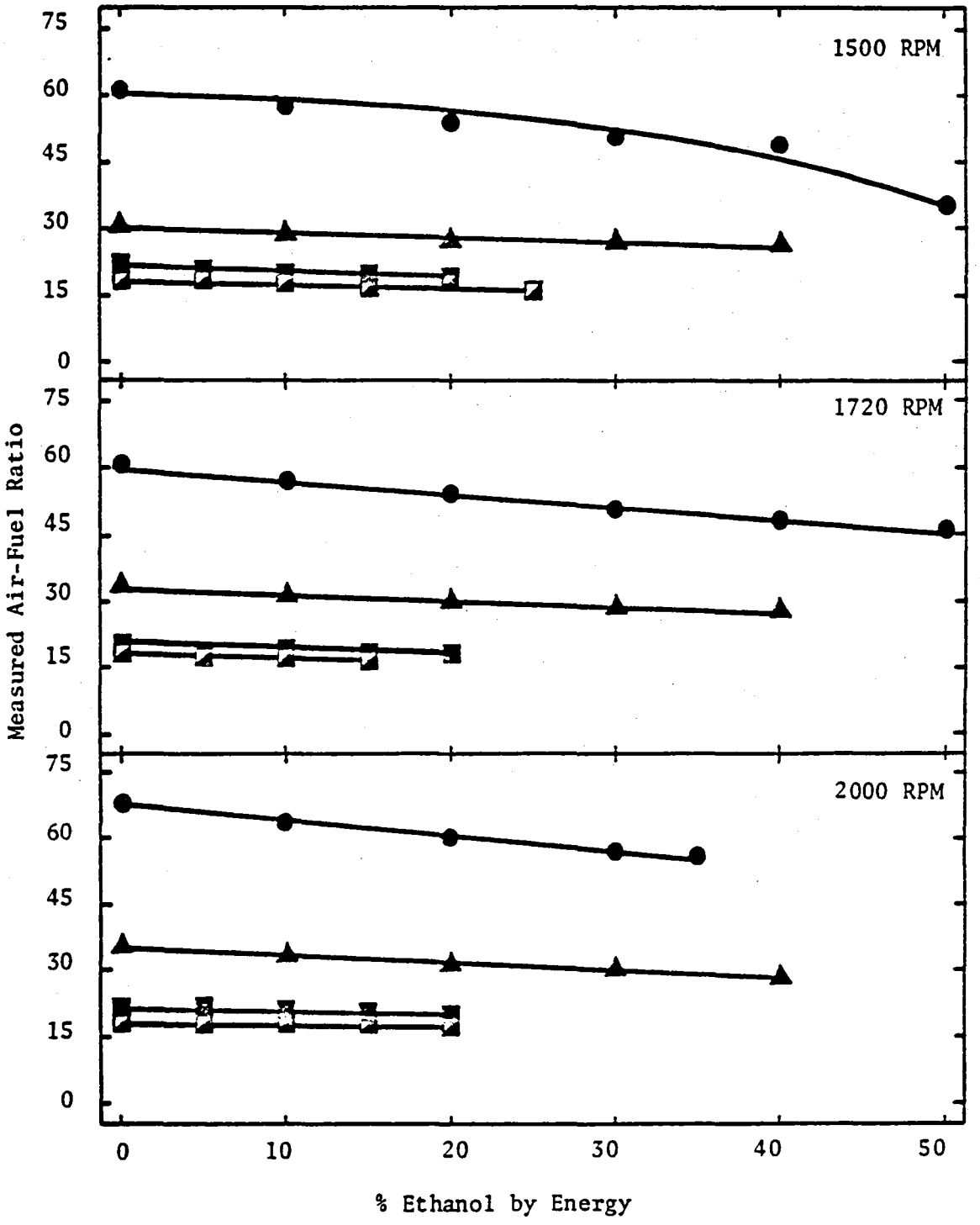


Fig. 4.6 Measured Air-Fuel Ratio as a Function of Fumigated Ethanol

● - 1/4 Rack

▲ - 1/2 Rack

■ - 3/4 Rack

◪ - Full Rack

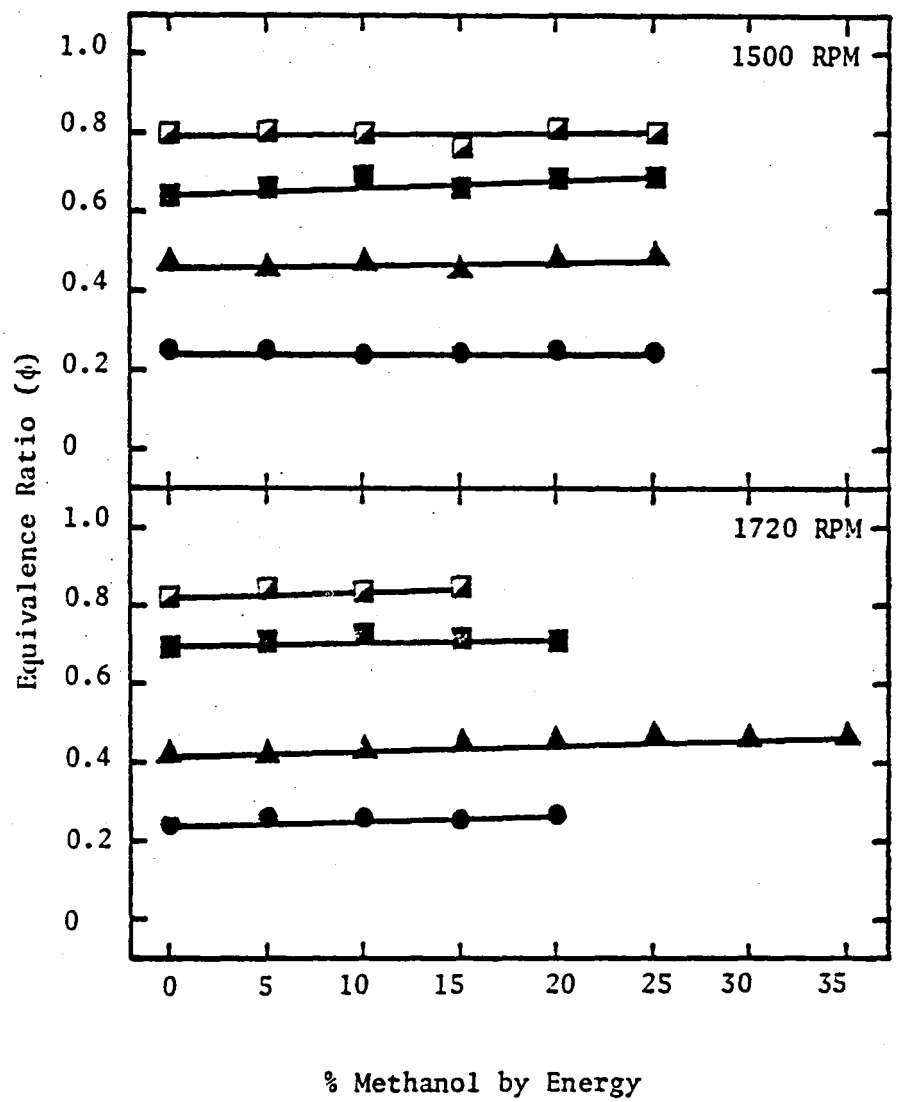


Fig. 4.7 Equivalence Ratio as a Function of Fumigated Methanol

● - 1/4 Rack

▲ - 1/2 Rack

■ - 3/4 Rack

◪ - Full Rack

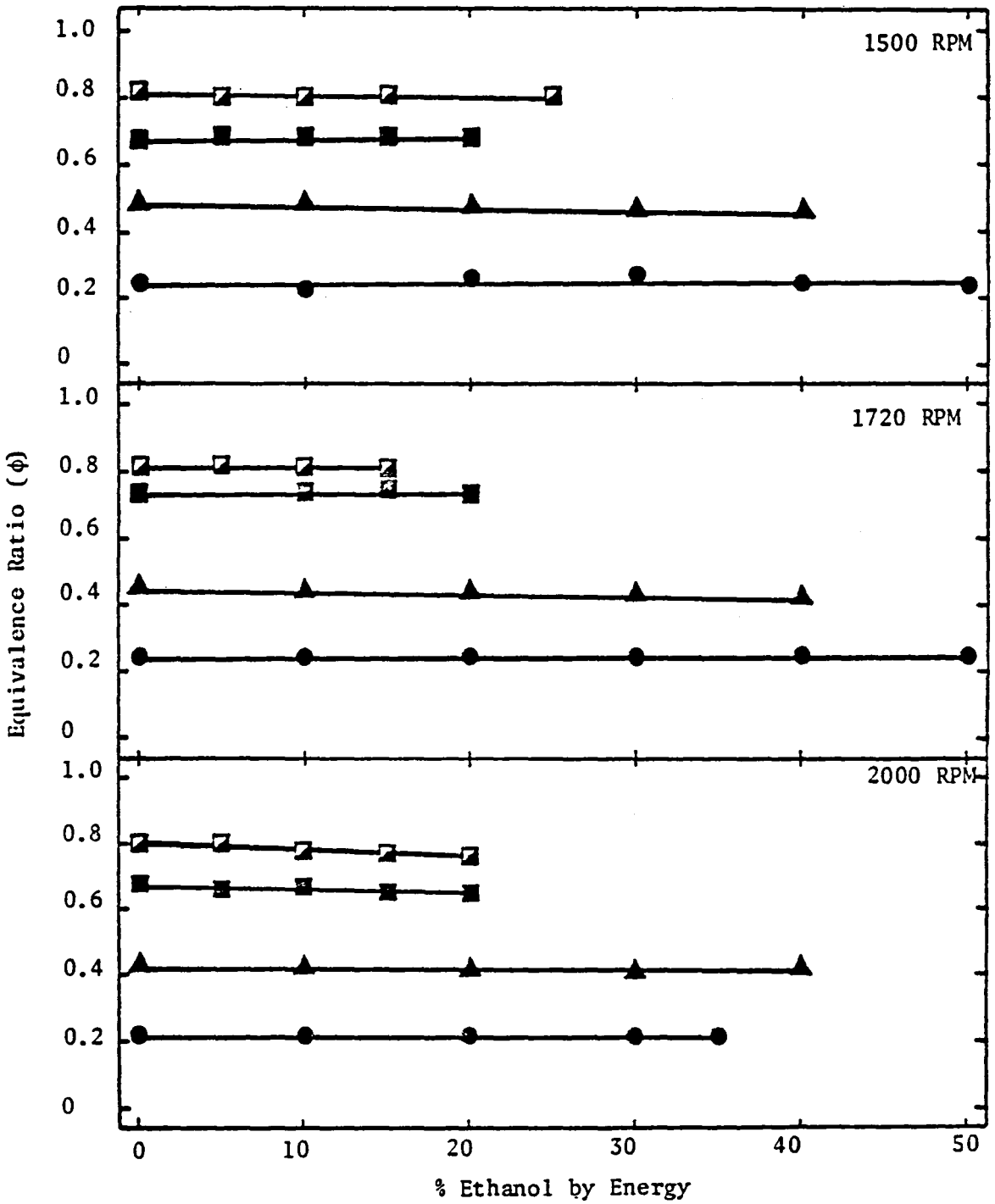


Fig. 4.8 Equivalence Ratio as a Function of Fumigated Ethanol

● - 1/4 Rack

▲ - 1/2 Rack

■ - 3/4 Rack

◪ - Full Rack

operation and 2) after ethanol fumigation for 250 additional hours of engine operation. In both cases, light carbon deposits were found on the cylinder, piston crown and nozzle, but no scoring or pitting was observed. All measurements fell within the manufacturers specifications. Overall engine wear was judged to be normal for 500 hours of operation. That is to say, the alcohol fumigation did not appear to cause any abnormal wear, mainly because fumigation always was done with the engine well warmed up.

4.7 The Effect of Alcohol Fumigation on Exhaust Emissions

In this study measurements were taken to determine the composition of exhaust gas emissions. Emission concentrations are shown in parts per million (PPM) or percent by volume or in gm/kW-hr. Also particulate emissions, their deposition rate and their biological activity are presented.

4.7.1 Smoke Opacity

Figures 4.9 and 4.10 show the effect of alcohol fumigation on smoke opacity at different rack settings and speeds. Here, in order to put these data in proper perspective, it must be pointed out that between three and four percent on this opacity scale represents the point at which the engine exhaust plume becomes visible when viewed against a light background. The following general observations are gleaned from the smoke opacity plots. For methanol fumigation, Fig. 4.9, at 1500 RPM, 1/4 rack setting showed a slight increase but other rack settings (1/2, 3/4 and full rack) showed a slight decrease. At 1720 RPM, 1/4 and 1/2 rack settings showed a slight decrease while a

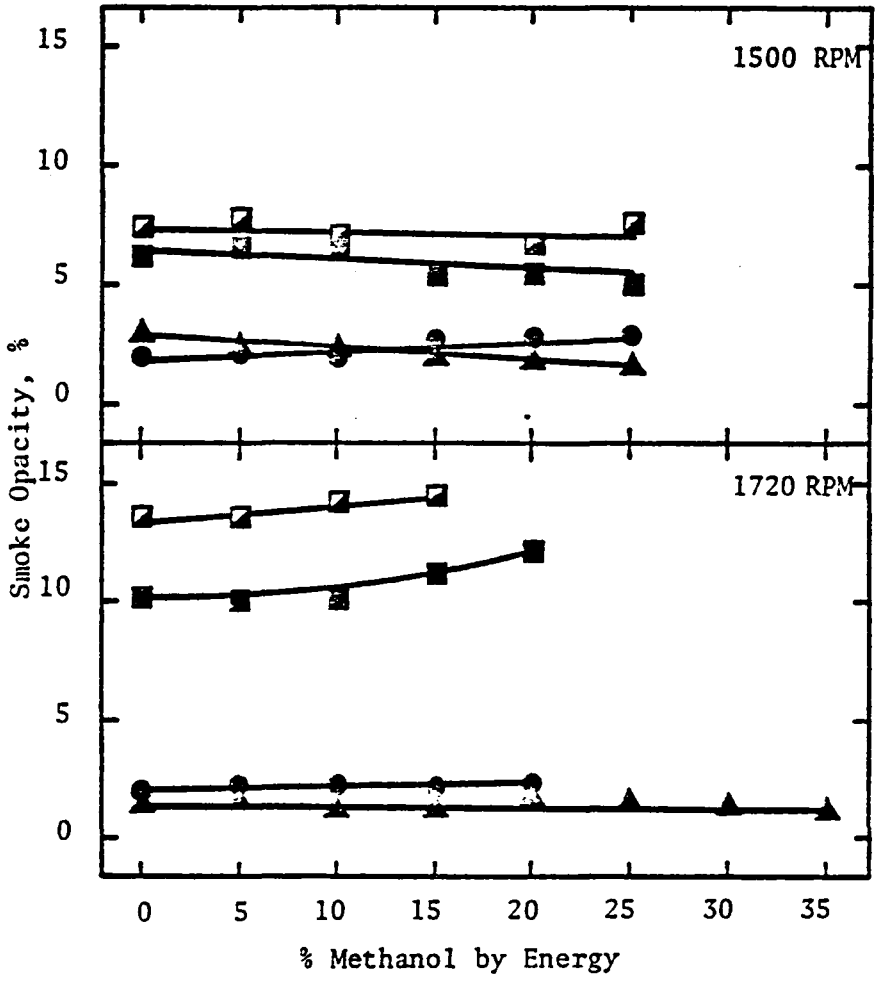


Fig. 4.9 Smoke Opacity as a Function of Fumigated Methanol

● - 1/4 Rack

▲ - 1/2 Rack

■ - 3/4 Rack

◪ - Full Rack

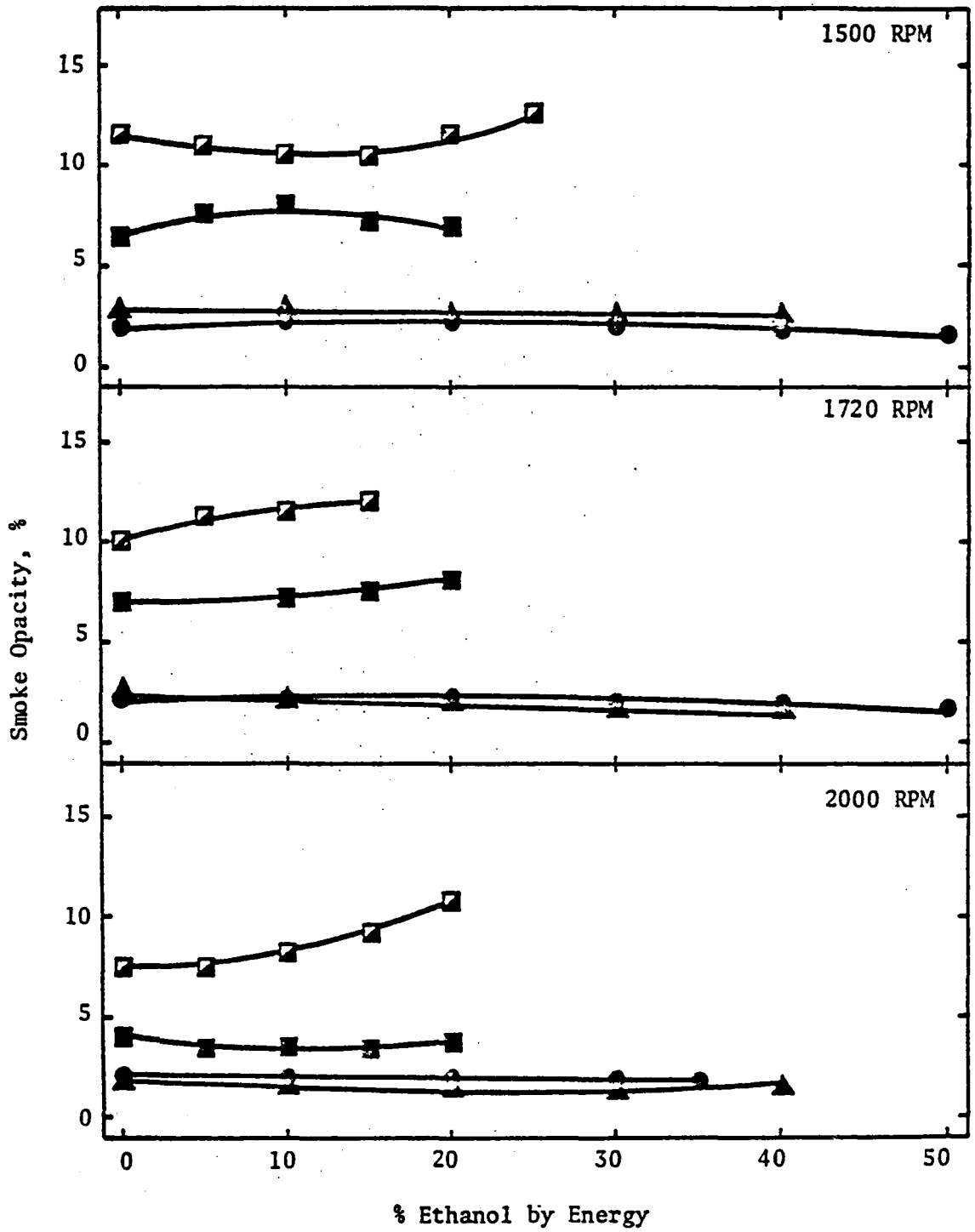


Fig. 4.10 Smoke Opacity as a Function of Fumigated Ethanol

● - 1/4 Rack
▲ - 1/2 Rack
■ - 3/4 Rack
◪ - Full Rack

slight increase is observed at 3/4 and full rack settings. For ethanol fumigation, Fig. 4.10, at the 1/4 and 1/2 rack settings, there appears to be a very slight decrease in smoke opacity as the amount of fumigated ethanol is increased while just the opposite effect, a slight increase, seems to be present at the 3/4 and full rack settings. In overall consideration, methanol and ethanol fumigation do not change the smoke opacity appreciably from that of the baseline condition.

4.7.2 Nitrogen Oxides Emissions

Figures 4.11 through 4.14 show the effect of alcohol fumigation on nitric oxide (NO) and oxides of nitrogen (NO_x) emissions. Figure 4.11 shows the effect of methanol fumigation on NO (PPM) and Figs. 4.12 and 4.13 represent the effect of ethanol fumigation on NO and NO_x (PPM). In both cases, methanol and ethanol fumigation, NO and NO_x decreased for all rack settings and speeds as the amount of alcohol was increased. Figure 4.14 shows the effect of ethanol fumigation on NO_x in gm/kW-hr. At the 3/4 and full rack settings for all speeds NO_x decreased but at the 1/4 and 1/2 rack settings, a slight increase at 10% ethanol fumigation occurs followed by a steady decrease. Table 4.4 represents the NO and NO_x emissions data at baseline, 10%, 20% and 30% ethanol fumigation for all speeds. These data show that at the low rack settings (1/4 and 1/2 rack) nitrogen dioxide (NO_2)* increases dramatically while overall NO and NO_x decreases from the baseline fuel value as a result of ethanol fumigation. It no longer makes up only a small fraction of NO_x, and as seen at some points the NO_x consists of

* NO_x is assumed to be the combination of NO and NO_2 because the other oxides of nitrogen are unstable.

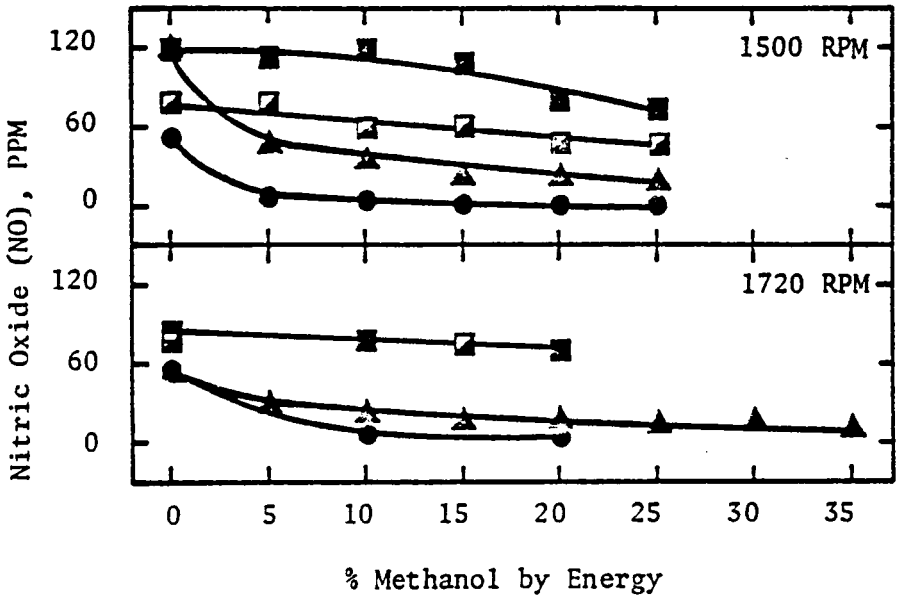


Fig. 4.11 Nitric Oxide Emissions as a Function of Fumigated Methanol

● - 1/4 Rack

▲ - 1/2 Rack

■ - 3/4 Rack

◪ - Full Rack

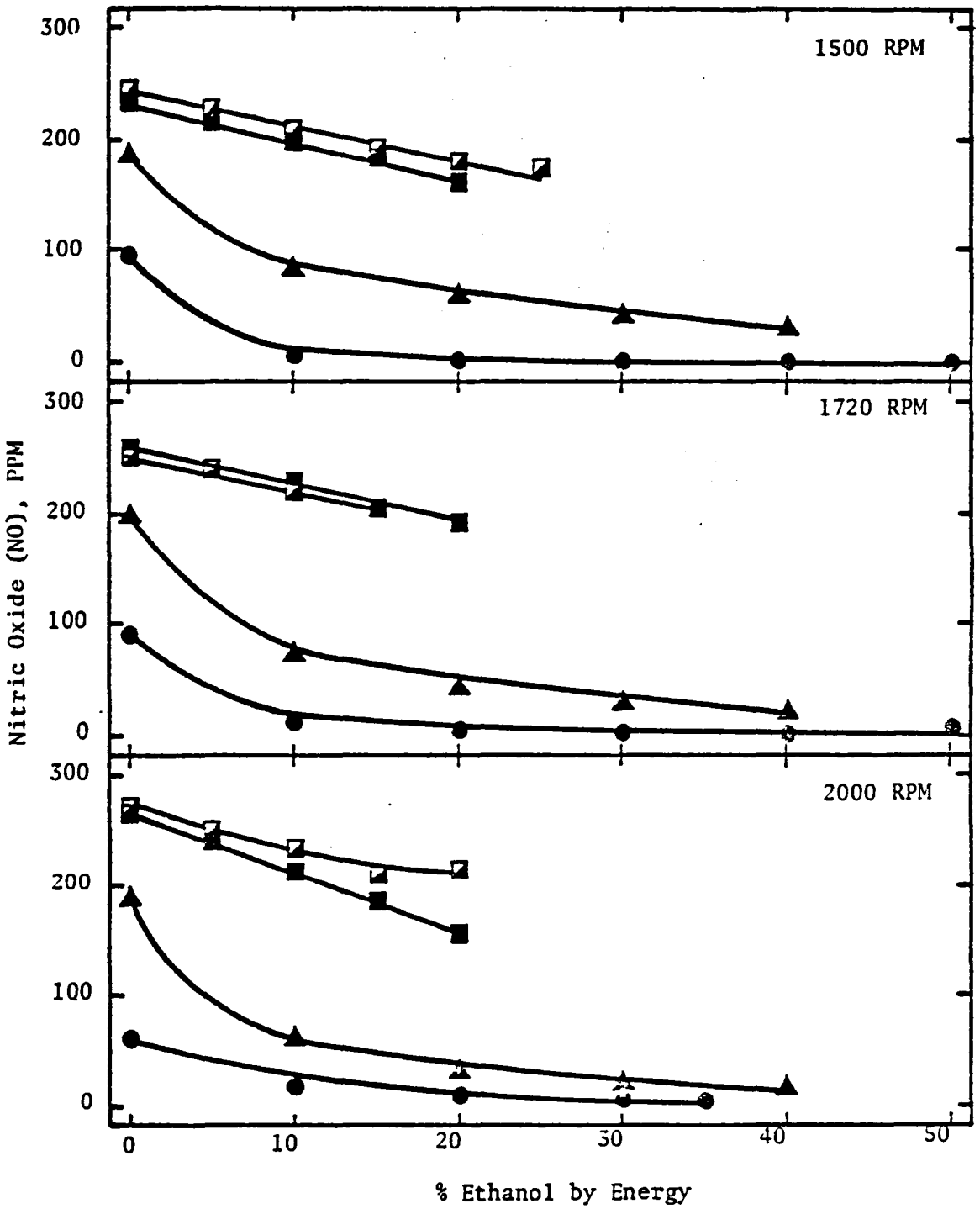


Fig. 4.12 Nitric Oxide Emissions as a Function of Fumigated Ethanol

●- 1/4 Rack

▲- 1/2 Rack

■- 3/4 Rack

◪- Full Rack

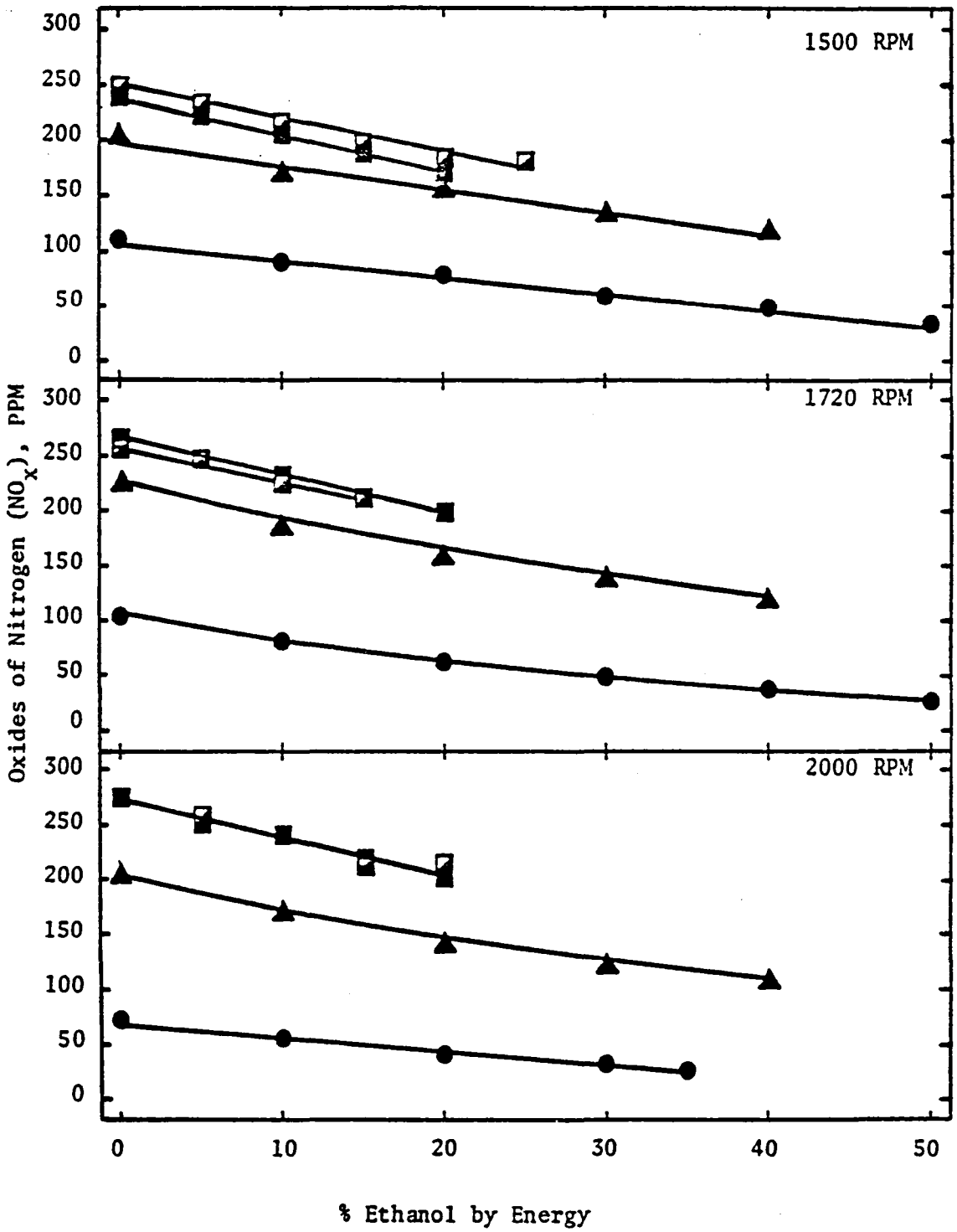


Fig. 4.13 Oxides of Nitrogen Emissions as a Function of Fumigated Ethanol

●- 1/4 Rack

▲- 1/2 Rack

■- 3/4 Rack

◪- Full Rack

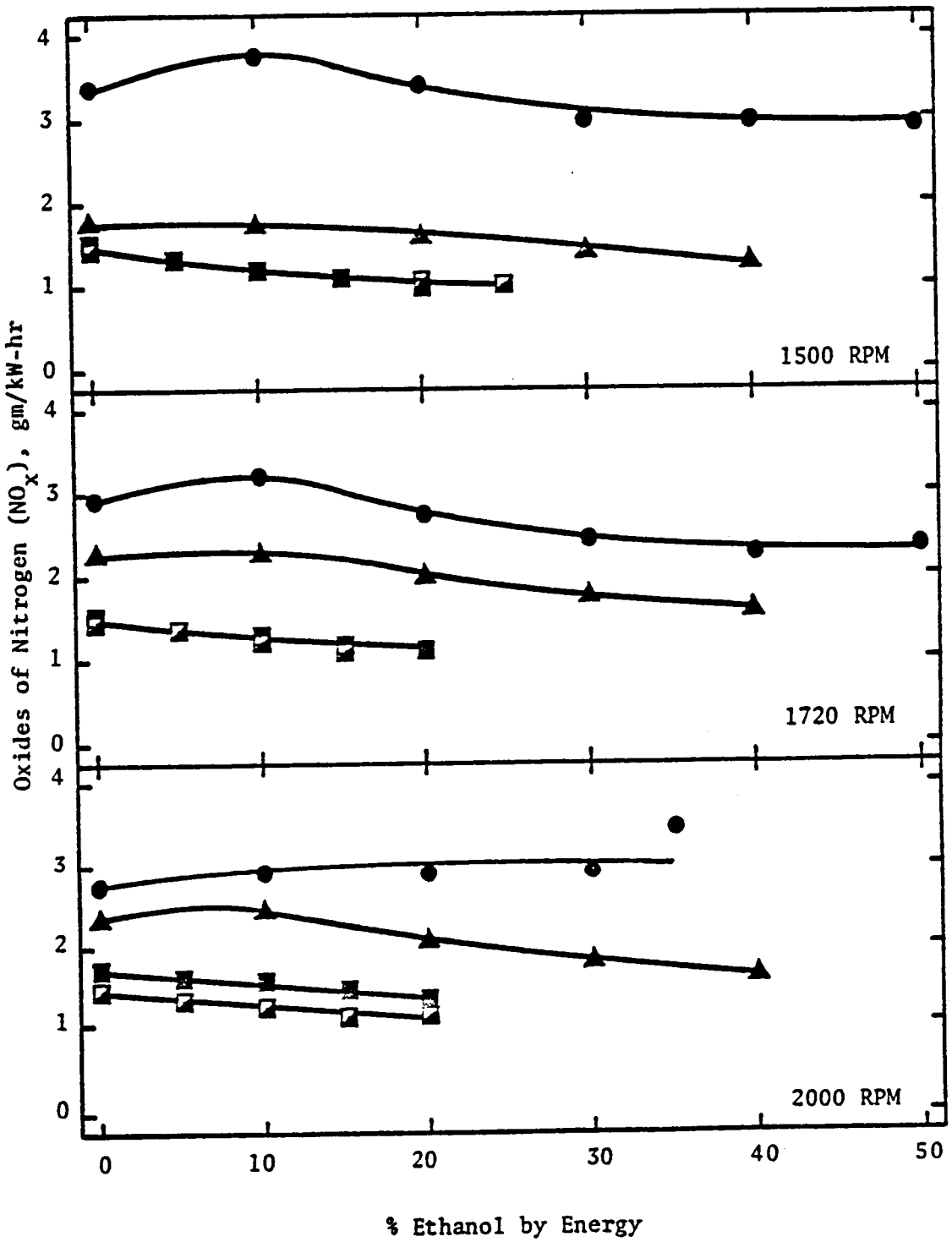


Fig. 4.14 Brake Specific Emission of Oxides of Nitrogen as a Function of Fumigated Ethanol

● - 1/4 Rack

▲ - 1/2 Rack

■ - 3/4 Rack

◪ - Full Rack

Table 4.4 - Oxides of Nitrogen Emissions in PPM

RACK	RPM	PERCENT OF ETHANOL BY ENERGY															
		0				10				20				30			
		NO	NO_x	NO_2	NO_2/NO_x	NO	NO_x	NO_2	NO_2/NO_x	NO	NO_x	NO_2	NO_2/NO_x	NO	NO_x	NO_2	NO_2/NO_x
1/4	1500	97.3	116.3	19	0.16	7	89.9	82.9	0.92	2.76	77.7	74.94	0.96	1	58.5	57.5	0.98
	1720	91	103.5	12.5	0.12	11.75	80.42	68.67	0.85	4.67	61.25	56.58	0.92	2.83	47.83	45	0.94
	2000	61.2	71.8	10.6	0.15	18.4	54.2	38.8	0.66	9.4	41	31.6	0.77	5.4	32	26.6	0.83
1/2	1500	187.6	204.2	16.6	0.08	83.8	171.2	87.4	0.51	59.4	154.2	94.8	0.61	41.6	133	91.4	0.69
	1720	198.5	223	24.5	0.11	71.8	184.4	112.6	0.61	43.6	157.6	114	0.72	28.8	136.2	107.4	0.79
	2000	188	205.7	17.7	0.09	61.2	170.2	109	0.64	32.6	139.4	106.8	0.77	23.6	120.2	96.6	0.80
3/4	1500	234.3	239	4.7	0.020	200	203.4	3.4	0.017	161.6	169.4	7.8	0.046	--	--	--	--
	1720	259	264	5	0.019	228	231	3	0.013	193	199	6	0.030	--	--	--	--
	2000	266.7	274.2	7.5	0.027	213.3	240	26.7	0.111	156.4	200.7	44.3	0.221	--	--	--	--
FULL	1500	245	250	5	0.020	210	215	5	0.023	180	185	5	0.027	--	--	--	--
	1720	253	258.3	5.3	0.021	220	225	5	0.022	--	--	--	--	--	--	--	--
	2000	270	273.8	3.8	0.014	254	258	4	0.016	216.3	219.8	3.5	0.016	--	--	--	--

more than 90% NO_2. Also at low rack settings (1/4 and 1/2 rack) fumigation of ethanol makes the ratio NO_2/NO large compared to the small fraction seen for baseline operation. Figure 4.15 shows the comparison between the variation of NO_2/NO as a function of rack setting for baseline operation and 20% ethanol fumigation at 2000 RPM.

4.7.3 Carbon Monoxide Emissions

Substitution of alcohol (methanol and ethanol) for fuel oil showed an increase in carbon monoxide (CO) emission for all rack settings and speeds. In all cases the baseline value for CO was quite low which is typical for a Diesel engine. Results are shown in Figs. 4.16 and 4.17.

4.7.4 Unburned Hydrocarbon Emissions

Figures 4.18 and 4.19 show that, in general unburned hydrocarbon emissions increased as the amount of fumigated ethanol was increased.

4.7.5 Particulate Emissions

Because of the large amount of soot formation in Diesel engines and its possible effect on human health, the effect of ethanol fumigation on soot formation was measured and the biological activity of the raw particulate and its SOF assayed. The results for ethanol fumigation at 1/2 rack for all speeds are presented. As Table 4.5 shows ethanol fumigation reduces the deposition rate below the baseline in most cases. The biological activity of the particulate emissions was measured by the Ames *Salmonella typhimuruim* test. Results are also presented in Table 4.5 and as seen ethanol fumigation enhances the

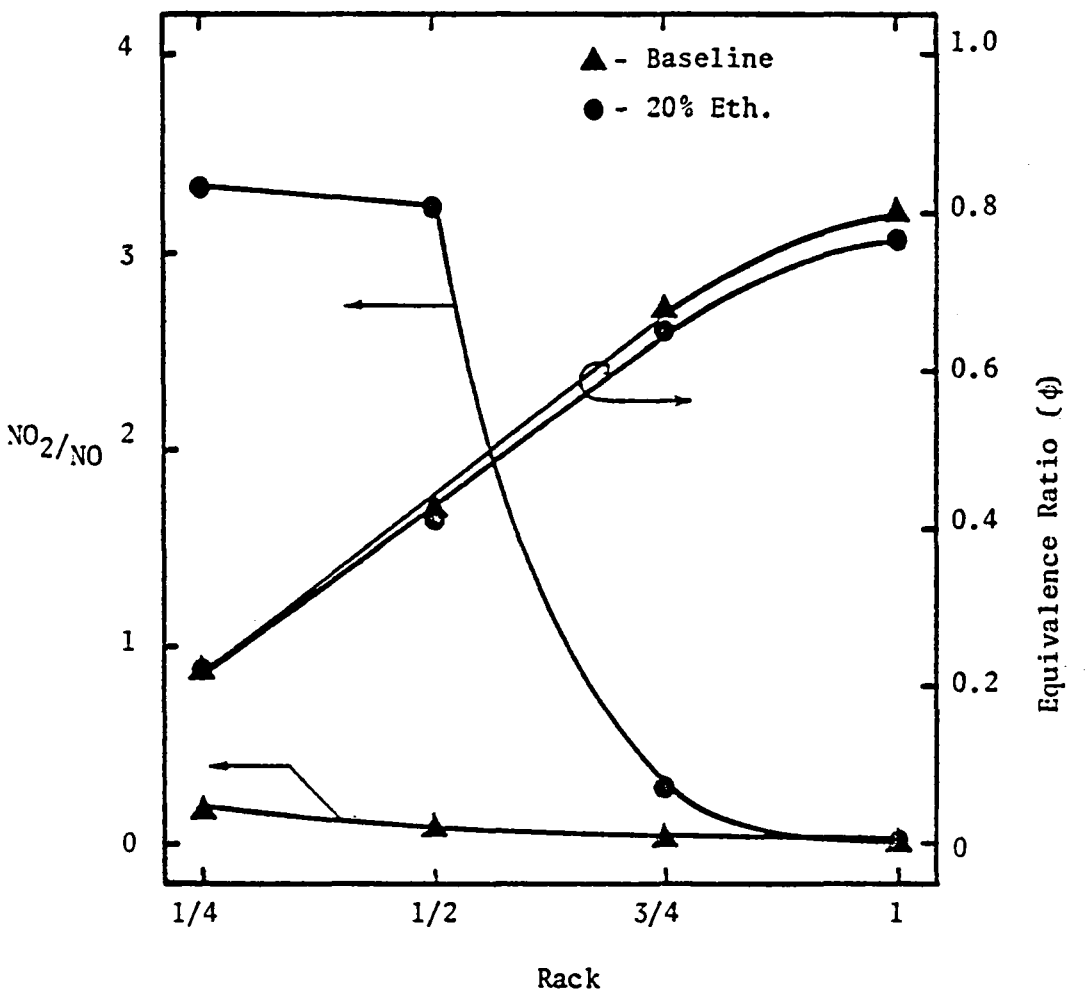


Fig. 4.15 Ratio of Nitrogen Dioxide to Nitric Oxide Production as a Function of Load at 2000 RPM

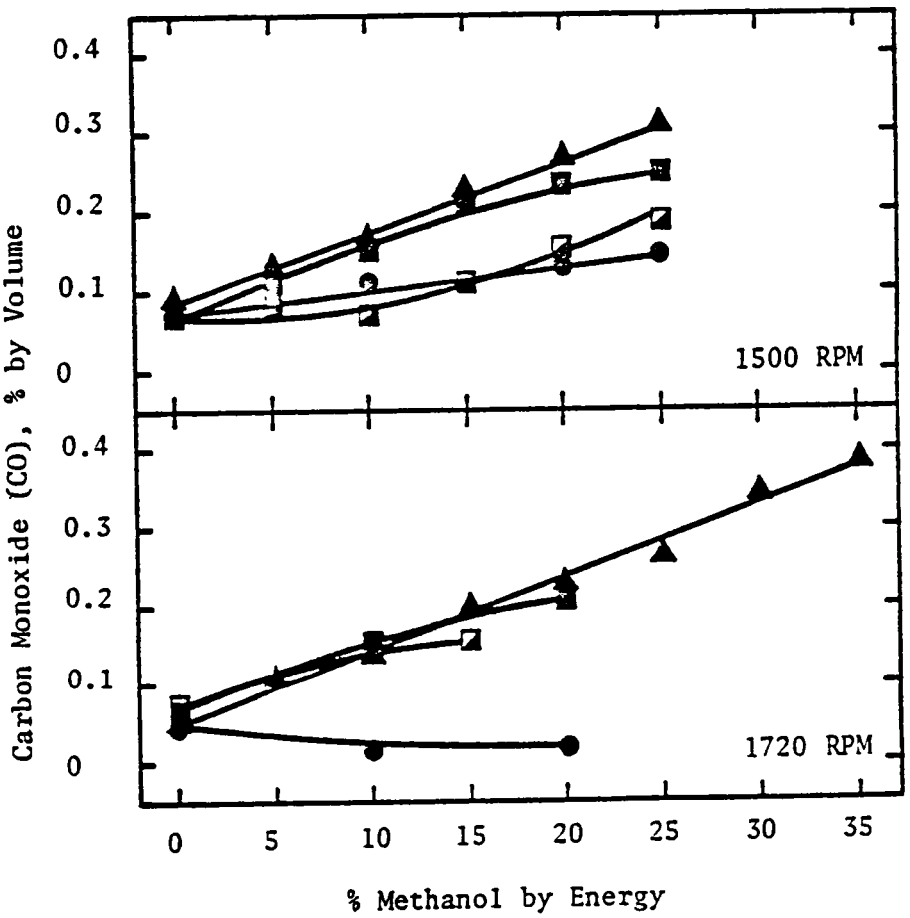


Fig. 4.16 Carbon Monoxide Emissions as a Function of Fumigated Methanol

● - 1/4 Rack

▲ - 1/2 Rack

■ - 3/4 Rack

◪ - Full Rack

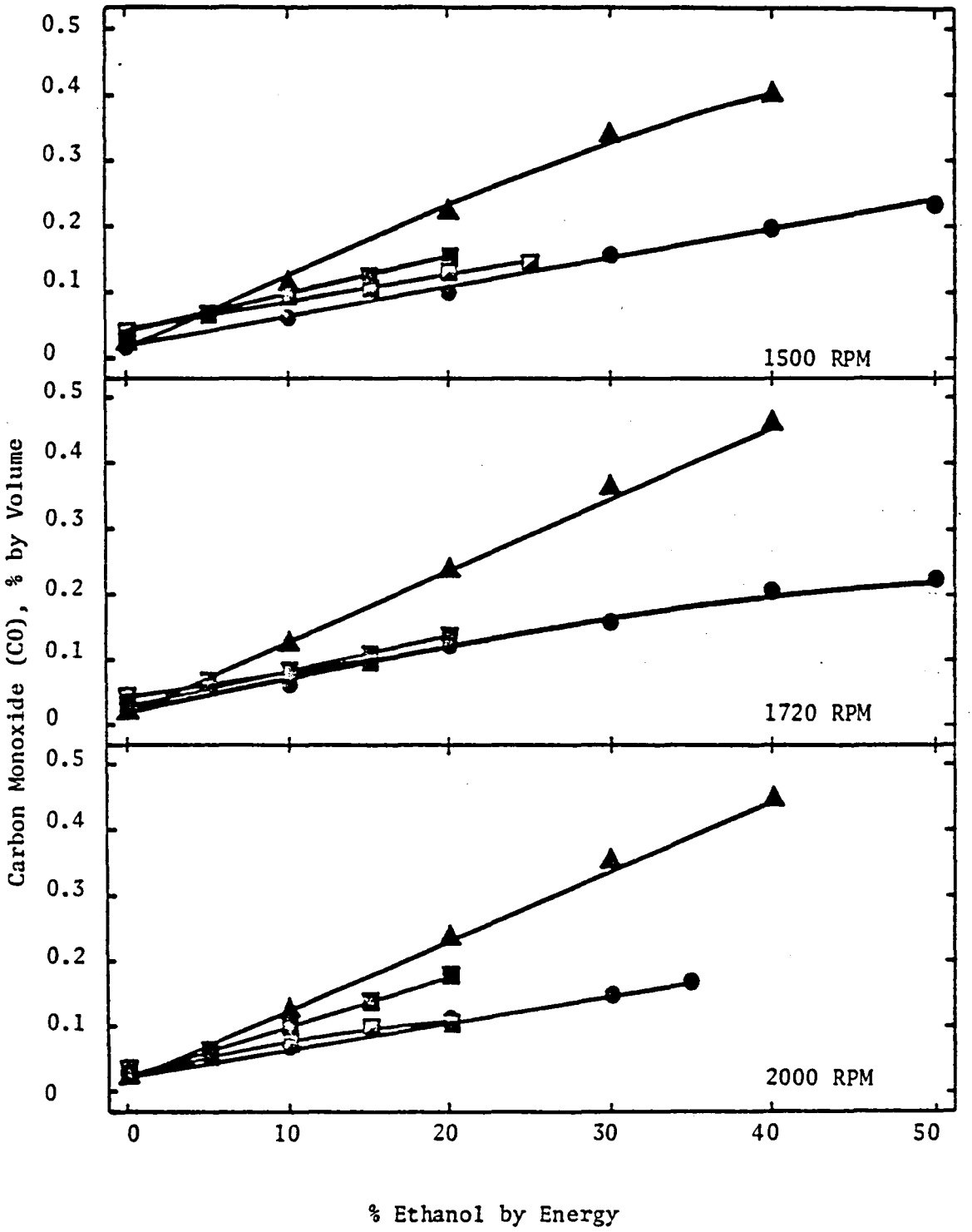


Fig. 4.17 Carbon Monoxide Emissions as a Function of Fumigated Ethanol

- ● - 1/4 Rack
- ▲ - 1/2 Rack
- ■ - 3/4 Rack
- ◪ - Full Rack

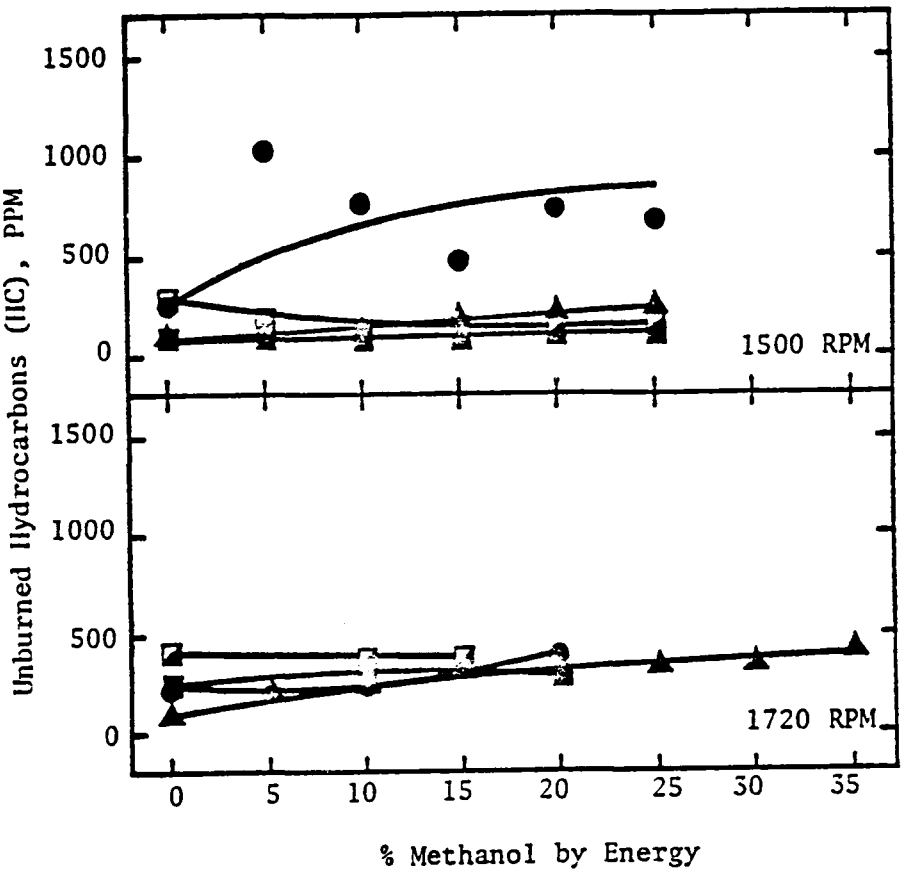


Fig. 4.18 Unburned Hydrocarbon Emissions as a Function of Fumigated Methanol

●- 1/4 Rack

▲- 1/2 Rack

■- 3/4 Rack

◪- Full Rack

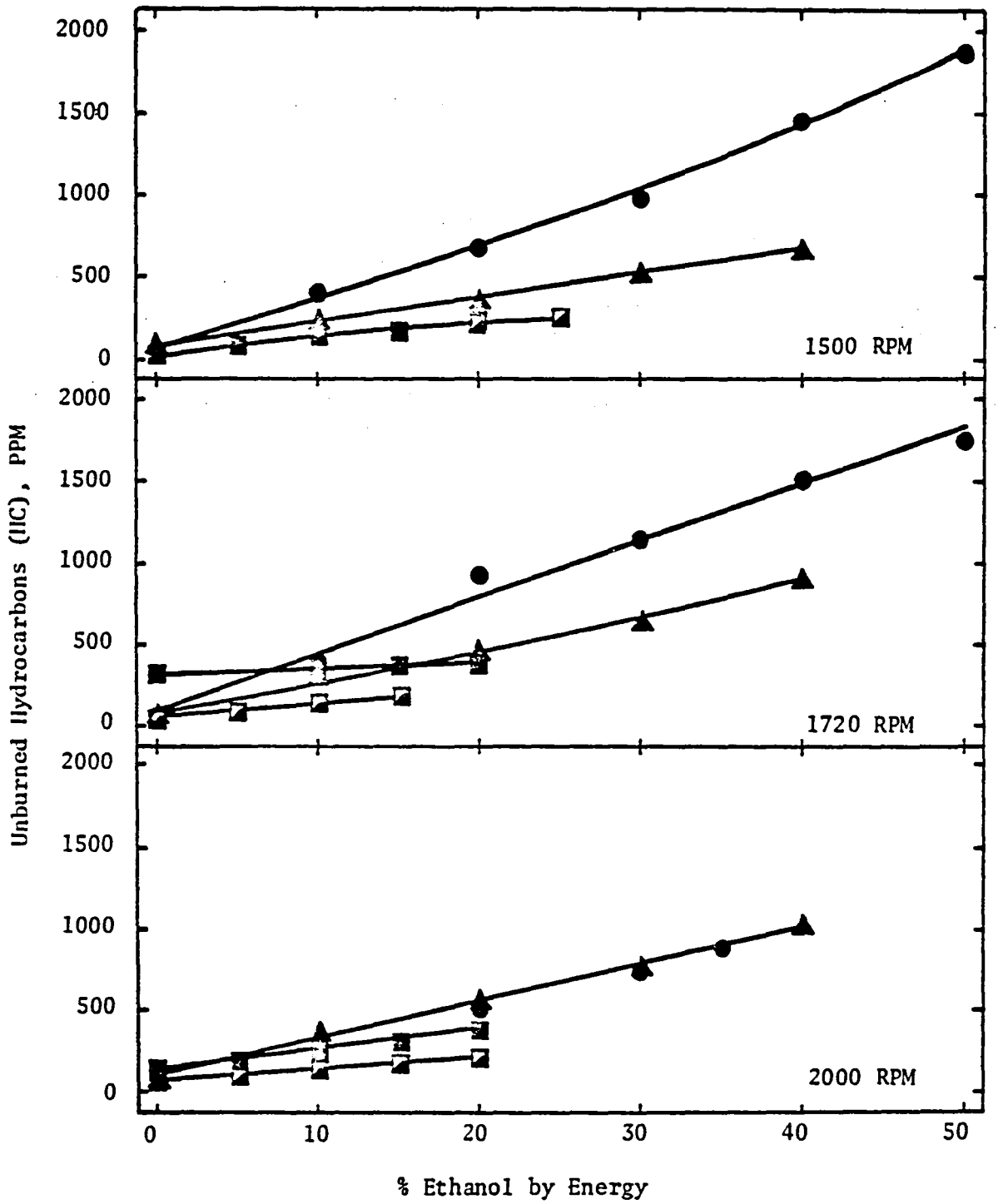


Fig. 4.19 Unburned Hydrocarbon Emissions as a Function of Fumigated Ethanol

● - 1/4 Rack

▲ - 1/2 Rack

■ - 3/4 Rack

◪ - Full Rack

Table 4.5 - Summary of Particulate Data

RPM \ RACK	1/4	1/2			3/4	FULL
2000	3265.* 0 1.9392 56.81 0.29 ±0.02 2.9	6302. 0 1.7245 28.24 0.13 ±0.01 1.3 ±0.3	6302. 20 1.6452 43.07 0.4 2.7 ±0.2	6302. 30 1.7252 50.54 0.57 ±0.07 2.4 ±0.1	9535. 0 2.8300 6.63 NS+ 1.75 ±0.6	11348. 0 3.2250 6.18 NS 1.6 ±0.1
1720		5751. 0 2.5278 28.22 NS 1.35 ±0.05	5751. 20 1.9325 52.75 0.24 1.8	5751. 30 2.3450 59.90 0.7 3.1		
1500		5360. 0 2.7134 19.47 NS 2.2	5360. 20 2.2375 26.30 NS 2.8 ±0.2	5360. 30 2.4925 31.88 0.29 2.9		

* Data in each block is tabulated as follows:
Total fuel energy input rate - btu/min
Percent of total fuel energy input as ethanol
Particulate deposition rate - mg/min
SOF percent
Ames Test Results, TA98, mean of slope ± stand. dev. (rev/μg)
Raw
SOF

\+ NS - Not significant < 0.1 rev/mg

biological activity in the raw particulate matter and its SOF. Figure 4.20 graphically shows enhancement caused by ethanol fumigation at 1/2 rack setting and 1720 RPM.

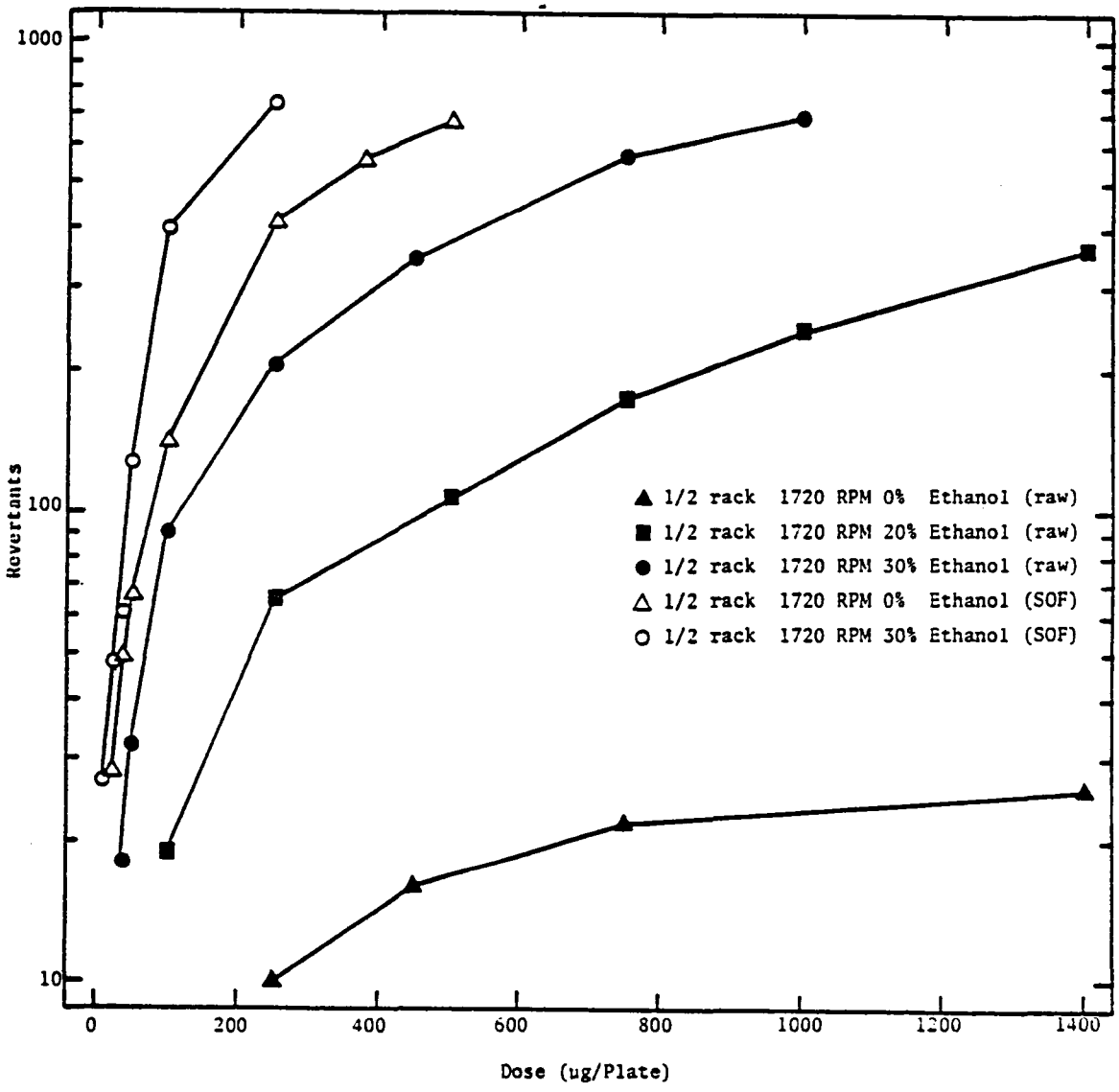


Fig. 4.20 Comparative Ames Test Results Illustrating the Increased Biological Activity Caused by Ethanol Fumigation.

CHAPTER V

DISCUSSION OF RESULTS
SUMMARY AND CONCLUSIONS

5.1 Introduction

The purpose of this chapter is to analyze the data presented and discuss possible correlations between the data and the theoretical considerations outlined in Chapter II. These data show the effects of ethanol and methanol fumigation on the performance, combustion knock characteristics and exhaust emissions of the Oldsmobile V-8 Diesel engine. To aid in this analysis, figures of combustion pressure traces, injector needle lift, and top-dead-center (TDC) marks are presented to establish their relative timing in the combustion cycle.

5.2 Knock

Figure 5.1 presents the combustion pressure data taken at the 1/4 rack, 2000 RPM test point, comparing the baseline fuel condition with that of 35% ethanol substitution. At this test point, eventually engine misfiring prevented further ethanol substitution. Examination of the needle lift traces shows a slight injection timing difference between the baseline and ethanol substitution conditions. This is attributed to the load-sensitive injection timing curve of the injection pump; that is, as the physical rack setting is decreased to permit ethanol substitution, the pump compensates for a perceived load reduction. Results of the ethanol introduction include charge cooling with a corresponding pressure drop at TDC. The ignition delay was observed to increase significantly, with combustion beginning well into the

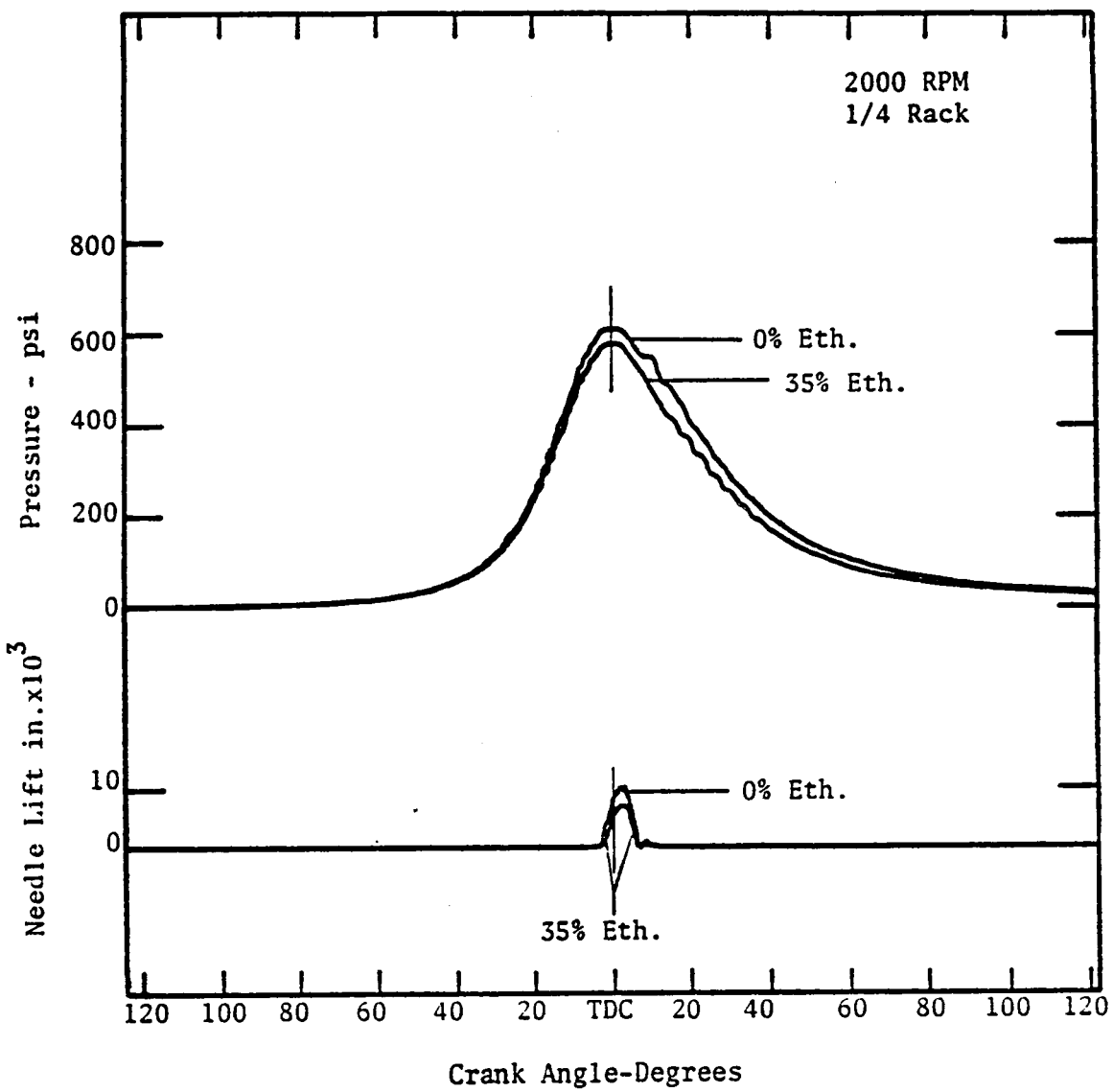


Fig. 5.1 Comparison of Representative Pressure and Needle Lift Histories for Baseline (0% Eth.) and 35% ethanol (35% Eth.) by Energy Tests.
Operating Condition: 1/4 Rack, 2000 RPM

expansion stroke; as expected, no knock was observed at this condition. Operation at the 1/4 rack, 1500 and 1720 RPM with methanol and ethanol substitution yielded similar results, no knock was detected.

The 1/2 rack, 2000 RPM condition produced different, though theoretically consistent, combustion pressure data (Fig. 5.2). Substitution of 40% ethanol by energy did not cause a significant reduction in peak pressures as compared with baseline fuel operation. The higher cylinder temperatures at this load condition reduced the ignition delay compared with 1/4 rack operation; consequently, combustion began sooner after TDC, causing a sudden pressure rise and rough combustion, which limited the ethanol substitution to 40% for all conditions. Methanol fumigation showed the same trend, but no rough combustion was detected.

At the 3/4 rack, 2000 RPM condition, the occurrence of severe knock limited ethanol substitution to 20% by energy. Figure 5.3 shows the very short ignition delay, rapid pressure rise, and higher peak pressures which characterized the engine operation at this level of alcohol substitution. The homogeneous charge of alcohol and air burned very rapidly at the elevated cylinder temperatures of the 3/4 rack condition. Similar phenomena were observed when operating at 1500 and 1720 RPM with ethanol fumigation and 1720 RPM with methanol fumigation. The 1500 RPM methanol-fumigated condition did not reach the knock-limited point.

There is almost no ignition delay at the full rack, 2000 RPM condition (Fig. 5.4). The extremely rapid combustion pressure rise again caused knock-limited operation with 20% ethanol substitution; it appears that the mixture of air and ethanol may have ignited slightly

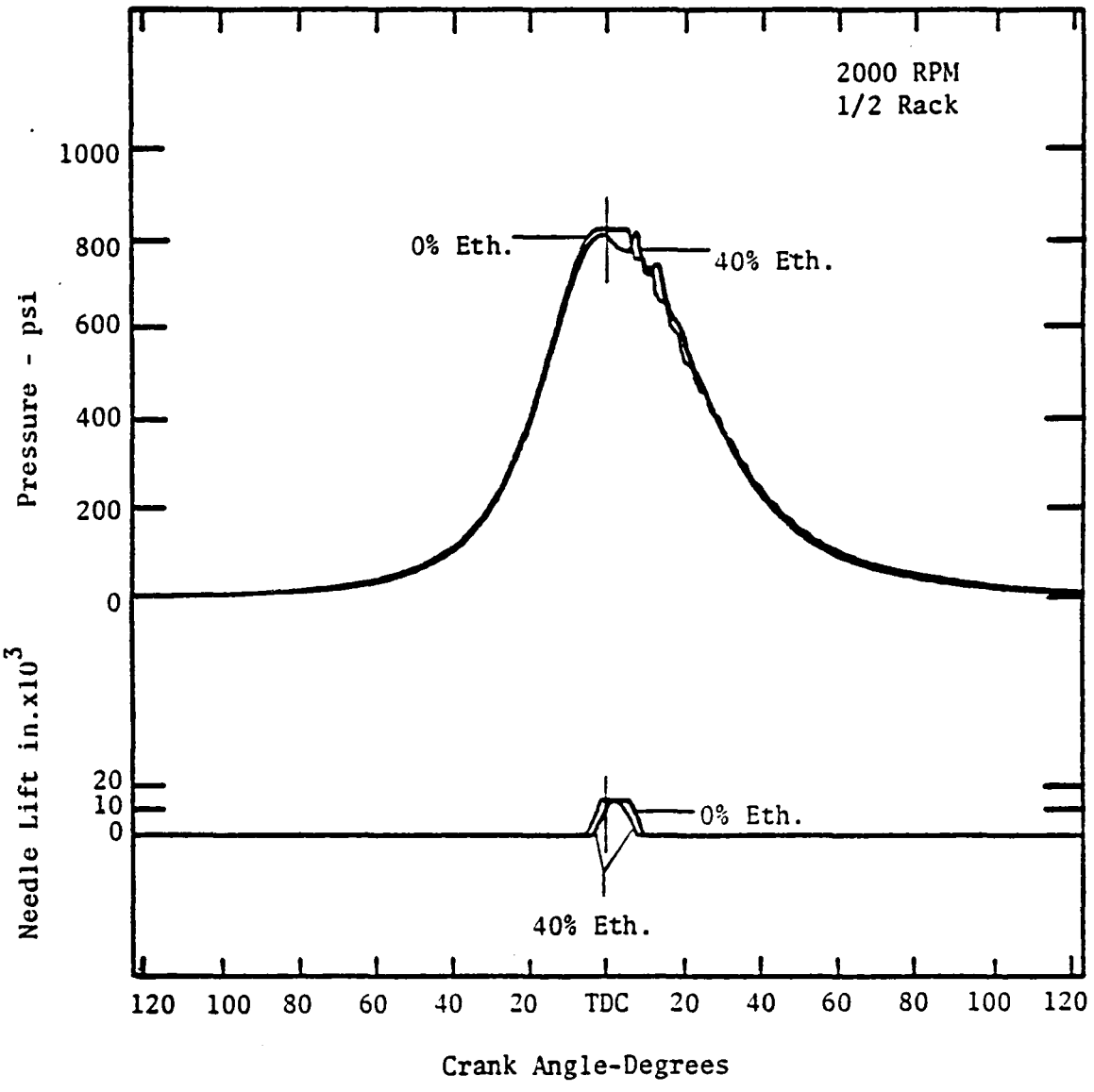


Fig. 5.2 Comparison of Representative Pressure and Needle Lift Histories for Baseline (0% Eth.) and 40% Ethanol (40% Eth.) by Energy Tests.
Operating Condition: 1/2 Rack, 2000 RPM

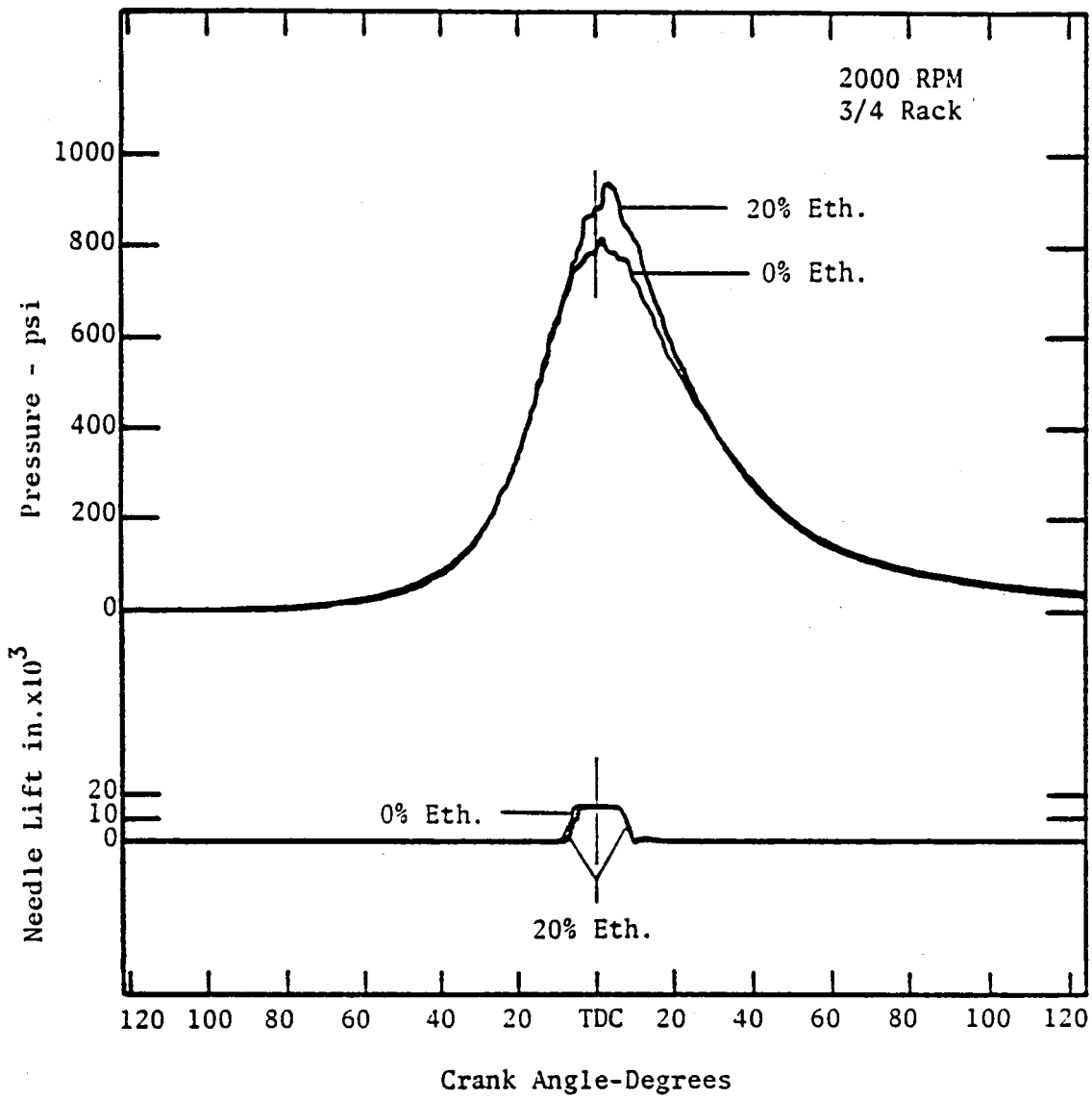


Fig. 5.3 Comparison of Representative Pressure and Needle Lift Histories for Baseline (0% Eth.) and 20% Ethanol (20% Eth.) by Energy Tests.
Operating Condition: 3/4 Rack, 2000 RPM

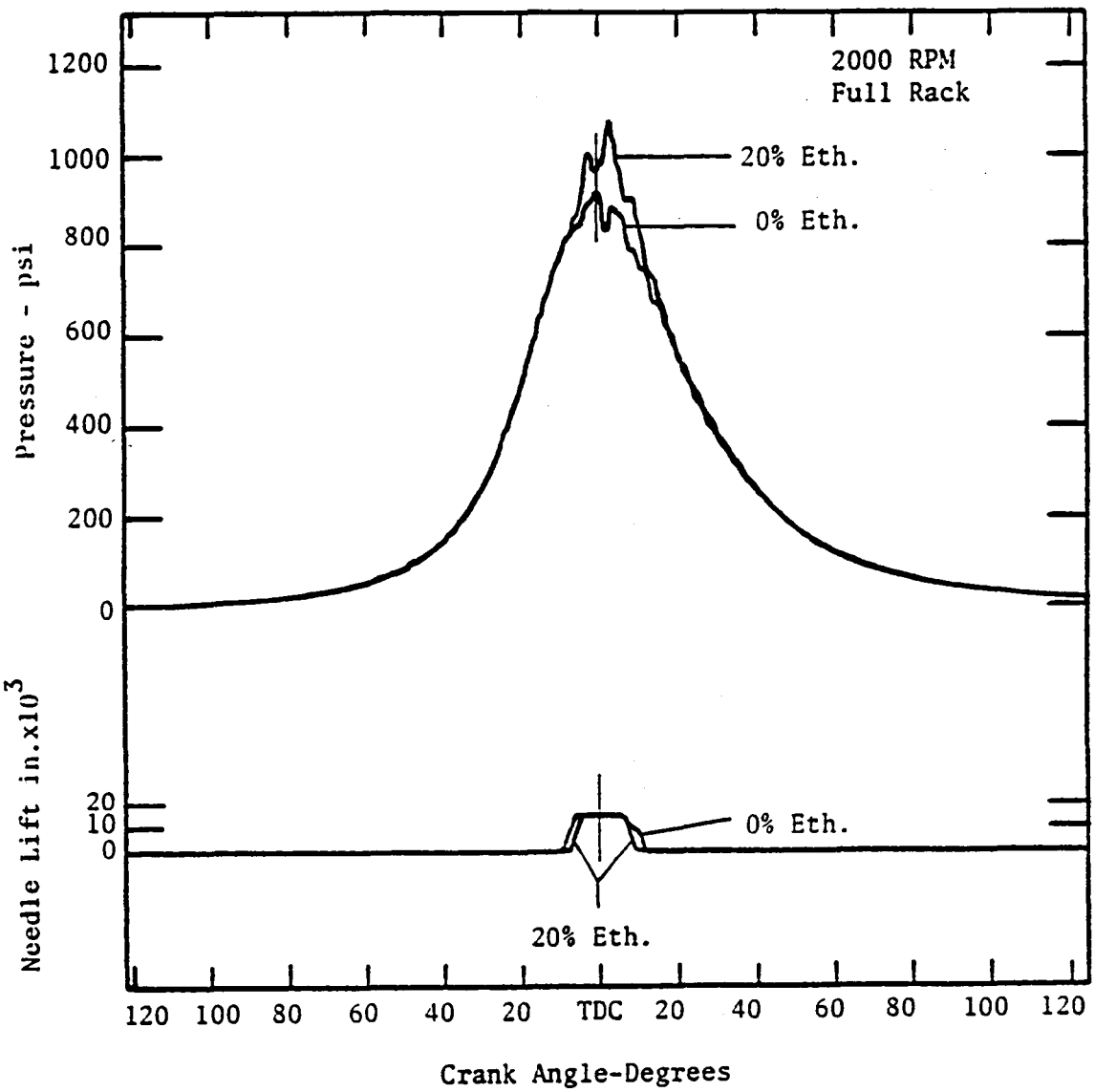


Fig. 5.4 Comparison of Representative Pressure and Needle Lift Histories for Baseline (0% Eth.) and 20% Ethanol (20% Eth.) by Energy Tests.
Operating Condition: Full Rack, 2000 RPM

before fuel oil injection began. As before, data from other speeds support these conclusions for ethanol substitution and for methanol substitution at 1720 RPM. Methanol substitution at 1500 RPM did not reach knock-limited operation.

5.3 Thermal Efficiency

Although thermal efficiency decreased with increasing ethanol substitution at the 1/4 rack condition, all other rack settings showed an increase in thermal efficiency with increasing ethanol substitution. The decrease at the 1/4 rack is attributed to the long ignition delay and burning during the expansion stroke which result from charge cooling. At the 1/2 rack condition, this effect is minimized by the higher cylinder temperatures.

As cylinder temperatures increase at the higher rack settings (3/4 and full rack), the dissociation of ethanol to ethylene (C_2H_4) and water may complement the shorter ignition delays to cause higher efficiencies. The high flame speed of ethylene may result in faster overall combustion with correspondingly less heat transfer from the cylinder; this nearly constant-volume combustion has a beneficial effect on thermal efficiency.

Increasing methanol fumigation resulted in higher thermal efficiency at high rack settings (3/4 and full rack). The high temperature at these high rack settings may cause the dissociation of methanol to hydrogen (H_2) and carbon monoxide (CO). The high flame speed of H_2 may result in faster overall combustion with correspondingly less heat transfer from the cylinder and nearly constant-volume combustion which has a beneficial effect on thermal efficiency.

Thermal efficiency dropped at the 1/4 and 1/2 rack settings for 1500 and 1720 RPM. The higher heat of vaporization of methanol compared to ethanol may cause the decrease in thermal efficiency at the 1/2 rack, methanol fumigated conditions compared to similar ethanol fumigated conditions.

5.4 Emissions

Section 4.5 presented the effect of alcohol substitution on A/F ratio and equivalence ratio [(F/A) act./(F/A) stoich.]. Figures 4.5 and 4.6 showed alcohol fumigation decreased A/F ratio. It is known that equivalence ratio has a significant effect on gaseous and particulate emissions. Methanol and ethanol, because of their heating values which are lower than Diesel fuel oil, necessitated more mass substitution to maintain a constant energy input. However, the stoichiometric A/F ratio also decreased because of the existence of oxygen in the alcohol molecule. Therefore, the equivalence ratio remained nearly constant (Figs. 4.7 and 4.8).

This brief discussion points out that factors other than mixture composition were responsible for the changes in gaseous and particulate emissions. The homogeneous mixture of alcohol and air and the heterogeneous combustion of Diesel fuel oil must be taken in account.

5.4.1 Gaseous Emissions

At the 1/4 and 1/2 rack conditions, smoke opacity decreased as larger amounts of ethanol were fumigated. As previously stated, charge cooling increased ignition delays; this enhances fuel oil, ethanol, and

air mixing and allows better air utilization and less smoke. Also, the effect of substituting clean burning ethanol for fuel oil must be noted. The same result was observed at the 1/2 rack, methanol fumigated condition. A small increase in smoke opacity at the 1/4 rack methanol fumigated condition may be the result of deleterious effects of methanol on combustion which dominates the effect of a more homogeneous mixture.

At higher rack settings, ignition delays are characteristically short and the rapidly burning homogeneous mixture of alcohol and air tends to deprive the slower burning fuel oil of air. As expected, the smoke opacity usually increased (Figs. 4.9 and 4.10).

Ethanol and methanol fumigation reduced NO and NO_x emissions on a volume basis. The cooling effect of alcohol lowers the combustion temperature which results in the reduction of NO and NO_x emissions at 1/4 and 1/2 rack settings. Although alcohol fumigation at 3/4 and full rack settings cools down the combustion temperature somewhat, the high engine temperature causes the alcohol and air mixture to burn near or prior to injection and consume some of the oxygen, reducing NO and NO_x formation (Figs. 4.11 through 4.13). Since the mass of NO_x varies directly with the weighted-averaged molecular weight of all the oxides of nitrogen (mainly NO and NO_2 present), a small increase in the specific NO_x curves (Fig. 4.14) at the 1/4 and 1/2 rack settings occurs because of production of a large amount of NO_2. The amount of NO_2 formation at the 3/4 and full rack settings is very small but at the 1/4 and 1/2 rack settings, NO_2 makes up the major part of NO_x (Table 4.4 and Fig. 4.15).

Alcohol fumigation caused an increase in CO emissions for all conditions. The cooling effect of alcohol causes a lower combustion temperature and should reduce the CO emissions. However, increasing flame and wall quench and a homogeneous alcohol-air charge tends to increase CO emissions (Figs. 4.16 and 4.17), resulting in an overall increase in CO emissions.

Unburned hydrocarbon emissions showed the same trend as CO emissions. Since in this study, hydrocarbon emissions were measured with a cold FID and the effect of alcohol and oxygen concentration on FID response were not fully understood, these data are not as accurate as they could be. However, from the trend which is represented in Figs. 4.18 and 4.19, the cooling effect of alcohol seems to increase the quench layer thickness which causes an increase in HC emissions. The rate of increase decreases at higher rack settings which can possibly be the result of higher temperatures and high flame speeds which reduce the quench layer thickness; however, this effect is not as great as the cooling effect of alcohol, resulting in an overall increase in HC emissions.

5.4.2 Particulate Emissions

The particulate deposition rate generally decreased with ethanol fumigation which can be attributed to the almost sootless burning of ethanol and also introduction of part of the fuel as a homogeneous charge.

The biological activity of the raw particulate matter and its SOF appears to have been enhanced by ethanol substitution (Table 4.5 and Fig. 4.20), but this increase was not as great as for methanol

substitution done by Houser et al. (47). Analysis of the exact mechanisms which are responsible for this increased biological activity are outside the scope of this study.

5.5 Summary and Conclusions

In order to conserve petroleum fuels, the feasibility of burning alcohol in Diesel engines is being considered. This work was undertaken to study the effects of alcohol fumigation on the performance (efficiency), combustion knock characteristics and exhaust emissions of an automotive Diesel engine. The engine chosen for this study was a 1978 Oldsmobile, 5.7ℓ, V-8 swirl-chamber Diesel engine. Except for the addition of the alcohol fumigation system, the engine was tested in the 'as received' condition, no attempt was made to alter or optimize the Diesel fuel oil injection system timing.

The conclusions which may be drawn from this study are as follows:

1) Alcohol fumigation increases thermal efficiency at all speeds for higher loads. Methanol fumigation showed an increase in thermal efficiency at 3/4 and full rack settings. Ethanol fumigation increased thermal efficiency at 1/2, 3/4 and full rack settings. However, since at these conditions engine operation becomes limited due to severe knock or roughness for alcohol substitution amounts in the 15 to 30% range, these efficiency gains are of small consequence in terms of stretching petroleum supplies.

2) Alcohol fumigation showed slight increases and decreases in smoke opacity but overall, remained almost constant for all conditions. Therefore it is concluded that exhaust smoke from an indirect injection (IDI) Diesel engine is little effected by alcohol fumigation.

3) For all conditions tested ethanol fumigation ultimately reduces brake specific NO_x to below its baseline value. It is felt that the production of the relatively large volumes of NO_2 as compared to NO when fumigating with ethanol at the lower rack conditions influences the shape of the brake specific NO_x plots.

4) Alcohol fumigation decreased oxides of nitrogen emissions on a volume basis for all conditions tested.

5) Carbon monoxide and unburned hydrocarbon emissions increased for all conditions.

6) Ethanol fumigation, while reducing the mass of exhaust particulate, seems to enhance the biological activity of the particulate. This enhancement does not appear to be as great as that found for methanol fumigation at similar operating conditions.

5.6 Suggestions for Future Work

This study was the first phase of the program to evaluate the utilization of alternative fuels in light duty automotive Diesel engines. No attempt was made to heat the alcohol air charge, and to optimize the injection timing of injection pump which was used in the 'as received' condition. Some reasons for the increase in NO_2 formation, CO and HC emissions may be revealed by using gas chromatography. Use of a microprocessor will allow a more thorough investigation on ignition delay and injection timing. Also further investigation in order to better understand particulate matter and its soluble organic extract are necessary.

APPENDIX A

THE SALMONELLA/AMES TEST (47)

"The Ames test involves several (usually 4) specially constructed strains of the bacterium Salmonella typhimurium (1, 2, 3). The tester strains all require an exogeneous supply of the amino acid histidine for growth. These strains contain unique types of DNA damage at the sites of mutation in the gene(s) which code for the enzymes necessary for the production of histidine. Because of these mutations the strains are auxotrophic (they require exogenous supplies). In strains TA1535 and TA100 there are base pair substitutions (the proper base in the DNA has been replaced by one of the three other bases). Strains TA1537, TA98 and TA1538 contain frame shift mutations (extra bases have been added or bases have been subtracted from the DNA strand).

Different doses of the compound to be tested are combined directly on a Petri dish along with a bacterial tester strain. A trace of histidine, which is not enough to permit colonies to form but which will allow sufficient growth for expression of mutations is added. About 10^8 bacteria are tested on a single Petri plate. The number of bacteria reverted back to an ability to grow without added histidine are measured by counting the revertant colonies on the plate after two days incubation at 37°C. Quantitative dose response curves are obtained which generally have linear regions.

Thus, if a compound causes changes in primary structure of the DNA it will cause one more of the test strains to revert so that they no longer require exogenous histidine for growth. The potency of compounds are compared by determining how many revertants per microgram

of sample are generated in the linear portion of the dose-response curve. The test is based on the high correlation which exists between an agent's ability to cause mutations in bacteria and cancer in animals. The Ames test is extremely sensitive; usually micrograms, and in some cases even nanograms of mutagen can be detected. It is important to note that some mutagens may not be carcinogenic. That is, there are agents which cause mutations in bacteria while they apparently do not cause cancer in animals. In spite of this, the Ames test has been the most successful widely used short term test."

APPENDIX B

Reduced Experimental Data

Key to Data

%ALCH	percent of total fuel energy supplied by fumigated alcohol
$(K_s)_f$	frequency of severe knock occurrence
THEFF	engine thermal efficiency (%)
BHP_c	brake horsepower corrected to standard conditions, T=540°R, P=29.38 in. Hg
BMEP	brake mean effective pressure (psi)
BSEC	brake specific energy consumption (Btu/BHP_c-hr)
AF	air-fuel ratio
PHI	equivalence ratio
TEX	exhaust temperature
SO	smoke opacity (%)
NOXB	brake specific emission of oxides of nitrogen (gm/kW-hr)
NOX	oxides of nitrogen emissions (PPM)
NO	nitric oxides emissions (PPM)
CO	carbon monoxide emissions (PPM)
HC	unburned hydrocarbon emissions (PPM)
O_2	oxygen emissions (% by volume)
CO_2	carbon dioxide emissions (% by volume)

Table B.1

Reduced Experimental Data

Operating Condition: 1/4 Rack, 1500 RPM

Fuel: Ethanol

%ALCH	0	10	20	30	40	50
$(K_s)_f$	0.038	0.052	0.147	0.187	0.147	0.059
THEFF	19.25	18.51	17.65	15.09	12.81	9.51
BHP_c	12.77	12.30	11.70	10.01	8.54	6.32
BMEP	19.27	18.57	17.65	15.10	12.88	9.54
BSEC	13212	13747.5	14424	16860.5	19857.5	26756.5
AF	61.02	57.66	53.51	50.39	48.11	45.82
PHI	0.2445	0.2428	0.2461	0.2469	0.2454	0.2444
TEX	356.5	355.5	351	336	323.5	303
SO	2.05	2.2	2.2	2.05	1.85	1.6
NOXB	3.3724	3.7508	3.4097	3.0147	2.9750	2.9121
NOX	116.25	89.9	77.7	58.5	48.7	35.10
NO	97.25	7	2.76	1	1	0.5
CO	0.0157	0.0594	0.1089	0.155	0.1952	0.2318
HC	45	404	674	976	1445	1865
O_2	16.75	16.79	17	17.35	17.5	17.54
CO_2	3.349	3.349	3.226	2.982	2.741	2.502

Table B.2

Reduced Experimental Data

Operating Condition: 1/2 Rack, 1500 RPM

Fuel: Ethanol

%ALCH	0	10	20	30	40
$(K_s)_f$	0.238	0.150	0.089	0.129	0.386
THEFF	31.03	32.27	33.54	34.75	35.08
BHP_c	39.23	40.96	42.67	44.28	44.6
BMEP	59.18	61.80	64.38	66.81	67.28
BSEC	8197.5	7883.5	7585.0	7321.5	7251
AF	30.66	28.70	27.63	26.75	25.71
PHI	0.4865	0.4876	0.4769	0.4655	0.4588
TEX	667	670	673.5	665	644
SO	2.75	2.8	2.55	2.6	2.4
NOXB	1.7737	1.7428	1.6135	1.4174	1.2777
NOX	204.2	171.2	154.2	133	117
NO	187.6	83.8	59.4	41.6	31
CO	0.0185	0.1092	0.2177	0.3338	0.4038
HC	60	232	357	500	648.33
O_2	11.8	12	11.65	11.69	11.89
CO_2	7.427	7.348	7.348	7.287	7.163

Table B.3

Reduced Experimental Data

Operating Condition: 3/4 Rack, 1500 RPM

Fuel: Ethanol

%ALCH	0	5	10	15	20
$(K_s)_f$	0.100	0.390	0.618	0.953	1.961
THEFF	29.74	30.56	31.46	31.52	32.06
BHP_c	52.40	54.44	55.83	55.98	56.96
BMEP	79.05	82.12	84.23	84.46	85.94
BSEC	8553.0	8324.5	8086.0	8070.5	7937.0
AF	22.14	20.96	20.31	19.68	19.18
PHI	0.6740	0.6893	0.6890	0.6898	0.6875
TEX	826	855.5	865	850	845
SO	6.4	7.5	8	7.15	6.9
NOXB	1.5288	1.3368	1.1977	1.1087	0.999
NOX	239	221.8	203.4	187	169.4
NO	234.33	218.2	200	183	161.6
CO	0.0273	0.0647	0.0931	0.1223	0.1520
HC	38.17	89	139	175	243
O_2	8.39	7.8	7.5	7.65	7.75
CO_2	9.684	10.11	10.233	10.145	10.162

Table B.4

Reduced Experimental Data

Operating Condition: Full Rack, 1500

Fuel: Ethanol

%ALCH	0	5	10	15	20	25
$(K_s)_f$	0.075	0.066	0.137	0.380	1.265	1.724
THEFF	27.08	27.85	28.29	28.87	28.92	28.97
BHP_c	57.45	59.28	59.95	61.29	61.28	61.64
BMEP	86.66	89.43	90.45	92.47	92.45	92.99
BSEC	9396.0	9137.0	8993.0	8810.5	8797.0	8781.5
AF	18.32	17.93	17.37	16.75	18.29	15.86
PHI	0.8144	0.8057	0.8054	0.8104	0.7200	0.8075
TEX	941	975	965	977.5	934	915
SO	11.5	11	10.5	10.4	11.5	12.6
NOXB	1.4695	1.3422	1.208	1.0886	1.1195	1.0047
NOX	250	235	215	198	185	181
NO	245	230	210	194	180	176
CO	0.0409	0.0663	0.0895	0.1068	0.1318	0.1417
HC	41.75	91.8	138	186	226.25	270
O_2	6.31	6	6.25	5.8	6.19	6.34
CO_2	11.155	11.303	11.155	11.34	11.155	10.934

Table B.5

Reduced Experimental Data

Operating Condition: 1/4 Rack, 1720 RPM

Fuel: Ethanol

%ALCH	0	10	20	30	40	50
$(K_B)_f$	0.043	0.044	0.151	0.086	0.071	0.009
THEFF	19.22	18.67	17.27	15.14	12.85	8.24
BHP_c	14.54	14.03	13.06	11.45	9.68	6.23
BMEP	19.13	18.46	17.18	15.06	12.73	8.19
BSEC	13235.0	13626.0	14726.5	16807.5	19797.0	30879.0
AF	60.57	56.80	53.99	50.65	48.20	45.91
PHI	0.2464	0.2464	0.2442	0.2458	0.2447	0.2443
TEX	384	382.5	370.5	357	346	318.5
SO	2.2	2.15	2.2	2	1.9	1.75
NOXB	2.9198	3.1965	2.7261	2.4301	2.2506	2.3559
NOX	103.50	80.42	61.25	47.83	37.17	24.92
NO	91	11.75	4.67	2.83	1.58	1.25
CO	0.0203	0.0639	0.1186	0.1555	0.2048	0.2281
HC	62.5	399.17	930.83	1138.33	1508.33	1750
O_2	16.64	16.79	17	17.25	17.50	17.79
CO_2	3.598	3.432	3.247	3.104	2.861	2.502

Table B.6

Reduced Experimental Data

Operating Condition: 1/2 Rack, 1720 RPM

Fuel: Ethanol

%ALCH	0	10	20	30	40
$(K_s)_f$	0.298	0.287	0.176	0.329	0.378
THEFF	29.23	30.37	30.85	31.49	31.38
BHP_c	39.65	41.07	42.00	42.90	42.34
BMEP	52.16	54.04	55.26	56.44	55.70
BSEC	8702	8377.5	8246.5	8078.0	8107.0
AF	33.12	31.50	29.93	28.54	27.55
PHI	0.4505	0.4442	0.4406	0.4364	0.4277
TEX	629.5	633.5	631	620	595
SO	2.5	2.15	2	1.7	1.5
NOXB	2.2554	2.2810	2.0255	1.7771	1.5935
NOX	223	184.40	157.60	136.20	117.40
NO	198.50	71.80	43.60	28.80	20.60
CO	0.0185	0.1234	0.2351	0.3601	0.4624
HC	45	284	451	618	884
O_2	12.8	13.14	13.10	12.94	13.25
CO_2	6.542	6.559	6.486	6.414	6.013

Table B.7

Reduced Experimental Data

Operating Condition: 3/4 Rack, 1720 RPM

Fuel: Ethanol

%ALCH	0	5	10	15	20
$(K_s)_f$	0.083	-	0.942	1.281	2.180
THEFF	28.92	-	30.55	31.05	31.43
BHP_c	62.06	-	65.85	67.40	67.80
BMEP	81.65	-	86.63	88.67	89.20
BSEC	8797.0	-	8327.5	8192.5	8095.0
AF	20.30	-	18.90	18.28	17.96
PHI	0.7354	-	0.7407	0.7430	0.7337
TEX	925	-	950	945	925
SO	7	-	7.25	7.5	8.38
NOXB	1.5323	-	1.2997	1.1714	1.1143
NOX	264	-	231	211	199
NO	259	-	228	206	193
CO	0.0296	-	0.0834	0.1071	0.1335
HC	318	-	346	370	377
O_2	7.8	-	7.65	7.3	6.94
CO_2	10.532	-	9.59	10.752	10.577

Table B.8

Reduced Experimental Data

Operating Condition: Full Rack, 1720 RPM

Fuel: Ethanol

%ALCH	0	5	10	15
$(K_s)_f$	0.278	0.363	0.858	1.722
THEFF	27.92	28.17	28.66	29.53
BHP_c	68.21	69.81	70.16	72.1
BMEP	89.74	91.85	92.30	94.86
BSEC	9113.0	9032.5	8876.5	8615
AF	18.30	17.58	17.15	16.76
PHI	0.8154	0.8216	0.8156	0.8097
TEX	1033.5	1060	1040	1033.5
SO	10	11.25	11.5	12
NOXB	1.4638	1.3626	1.2417	1.114
NOX	258.33	246.33	225	209.67
NO	253	241	220	206.5
CO	0.0392	0.0659	0.0841	0.0996
HC	50.83	81.67	132.5	180
O_2	5.6	5.34	5.4	5.4
CO_2	11.637	11.812	11.685	11.654

Table B.9

Reduced Experimental Data

Operating Condition: 1/4 Rack, 2000 RPM

Fuel: Ethanol

%ALCH	0	10	20	30	35
$(K_B)_f$	0.005	0.000	0.000	0.000	0.000
THEFF	16.19	14.36	11.55	9.12	6.19
BHP_c	12.47	11.10	8.93	7.06	4.77
BMEP	14.11	12.57	10.11	7.99	5.39
BSEC	15710	17721.0	22010.0	27878.5	41050.0
AF	68.29	63.88	60.48	57.18	55.94
PHI	0.2185	021.91	0.2178	0.2178	0.2164
TEX	395	388.5	371.5	356.5	340.5
SO	2	1.95	1.95	1.8	1.7
NOXB	2.7441	2.9247	2.8877	2.9234	3.4562
NOX	71.80	54.20	41	32	25.4
NO	61.20	18.40	9.40	5.4	4
CO	0.0288	0.0675	0.1096	0.1470	0.1649
HC	91	296	501	734	884
O_2	16.85	17.04	17.35	17.5	17.64
CO_2	3.424	3.212	2.982	2.741	2.561

Table B.10

Reduced Experimental Data

Operating Condition: 1/2 Rack, 2000 RPM

Fuel: Ethanol

%ALCH	0	10	20	30	40
$(K_s)_f$	0.628	0.710	1.006	1.058	1.154
THEFF	27.31	28.18	28.64	28.99	28.17
BHP_c	40.58	41.66	42.17	43.02	41.69
BMEP	45.92	47.14	47.72	48.68	47.17
BSEC	9316.0	9029.0	8881.5	8775	9033.5
AF	35.17	33.43	31.86	30.22	28.48
PHI	0.4242	0.4186	0.4133	0.4118	0.4140
TEX	639.50	639.50	630	628.5	611
SO	1.6	1.45	1.40	1.25	1.45
NOXB	2.3366	2.4348	2.0903	1.8155	1.6495
NOX	205.67	170.17	139.40	120.2	104.8
NO	188	61.17	32.60	23.6	16.6
CO	0.0185	0.1236	0.2357	0.3517	0.4549
HC	50	330	546	750	1010
O_2	13.39	13.75	13.75	13.55	13.75
CO_2	6.044	6.187	5.999	5.929	5.538

Table B.11

Reduced Experimental Data

Operating Condition: 3/4 Rack, 2000 RPM

Fuel: Ethanol

%ALCH	0	5	10	15	20
$(K_s)_f$	0.508	0.575	0.692	1.412	1.849
THEFF	29.22	29.50	30.54	30.58	32.00
BHP_c	65.70	66.07	68.66	68.57	71.63
BMEP	74.34	74.76	77.69	77.59	81.04
BSEC	8706.50	8625.5	8331	8320.5	7949.5
AF	21.95	21.87	21.02	20.81	20.26
PHI	0.680	0.661	0.666	0.652	0.650
TEX	858.5	851	840	845	850
SO	4	3.5	3.5	3.5	3.75
NOXB	1.739	1.6585	1.6164	1.5085	1.3878
NOX	274.17	251.67	240	220	200.71
NO	266.67	242.5	213.33	185.83	156.43
CO	0.0233	0.0604	0.0992	0.1370	0.1785
HC	129.17	182.5	245	307.5	379.29
O_2	8.35	8.35	8.39	-	-
CO_2	9.428	9.414	9.330	9.259	9.155

Table B.12

Reduced Experimental Data

Operating Condition: Full Rack, 2000 RPM

Fuel: Ethanol

%ALCH	0	5	10	15	20
$(K_s)_f$	0.474	0.602	0.648	1.359	2.027
THEFF	28.96	29.41	30.23	30.58	31.08
BHP_c	77.51	79.07	81.99	81.59	83.29
BMEP	87.71	89.46	92.77	92.32	94.24
BSEC	8783.0	8651.0	8416.0	8320.0	8184.0
AF	18.56	18	18	17.51	17.22
PHI	0.8037	0.8029	0.7779	0.7743	0.7640
TEX	1090	1087.5	1140	1061	1040
SO	7.5	7.5	8.25	9.25	10.75
NOXB	1.4902	1.3832	1.2991	1.1674	1.1921
NOX	273.8	258	238	215	219.83
NO	270	254	233	210	216.33
CO	0.0377	0.0573	0.0748	0.095	0.1043
HC	69	85	127	166.67	196.67
O_2	5.55	5.94	5.25	5.65	5.44
CO_2	11.793	11.679	11.888	11.513	11.654

Table B.13

Reduced Experimental Data

Operating Condition: 1/4 Rack, 1500 RPM

Fuel: Methanol

%ALCH	0	5	10	15	20	25
$(K_B)_f$	0.043	0.088	0.15	0.182	0.278	0.119
THEFF	22.04	20.90	17.90	14.93	13.41	10.47
BHP_c	14.84	13.38	11.55	9.90	8.92	6.82
BMEP	22.38	20.18	17.43	14.93	13.45	10.29
BSEC	11543.7	12171.6	14214.1	17034.8	18973.4	24295.8
AF	58.96	59.17	56.59	54.02	51.31	50.36
PHI	0.2542	0.2477	0.2412	0.2434	0.2473	0.2437
TEX	390	380	365	323	315	297
SO	2	2.1	2	2.7	2.8	2.9
NOXB	-	-	-	-	-	-
NOX	-	-	-	-	-	-
NO	52.5	7	4	2	1	1
CO	0.0734	0.0924	0.1116	0.1116	0.1309	0.1464
HC	260	1040	770	480	740	690
O_2	16.75	17.25	17	17.5	17.5	18.25
CO_2	3.45	3.26	3.14	3.08	2.91	2.67

Table B.14

Reduced Experimental Data

Operating Condition: 1/2 Rack, 1500 RPM

Fuel: Methanol

%ALCH	0	5	10	15	20	25
$(K_B)_f$	0.07	0.11	0.128	0.157	0.317	0.758
THEFF	32.05	30.75	29.76	27.77	27.37	26.85
BHP_c	40.75	37.90	37.67	34.10	34.59	33.91
BMEP	61.48	57.18	56.83	51.44	52.18	51.15
BSEC	7939.6	8275.3	8550.7	9161.2	9295.4	9477.8
AF	31.70	31.34	29.04	28.88	26.28	25.07
PHI	0.4729	0.4575	0.4726	0.4556	0.4822	0.4872
TEX	688	635	686	580	571	565
SO	2.9	2.4	2.5	2	1.9	1.7
NOXB	-	-	-	-	-	-
NOX	-	-	-	-	-	-
NO	118	49	36.5	23	21	18.5
CO	0.0924	0.1348	0.1697	0.2281	0.2707	0.3093
HC	97	115	134	195	220	250
O_2	10.50	11	11.5	12.25	12	11.75
CO_2	7.43	6.99	6.99	6.27	6.20	6.27

Table B.15

Reduced Experimental Data

Operating Condition: 3/4 Rack, 1500 RPM

Fuel: Methanol

%ALCH	0	5	10	15	20	25
$(K_s)_f$	0.082	0.07	0.127	0.11	0.183	0.37
THEFF	29.38	30.94	31.90	31.68	31.43	31.81
BHP_c	52.03	54.62	57.40	54.57	55.15	55.45
BMEP	78.50	82.39	86.60	82.33	83.19	83.66
BSEC	8661.1	8223.3	7975.1	8031.2	8096.4	7998.2
AF	23.21	21.73	19.52	19.93	18.48	17.94
PHI	0.6457	0.6599	0.6936	0.6633	0.6870	0.6914
TEX	905	940	963	870	865	874
SO	6.2	6.6	6.7	5.5	5.5	5
NOXB	-	-	-	-	-	-
NOX	-	-	-	-	-	-
NO	119.0	113	120	110	81	74.5
CO	0.0715	0.1116	0.1542	0.2086	0.2358	0.2475
HC	113	98	72	85	101	104
O_2	8.25	7.25	6.25	8	8.25	8.25
CO_2	9.90	11.07	11.64	10.08	9.65	9.48

Table B.16

Reduced Experimental Data

Operating Condition: Full Rack, 1500 RPM

Fuel: Methanol

%ALCH	0	5	10	15	20	25
$(K_s)_f$	0.014	0.025	0.056	0.13	0.138	0.412
THEFF	26.44	27.36	27.79	28.17	28.27	29.41
BHP_c	56.37	56.92	57.48	56.92	58.27	59.24
BMEP	85.05	85.87	86.71	85.88	87.91	89.37
BSEC	9620.9	9299.1	9155.3	9033.0	9000.9	8650.3
AF	18.86	17.93	17.19	16.82	15.70	15.21
PHI	0.7947	0.7993	0.7989	0.7832	0.8101	0.8043
TEX	972	965	945	910	905	895
SO	7.5	7.8	7.1	5.4	6.9	7.6
NOXB	-	-	-	-	-	-
NOX	-	-	-	-	-	-
NO	77	79.5	58.5	61	49.5	47.5
CO	0.0734	0.0810	0.0734	0.1116	0.1542	0.1892
HC	280	205	135	150	140	165
O_2	7.5	8.25	8.5	8.25	8.75	9
CO_2	10.35	11.64	10.62	10.26	10.35	9.9

Table B.17

Reduced Experimental Data

Operating Condition: 1/4 Rack, 1720 RPM

Fuel: Methanol

%ALCH	0	5	10	15	20
$(K_s)_f$	0.055	0.04	0.03	0.017	0.025
THEFF	17.32	16.08	15.10	12.88	12.82
BHP_c	13.30	12.50	11.79	9.87	10.21
BMEP	17.50	16.45	15.51	12.99	13.43
BSEC	14694.9	15826.1	16851.0	19750.6	19850.5
AF	61.26	56.24	54.40	51.81	46.94
PHI	0.2446	0.2593	0.2607	0.2545	0.2722
TEX	338	332	330	313	315
SO	2	2.2	2.2	2.2	2.3
NOXB	-	-	-	-	-
NOX	-	-	-	-	-
NO	56	-	6	-	4
CO	0.0453	-	0.0179	-	0.209
HC	219	-	238	-	400
O_2	17.5	-	18	-	19
CO_2	3.63	-	3.08	-	2.56

Table B.18

Reduced Experimental Data

Operating Condition: 1/2 Rack, 1720 RPM

Fuel: Methanol

%ALCH	0	5	10	15	20	25	30	35
$(K_B)_f$	0.045	0.128	0.075	0.334	0.271	0.325	0.423	0.351
THEFF	26.10	25.88	24.56	24.59	25.09	24.89	23.91	23.37
BHP_c	35.35	34.28	33.75	33.92	34.44	34.17	32.92	32.17
BMEP	46.50	45.10	44.40	44.62	45.31	44.95	43.32	42.32
BSEC	9748.1	9832.8	10358.7	10346.3	10139.6	10223.5	10641.0	10889.4
AF	35.24	33.85	31.09	28.93	27.56	25.87	25.54	24.61
PHI	0.4254	0.4226	0.4426	0.4568	0.4604	0.4743	0.4649	0.4683
TEX	576	566	605	515	515	505	504	480
SO	1.6	1.6	1.2	1.3	1.8	1.5	1.4	1.2
NOXB	-	-	-	-	-	-	-	-
NOX	-	-	-	-	-	-	-	-
NO	53	28.5	22.5	19	17	15	16.5	12.50
CO	0.0546	0.1116	0.1386	0.1969	0.2281	0.2669	0.3439	0.3820
HC	100	228	250	320	340	345	360	420
O_2	15.4	15.2	14.5	14	14	14	14.25	14.5
CO_2	5.61	5.78	5.64	5.29	5.33	5.15	4.81	4.94

Table B.19

Reduced Experimental Data

Operating Condition: 3/4 Rack, 1720 RPM

Fuel: Methanol

%ALCH	0	5	10	15	20
$(K_B)_f$	0.087	0.125	0.492	0.186	2.384
THEFF	27.83	28.54	28.60	29.41	31.32
BHP_c	59.66	61.46	62.34	62.64	64.32
BMEP	78.49	80.86	82.01	82.41	84.62
BSEC	9141.6	8914.4	8896.6	8651.3	8124.2
AF	21.46	20.08	19.01	18.24	18.05
PHI	0.6985	0.7144	0.7307.	0.7222	0.7129
TEX	860	875	869	848	855
SO	10.2	10	10.15	11.2	12.2
NOXB	-	-	-	-	-
NOX	-	-	-	-	-
NO	85.5	-	78	-	70
CO	0.0658	-	0.1503	-	0.2086
HC	250	-	328	-	300
O_2	11	-	10.75	-	10
CO_2	9.14	-	9.48	-	10.89

Table B.20

Reduced Experimental Data

Operating Condition: Full Rack, 1720 RPM

Fuel: Methanol

%ALCH	0	5	10	15
$(K_s)_f$	0.295	0.546	0.618	1.982
THEFF	26.84	27.25	27.71	28.36
BHP_c	65.91	67.42	67.99	69.33
BMEP	86.72	88.70	89.45	91.22
BSEC	9482.7	9336.1	9181.9	8972.1
AF	18.17	16.89	16.37	15.49
PHI	0.8252	0.8504	0.8411	0.8543
TEX	990	992	975	955
SO	13.5	13.5	14.2	14.5
NOXB	-	-	-	-
NOX	-	-	-	-
NO	80	-	78	76
CO	0.0734	-	0.1425	0.1580
HC	425	-	386	376
O_2	9.38	-	9	9
CO_2	10.47	-	10.52	-

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15. Supplementary Notes

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16. Abstract

A light-duty automotive Diesel engine was fumigated with methanol and ethanol in amounts up to 35% and 50% of the total fuel energy respectively. The main purpose of this study was to determine the effect of alcohol (methanol and ethanol) fumigation on engine performance at various operating conditions. Engine fuel efficiency, emissions, smoke, and the occurrence of severe knock were the parameters used to evaluate performance. Raw exhaust particulate and its soluble organic extract were screened for biological activity using the Ames Salmonella typhimurium assay. Results are given for a test matrix made up of twelve steady-state operating conditions. For all conditions except the 1/4 rack (light load) condition, modest thermal efficiency gains were noted upon ethanol fumigation. Methanol showed the same increase at 3/4 and full rack (high load) conditions. However, engine roughness or the occurrence of severe knock limited the maximum amount of alcohol that could be fumigated. Brake specific NO_x concentrations were found to decrease for all ethanol conditions tested. Oxides of nitrogen emissions, on a volume basis, decreased for all alcohol conditions tested. Based on the limited particulate data analyzed, it appears as though ethanol fumigation, like methanol fumigation, while lowering the mass of particulate emitted, does enhance the biological activity of that particulate.

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