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Performance and Emissions Characteristics of Aqueous Alcohol Fumes in a DI Diesel Engine

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NOTICE

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APPENDIX

NOMENCLATURE

A,B,C	Constants for Antoine vapor pressure equation
ATDC	After top dead center
A/F	Air-fuel ratio
B(a)P	Benzo(a)pyrene
BBDC	Before bottom dead center
bhp	Brake horsepower test
BHP_c	Brake horsepower corrected
BMEP	Brake mean effective pressure
BSFC	Brake specific fuel consumption
BTDC	Before top dead center
C_p	Constant pressure specific heat
CI	Compression ignition
DI	Direct injected
DNA	Deoxyribonucleic acid
dp/dt	Time rate of pressure change
E	Fraction of fuel energy supplied as ethanol
fhp	Friction horsepower
ihp	Indicated horsepower
k	Reaction rate constant
M	Fraction of fuel energy supplied as methanol
P	Pressure
PAH	Polycyclic aromatic hydrocarbon
ppm	Parts per million
R	Pressure regulator
R_o	Ideal gas constant

RPM	Revolutions per minute
SI	Spark ignition
SOF	Soluble organic fraction
T	Temperature
TDC	Top dead center
η_{th}	Thermal efficiency

Chapter 1

INTRODUCTION

1.1 Background and General Statement of Problem

During the past decade, the importance of alternative energy source development has gained increased recognition. Faced with both rapidly depleting domestic oil reserves as well as politically and economically unstable foreign sources of petroleum crude, the United States must unfold alternative domestic energy sources.

In the past, liquid hydrocarbons have provided an abundant, economical source of easily refined and combusted mobile fuels. However, as various factors deleteriously affect the availability of these petroleum products, significant reductions in their use through energy conservation or fuel substitution is demanded.

Historically, the automotive and transportation sector has formed a large segment of the total U.S. petroleum diet. Combined, transportation demands comprise over 53% of all United States' petroleum requirements and nearly 25% of its total energy needs (1)*. In response to this concentrated demand on petroleum-based fuels, both government and private programs have directed major efforts toward developing alternative mobile fuels for specific application to transportation and automotive areas. In addition, equal importance has been placed upon the goal of more efficient use of presently available energy sources by transportation.

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*Numbers in parentheses refer to entries in Reference List on page 102.

In the quest to cultivate viable near-term alternative fuels, alcohols have received significant consideration for motor fuel applications. There are three common types of alcohol: ethyl, methyl, and isopropyl. The former two alcohols warrant evaluation as gasoline and fuel oil substitutes since they are generally produced from non-petroleum distillates (2).

Arguments in favor of and against development of alcohols as petroleum fuel alternatives are numerous. Proponents emphasize the potential "renewability" of alcohols, a result of their ability to be produced from biomass, a seemingly perennial supply of fuel feedstock. Agriculture leaders point out the beneficial effects of alcohol development: reduction of United States foreign oil dependency and the creation of a domestic market for surplus agricultural commodities. Thus, a sound economic base is created for an often ailing and unstable farm community (3). Critics contend that present gasohol subsidies have already introduced several negative factors; energy imbalance in alcohol production and increased pollution during alcohol refining are cited as disbenefits resulting from gasohol production (4).

Alcohols may not necessarily represent the most resource-efficient or cost-effective use of available fuel feedstocks. However, it is generally recognized that these fuels are among only a very few alternative energy sources which resemble currently used petroleum fuels and permit consumption in existing power plants with minimal modifications (5). As a result, alcohols will probably continue to receive consideration and development as petroleum-based fuel substitutes.

Methanol, the most simple of alcohol fuels, has generated significant interest among alcohol proponents. Its production involves a catalytic reaction of carbon monoxide and hydrogen with the fuel feedstocks including natural gas, wood wastes, and coal. In an effort to make use of large, undeveloped resources of midwestern lignite, studies have shown that this economical and abundant feedstock can be used to produce equivalent methanol yields comprising nearly 45% of the total lignite available energy. Current (1979) costs for methanol production from this source are estimated to be approximately twice that of gasoline (3).

Ethanol is commonly produced by one of two processes: fermentation of grains and other sugar or starch feedstocks or by the synthesis of ethylene (6). Since ethylene is in fact a petroleum distillate, advocates of ethanol development favor fermentation of renewable feedstocks for its production. In the fermentation process, a decomposition is brought about by enzymes on sugar solutions or saccharafied mashes of starch containing materials such as grains and grasses (7). Ethanol production facilities include distillation columns designed to separate ethyl alcohol from water used and formed in the fermentation process. Since ethanol is completely soluble in water, the final proof, generally between 100 and 190, is a direct function of the distillation efficiency. Due to the formation of azeotropic bonds with water, an additional processing step beyond distillation is necessary to obtain 200 proof ethanol (7). This final processing step usually involves the addition of bond-breaking benzene, followed by a final distillation; this adds significantly to production costs.

Methanol, due to the more complex nature of its production relative to ethanol, is commonly manufactured in large-scale industrial facilities. The simple fermentation process involved in ethanol production on the other hand, lends itself more readily to small-scale applications and in particular to individual farm manufacture. However, the natural presence of water following initial distillation is of concern especially as applied to these small, farm-operated ethanol stills. Since higher quality ethanol (i.e. higher proof) entails higher production costs, justification of further distillation must be based on several factors: engine performance and emissions when using anhydrous fuel, and engine wear rates resulting from water effects.

The use of alcohol fuels in internal combustion engines is not new. These fuels have been used intermittently in spark ignition engines since their invention. The lower alcohols, methanol and ethanol, are known to be excellent fuels for spark ignition (SI) engines due to their high octane ratings, lean flammability limits, increased thermal efficiency and low exhaust emissions (8, 9). Numerous studies have examined the effects of alcohol both as a fuel supplement and as the primary fuel. Significant SI performance improvement and emission reduction have been shown (10-12). In the past, little attention has been given to the utilization of alcohol fuels in compression ignition (CI) engines. This is due largely to the low cetane number (extrapolated to be approximately 0-5 for alcohols (8)) and the resultant combustion-related problems associated with alcohol usage in these power plants.

The development of alternative fuels, however, must be coupled with more efficient use of all fuels. The compression ignition engine has surfaced as a potential ingredient of this more efficient energy utilization, particularly in its broad application as a substitute for the SI engine in light-duty service. Factors including higher compression ratio, reduction in pumping losses, and lean part load combustion are primarily responsible for the Diesel engine's improved thermal efficiency compared to its spark-ignited counterpart.

Tests conducted by Southwest Research Institute (13) have compared the Oldsmobile Cutlass Diesel and a production Volkswagen Rabbit Diesel with similar gasoline-fueled models. "The results corroborate recognized tradeoffs of Dieselization: some 30-60% better fuel economy, moderate losses in acceleration, a bit more noise and levels of regulated emissions comparable to those of converter equipped gasoline cars (14)."

However, previously unregulated Diesel emissions are currently creating concern. Particulates, reactive hydrocarbons, smoke, and odor created during fuel-rich combustion (characteristic of diffusion controlled reactions) may pose serious problems with an expanding market of light-duty Diesel-powered vehicles. In particular, the formation and biological activity of Diesel particulates have recently generated substantial concern.

Despite the several emission problems linked with Diesel combustion, the CI engine is likely to play a major role in future automotive applications. Diesel engines already assume much greater importance than SI engines in the economies of developing nations; particularly concentrated use exists in heavy transportation and agriculture (15).

It is apparent then that alternative fuels such as alcohols must be applicable to Diesel combustion if they are to contribute significantly as long-term substitutes for petroleum-based fuels.

The poor autoignition properties of alcohols are responsible for two types of combustion complications in Diesel engines: severe knock due to rapid burning of vaporized alcohol, and combustion quenching caused by high latent heats of vaporization and subsequent charge cooling. Rough or knocking combustion in Diesel engines is a complex phenomenon. It is generally associated with rapid rates of pressure rise resulting from increased ignition delay and rapid combustion of gaseous fuel. Attempts to improve the combustion of alcohols in unmodified Diesel engines have included the use of ignition accelerators such as amyl nitrate and cyclohexyl nitrate (16). Although having a beneficial effect on knock reduction, these additives are prohibitively expensive and tend to produce increased levels of nitric oxide emissions (17, 18).

Engine modifications to improve alcohol combustion have included increased compression ratios, spark ignition assist, and imposed surface ignition (16, 19, 20). Modifications of this type frequently improve alcohol combustion, but are costly. Documented usage of neat alcohol has been achieved by utilizing these fuel and engine modifications. The poor compression-induced ignition of alcohols, however, has directed the major investigation of alcohol fuel utilization in Diesel engines toward variations of dual fueling. In such cases, alcohols are burned in conjunction with an amount of Diesel fuel oil which acts as an ignitor for the alcohol fuels. Typically, dual fueling has been conducted by one of three major processes:

1) formation of alcohol-fuel oil blends or emulsions,
2) separate injection of alcohol and fuel oil, and
3) fumigation, carburetion, or injection of alcohol into the intake air change.

To date, extensive work has been conducted evaluating each of these methods of dual fueling. Attempts to form stabilized emulsions or blends of alcohol and fuel oil have been unsuccessful when even trace amounts of water were present in the mixture. Strait et. al. injected stabilized emulsions of Diesel fuel oil and up to 40% absolute ethanol; separation problems were noted if as little as 0.27 percent water was present in the alcohol (21). Methanol presents a more severe separation problem; water tolerance of methanol-fuel oil blends is only about 1/5 that of similar ethanol blends (6, 22).

Surfactants have been added to stabilize alcohol-fuel oil blends. Although infinitely stable mixtures of up to 30% alcohol were produced, these additives added significantly to the cost of the fuel blend (5, 23).

Fuel separation problems have directed some investigators toward the direct use of unstable mixtures. In most investigations, fuels were stored in separate tanks and mixed ahead of the injection pump or nozzle. Holmer (24) experimented with forming mixtures in the injection pump and in the injection nozzle. Engine performance, in general, was comparable to that during fuel oil operation. However, unanticipated nozzle problems resulted from the lower dampening qualities of methanol; injector needle contact points showed increase wear. Mechanical mixing devices create unstable or moderately stable blends which have been used with limited success. Lawson et al. (25) produced an acceptably stable emulsion which was supplied directly to the standard fuel injection system. Mixtures containing up to 30% methanol were injected with noted increases in engine efficiency and reductions in particulate emissions.

Separate direct injection of alcohol eliminates the need to form stabilized alcohol-fuel oil blends. Dual fuel systems, including reservoirs, pumps and injectors, are required for this method of dual fueling.

Fuel of poor combustion properties is injected through the primary nozzle and supplies the bulk of the engine fuel requirements. A good autoignition-property fuel (high cetane number) injected through a secondary nozzle, acts as a pilot which ignites and supports combustion of the primary fuel. Factors controlling the start of combustion in dual injected Diesel engines utilizing alcohol fuels have been identified as: (26)

1) pilot fuel quantity used to liberate the ignition energy,
2) timing and duration of the pilot oil and ethanol injection periods,
3) injection pressures, and
4) compression temperature at the time of injection.

Important influences on the type of combustion are associated closely to the interaction of the pilot and alcohol fuel charges. This interaction is often regulated by the nozzle spray pattern and injector location (27, 28, 29). Tests generally confirmed that combustion started in portions of the fuel oil spray which had not mixed with the alcohol charge. Undue ignition delay was also avoided by minimizing unfavorable interaction of the two sprays.

Optimization of factors most strongly affecting dual injection combustion has produced excellent results. Gaseous pollutants and visible smoke have been reduced while combustion problems related to increased ignition delay have been minimized (27). Despite the noted advantages, negative factors unique to dual injection of alcohols do exist. Cost and complexity are significantly increased by the requirement of two separate fuel injection systems. Related fuel system problems, ranging from insufficient alcohol lubricity for critical pump parts to corrosive and mechanical injector problems, have also been reported (30).

Early investigation of methods to improve Diesel engine performance showed that substantial gains could be made by fuel addition to the intake air (31). This work formed the basis for fumigation, a commonly used method of alcohol introduction in Diesel engines.

Fumigation is a particularly attractive means of alcohol utilization in Diesel engines; successful operation is achieved with minimal engine modifications. Up to 80% of the total fuel requirements have been carburetted into a Diesel engine at the Indian Institute of Technology with generally satisfactory performance over most of the load range (32). In multicylinder testing, Houser et al. (33) fumigated up to 40% methanol into an Oldsmobile Diesel engine and found methanol substitution to be limited either by poor combustion or knock limited operation. (A more thorough discussion of fumigation is presented in Section 2.2.3.)

Results of preliminary studies indicate that although alcohols are not generally suitable for neat alcohol combustion in Diesel engines, performance results generally support at least some percentage of alcohol substitution via dual fueling. However, further investigation of alcohol dual fueling should be focused on two areas:

1) the potential hazards associated with particulate and exhaust emissions produced under alcohol-fueled conditions, and
2) the performance and emissions related effects of low-quality, i.e., lower proof, alcohols.

Initial work by Houser et al. (33) has indicated that particulate mass emissions, although reduced by alcohol substitution, are of enhanced biological activity, thus posing a possible health problem. Low-quality alcohols are likely to be produced by cost-conscious small-farm distillaries. To date, only limited investigation of these low-quality fuels has been conducted.

If alcohols are to achieve significance as a Diesel fuel substitute, further work is justified to provide additional information in these areas. Further evaluation of emissions-related biological hazards as well as performance and emission trends of low-quality alcohol combustion forms the basis of this experimental program.

1.2 Specific Objectives

In light of past studies of performance and exhaust emissions in alcohol-fumigated Diesel engines, the specific objectives of this study are set forth as follows:

1. Establish a baseline matrix of engine operating conditions as defined by rack setting and engine speed. Document engine performance as well as exhaust emissions at these conditions for certified No. 2 Diesel fuel oil operation.

2. Develop and install instrumentation to provide information regarding injection timing, ignition delay, pressure, rate of pressure rise, and knock intensity for baseline and alcohol operation.

3. At each 2400 RPM test condition, fumigate various proofs of ethanol and methanol as limited by engine knock or misfire. Obtain for each operating condition, performance data including thermal efficiency and power output as well as regulated emissions data (CO, HC, NO_x).

4. For various test conditions collect sufficient exhaust particulate to document the effects of alcohol fumigation on the biological activity of these solid phase emissions.

Chapter 2

THEORETICAL CONSIDERATIONS

2.1 Fuel Characteristics

Complete assessment of changes in performance and emissions of an engine utilizing alternative fuels must include an insight into the differences in physical, chemical, and combustion properties of the fuels. Alcohols, although representing the most attractive near-term substitute for petroleum-based fuels, differ greatly with respect to certain fluid and combustion characteristics from their petroleum-based hydrocarbon counterparts. Selected liquid fuel properties comparing anhydrous ethanol and methanol with fuel oil are presented in Tables 2.1 and 2.2. Before attempting to interpret observed changes in engine emissions and performance occurring during alternative fueling, some of the important characteristics of the different fuels must be considered.

2.1.1 Stoichiometry and Charge Density

Diesel combustion unlike spark ignition combustion is a diffusion-controlled, heterogeneous process. As such, localized regions of air and fuel burn in nearly stoichiometrically correct proportions. The Diesel cycle's unthrottled aspiration results in an overall stoichiometry which is quite lean at light load and would not normally result in spontaneous ignition. By proper design of the combustion chamber, combustion can be initiated and maintained to yield a nearly complete reaction.

Table 2.1

Physical and Chemical Liquid Fuel Properties
(Adapted from Houser (33), Obert (35) and Yost (36))

	Ethanol	Methanol	No. 2 Fuel Oil
Physical Properties			
Specific Gravity at 68°F	.795	.796	.846[+]
Liquid Density (lbm/gal)	6.60	6.61	7.05
Boiling Temperature (°F)	172.0	149.0	376-627[+]
Freezing Temperature (°F)	-170.0	-144.0	-
C_p at 60°F (Btu/lbm-°F)	.648	.60	∿.52
Heat of Vaporization (Btu/lbm)	396.	502.	110.[++]
Viscosity at 68°F (cp)	-	.595	.785[+]
Water (H_2O) Solubility	∿∞	∿∞	∿0
Chemical Properties			
Formula	C_2H_5OH	CH_3OH	-
Molecular Weight	46.06	32.04	-
Composition by Weight			
% Carbon	52.2	37.5	86-87
% Hydrogen	13.1	12.6	11.13.5
% Oxygen	34.7	49.9	∿0

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

Table 2.2

Gaseous Fuel Combustion and Ignition Properties
(Adapted from Houser (33), Obert (35), and Yost (36))

	Ethanol	Methanol	Fuel Oil
Combustion Property			
Combustion Equation	$C_2H_5OH+3O_2+11.3N_2$ $\rightarrow 2CO_2+3H_2O+11.3N_2$	$CH_3OH+1.5O_2+5.66N_2$ $\rightarrow CO_2+2H_2O+5.66N_2$	$C_nH_{2n}+1.5nO_2+5.66nN_2$ $\rightarrow nCO_2+nH_2O+5.66nN_2$
Stoichiometric Air/Fuel Ratio	9.0	6.4	∿15.0
Moles Product Per Mole Charge	1.065	1.061	1.062^{++}
Constant Pressure Heating Value at 77°F,-ΔH (Btu/lbm)			
HHV } liquid fuel	12,780	9,770	-
LHV	11,604	8,644	$19,192^{+}$
Btu/ft^3 (Stoich. Mix.)	92.9	89.4	96.9^{++}
Btu/lbm (Stoich. Mix.)	1184.	1069.	$1204.^{++}$
Ignition Properties			
Octane Number			
Research Method	107	106	-
Pump (RON+MON)/2	98	99	-
Cetane Number	0-5	0-5	47.5^{+}
Flash Point (°F)	54-55	52	$158.^{+}$
Autoignition Temperature (°F)	738-964	867-878	∿230

+ Indicates property value is from laboratory analysis of No. 2 Diesel fuel oil.
++ Indicates property value is for Dodecane.

Stoichiometrically correct air-fuel ratios (A/F) for Diesel fuel oil are typically about 15:1, i.e., approximately 15 pounds of air are required to burn one pound of fuel. The charge density of a stoichiometrically correct mixture of fuel oil and air is approximately 97 Btu/ft^3. However, incomplete combustion occurring in fuel-rich zones, a consequence of diffusion combustion processes, generally limits A/F ratios to a minimum of 20:1. This is referred to as smoke-limited power.

The stoichiometrically correct A/F ratios for ethanol and methanol are lower than those for Diesel fuel oil due to bound oxygen in the alcohol molecule supplying a portion of the oxygen required for stoichiometrically correct combustion. Oxygen constitutes 34.7 percent of the molecular weight of ethanol (34), and contributes approximately 14 percent of the stoichiometric oxygen requirement, yielding an overall A/F ratio of 9.0:1. Methanol by comparison, has respective values of 50 percent and 25 percent with a stoichiometric A/F ratio of 6.4:1 (37). In addition, the higher hydrogen to carbon ratio of the alcohols results in lower oxygen requirements since hydrogen requires less air to burn than carbon (Table 2.1 and 2.2).

As noted in Table 2.2, the charge densities of a stoichiometrically correct mixture of either alcohol fuel and air is approximately equivalent to that of fuel oil. This may seem confusing in light of the heats of combustion listed for ethanol and methanol which are approximately 60 percent and 45 percent, respectively, that of fuel oil. However, the proportionately lower oxygen requirement of the alcohols results in similar charge densities for air combustion of these three fuels.

2.1.2 Vapor Pressure, Volatility and Latent Heat of Vaporization

In normal Diesel combustion, liquid fuel oil is injected into hot compressed gases which supply the required energy for fuel vaporization. A multiconstituent fuel such as Diesel fuel oil has a range of boiling temperatures. A single vapor pressure is undefined since the vapor and liquid phase contain different amounts of each constituent. It is apparent then that the vapor pressure of a multiconstituent liquid fuel depends upon the extent of vaporization - upon the vapor volume (35).

The volatility, nevertheless, can be defined as the percent of liquid vaporized at a particular temperature. Typical distillation curves for Diesel fuel may have initial vaporization points beginning around 370°F, with an end point around 640°F. The latent heat of vaporization of Diesel fuel oil is quite low - only 110 Btu/lbm for dodecane, a pure hydrocarbon characteristic of fuel oil. This low heat of vaporization results in minimal cooling of the compressed air-fuel charge preceding combustion; chemical delay, an important factor affecting the intensity and noise of Diesel combustion, is minimized.

Since anhydrous ethanol and methanol are pure substances, their vapor pressure can be determined as a function of temperature by the Antoine equation,

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

where

P = vapor pressure (mm Hg)

T = temperature (°C)

and A, B, C are constants characteristic of pure compounds. Typical values are listed in Table 2.3. At room temperature, these vapor pressures are higher than that of many fuel oil distillates resulting in relatively high volatility.

Table 2.3

Constants for Antoine Equation

Fuel	Temperature Range (°C)	A	B	C
Ethanol	-2 to 100	8.32109	1718.10	237.52
Methanol	-14 to 65	7.89750	1474.08	229.13
	65 to 110	7.97328	1515.14	232.85

Source: R. C. Wilhoit and B. J. Zwolinski (38)

Strong bonding of the hydroxyl group (-OH) in ethanol and methanol is responsible for high latent heats of vaporization compared to other hydrocarbon fuels. Ethanol, higher in the aliphatic alcohol series, has a lower heat of vaporization (396 Btu/lbm) than methanol due to its smaller relative oxygen content and subsequently weakened hydroxyl bond (39).

The vaporization of a stoichiometric amount of ethanol and methanol in air will lower the adiabatic mixture temperature by 200°F and 356°F, respectively, compared to only a 30°F drop for a stoichiometric mixture of cetane in air (40). As pure substances, ethanol and methanol also have unique boiling points which couple with their low vapor pressures at low temperature and high latent heats of vaporization to cause driveability problems in SI engines (41). In Diesel combustion, these characteristics cause charge cooling, leading to excessive ignition delay and related combustion problems.

2.1.3 Engine and Fuel System Compatibility

The natural lubricity of Diesel fuel oil makes it a compatible fuel for most injection pumps and nozzles. In these systems, the fuel oil commonly acts directly as both a lubricating agent and coolant for the fuel system components. Wear rates for both engine and fuel system are generally low for Diesel engines burning fuel oil. Formation of carbonaceous residue during the combustion process, however, results in accelerated engine oil contamination and may lead to nozzle fouling during overfueled operation.

Unfortunately, alcohols lack the good lubricating properties characteristic of most petroleum-based hydrocarbon fuels. As a result, attention has been focused on the possibility of increased wear rates in engines and fuel systems utilizing alcohol fuels. In addition, alcohols are highly polar; therefore, they tend to be more reactive than conventional nonpolar hydrocarbon fuels with certain materials. Corrosion and degradation problems have been noted in engines where methanol-gasoline mixtures have contacted lead, magnesium, aluminum, and certain plastics (42). Most elastomeric materials used in seals and gaskets, such as nitrile rubber, are oil resistant due to highly polar acrylonitrile molecules. However, these materials are generally susceptible to attack by other highly polar groups such as alcohols, which may lead to problems including swelling (43).

Although numerous studies report material compatibility problems associated with alcohols in the fuel and air management systems, less work has been completed regarding the effects of alcohol on basic engine components. Unburned alcohol may have a harmful effect on lubricant additives due to its highly polar nature. Blowby gases produced during alcohol combustion may contain organic acids and aldehydes which may further increase corrosion by mixing with the engine lubricant (44). Tests simulating short-trip winter service have confirmed that ethanol produced a 180 percent increase in iron wear when compared to gasoline (45).

2.2 Combustion Considerations

Normal Diesel fuel oil combustion in a compression ignition engine proceeds as a petroleum-based fuel vaporizes and diffuses into surrounding oxygen. Since fuel oil is a full boiling range fuel, lighter ends vaporize initially and ignite, providing additional energy to vaporize the heavier fuel species. Typically, nonuniformities will exist in the air-fuel mixture, causing carbonaceous residue or particulate to be formed as a result of incomplete combustion and pyrolysis.

The differences in fuel properties noted earlier (most importantly molecular structure, high latent heat of vaporization and high vapor pressure), combined with specific combustion properties typical of the alcohols, produce varied combustion characteristics in comparison to Diesel fuel oils.

2.2.1 Combustion Properties

Ethanol and methanol burn with a nonluminous, sootless flame. Products of combustion normally include carbon monoxide and water. Alcohols generally have a higher flame speed than typical petroleum-based fuels due to the formation of hydrogen during high temperature dissociation of the alcohols. Ethanol undergoes this thermal decomposition at temperatures exceeding 1472°F, producing ethylene, acetaldehyde, water and hydrogen (7). Similarly, methanol dissociates at high temperatures into carbon monoxide and hydrogen.

Alcohols tend to burn somewhat cooler than most petroleum-based hydrocarbons. Factors including high latent heat of vaporization (resulting in intake charge cooling), an increase in the number of moles during combustion, and thermal decomposition of the fuel are responsible for this phenomenon.

Flammability limits are significantly wider for alcohol fuels. Methanol has the widest limits, nearly 5 times that of gasoline; the limits for ethanol fall in a range between these two fuels (34). These wider limits have been used beneficially to achieve lean combustion in SI engines, yielding lower emissions and higher thermal efficiencies (12).

2.2.2 Cetane Rating and Knock

In a Diesel engine, the cetane rating is a measure of a fuel's ability to ignite spontaneously upon injection. Tied intimately to this fuel characteristic is the phenomenon of knock, a result of rapid rates of pressure rise in the cylinder.

Typical SI combustion occurs by ignition at a single point with an orderly movement of the flame front through the homogeneous air-fuel mixture. In this case, knock is associated with autoignition of unburned gases compressed ahead of the flame front and the subsequent rapid rise of pressure. Since orderly flame propagation and autoignition are two distinctly different types of combustion, knock is easily detected by audible percussions emanating from the combustion chamber.

Diesel knock is more difficult to describe quantitatively than SI knock. Diesel combustion begins by autoignition occurring at many points within the mixture, resulting in rates of pressure rise which are much higher than normal SI combustion. However, this does not necessarily imply that audible knock must be occurring (35). Combustion in most Diesel engines follows four major stages:

1) ignition delay - the period of time between start of injection and autoignition,
2) uncontrolled combustion - fuel accumulated during ignition delay burns rapidly,
3) controlled combustion - diffusion processes control burning of fuel injected after combustion has been initiated, and
4) late combustion - combustion of fuel occurring after injection ceases.

Diesel knock is most closely associated with the first two stages of combustion. If the ignition delay is extended, indicative of a low cetane number fuel, a larger amount of fuel will accumulate in the cylinder preceding autoignition. This may cause high rates of pressure rise to occur during the uncontrolled combustion stage. Another factor influencing the severity of stage-two combustion is the ratio of vaporized fuel to cylinder displacement at the instant of ignition.

Cetane ratings for Diesel fuel oil indicate that the ignition qualities of the respective fuel are comparable to a mixture of reference fuels under similar conditions. Ratings for most commercial Diesel fuels range around a cetane number of 50; gasoline, by comparison, has poorer ignitability characteristics and a cetane number of near 20. Alcohols have very low cetane numbers; generally, extrapolated values are between 0 and 5. In addition, alcohols have high autoignition temperatures (46) which may range to nearly 5 times that of gasoline under the same conditions. These ignition characteristics make alcohol a poor CI engine fuel as confirmed by many investigators utilizing alcohol fuels in Diesel engines (33).

Other chemical characteristics aside from low cetane number and high autoignition temperature can be cited as a cause of knock enhancement during alcohol usage. One method of reducing knock in Diesel engines involves reducing the amount of fuel that simultaneously autoignites in stage two, uncontrolled combustion (47). Fuel oil, being a multiconstituent fuel, has a full boiling range; the lighter ends vaporize first and ignite, providing a source of heat energy for further vaporization of the heavier ends. However, pure substances such as alcohols are single boiling point fuels, and will vaporize at the same instant, forming a vapor envelope which is active in the initial combustion reaction. The high energy content of this vapor envelope, released rapidly by high flame speeds and reaction kinetics, creates strong amplitude shock pulses which strike the cylinder wall and cause Diesel knock.

As a result of these problems associated with Diesel combustion of alcohols, present efforts to utilize alcohol fuels generally employ some form of dual fueling. Fumigation is a particularly attractive means of alcohol utilization in Diesel engines; successful operation is achieved with minimal engine modifications. This method of alcohol utilization was used in this work.

2.2.3 Fumigation

Utilization of fuels by spraying, carbureting or injecting the fuel into the intake air stream has been termed fumigation. Fumigation, the standard method of fueling SI engines, has received considerable attention with regard to applications for using alternative fuels in CI engines. Tests in the 1950's showed that a variety of fuels could be utilized in Diesel engines resulting in reduced smoke and increased power depending on the fuel quantity and type (48). Two theories were advanced to explain the observed beneficial effects of fumigation:

1) better air utilization resulting from good mixing of the inlet air and fuel, and
2) preflame reactions during the compression stroke accelerating the reaction process.

More recently, fumigation has been investigated as a means of burning alcohol fuels in Diesel engines. The injected fuel oil acts as a pilot charge, similar to dual injection, providing a source of ignition for the more homogeneous alcohol-air vapor mixture. Both diffusion controlled heterogeneous combustion of the pilot charge as well as homogeneous combustion of the vaporized mixture of alcohol are, therefore, present in fumigated Diesel engines.

The degree of alcohol fuel vaporization in the intake air stream controls the charge temperature as well as the homogeneity of the inducted mixture. As discussed in Section 2.2.2, charge cooling in Diesel combustion may cause increased ignition delay periods, combustion knock, and in some cases engine misfire due to quenching of the combustion process. Some investigators have utilized waste exhaust or supplementary energy sources to preheat the inlet air to minimize these effects.

Work by Houser et al. (33) involved methanol fumigation of a light-duty automotive Diesel engine with up to 40% of the energy requirement. Upper limits of alcohol substitution were set by deterioration of thermal efficiency at light load due to poor combustion and knock limited operation at higher loads. Investigators using fumigation have attempted to achieve a large amount of alcohol fuel vaporization well ahead of the intake valve and combustion chamber. Varying from this practice, Bro and Pederson (49) used an electronic fuel injection nozzle positioned in the intake air circuit, close to the cylinder head. Injection was intermittent rather than continuous. Intake air temperature, tied intimately to the degree of fuel vaporization, was found to have a significant effect on the combustion process. Three types of combustion were identified:

1) single combustion, in which only the pilot spray envelope and alternative fuel contained within the envelope were combusted,
2) consecutive combustion, in which combustion of the pilot charge preceded and acted as an ignition source for the alternative fuel, and
3) simultaneous combustion, in which the pilot charge as well as the alternative fuel burned coincidentally.

The cooling effect, which so profoundly alters combustion in alcohol-fumigated Diesel engines, may be put to practical use, however. Since turbocharged engines tend to benefit either by an aftercooler or liquid evaporator, the high latent heat of vaporization of alcohol may be utilized. In one test, methanol was added to the intake air through a pressurized valve at the turbocharger inlet (50). Better air utilization led to reduced smoke and increased power-per-weight with the same emission levels.

By nature of its simple application to existing Diesel systems, fumigation presents an attractive means of dual fueling. Waste heat reclamation could conceivably be applied to minimize combustion problems common during alcohol fumigation, making this method of dual fueling even more attractive.

2.2.4 Effects of Water Addition

Water and alcohols are infinitely soluble. During distillation, particularly in the case of ethanol, water which is produced during manufacture is removed. The quality or water content of the final product is a direct function of the degree of distillation as well as the opportunity for contamination of the alcohol following this process. As mentioned earlier, commercial stills may easily obtain 190 proof alcohol; however, smaller home-operated stills may produce lower quality alcohol, often 140-180 proof.

If lower quality alcohol is used in Diesel engine applications, the effects of water on combustion must be considered. Water has been added to the combustion chamber in many SI engine tests to reduce knock and certain gaseous exhaust emissions (51, 52). More recently, water addition has been investigated as an aid to Diesel combustion (53). Oxides of nitrogen (NO_x) emission reduction has frequently been a goal of water addition studies. Concentrations of NO_x in exhaust gases are closely related to peak temperatures reached during combustion. Since chemical equilibrium and reaction rates are highly temperature dependent, the significant action of water vapor appears to be its ability to reduce peak combustion temperatures.

When considering the proposed Zeldovich mechanism of NO production, the temperature-related effects of water appear to dominate by slowing the reaction kinetics. However, the presence of water vapor during combustion produces several results. Oxygen radicals may be reduced by the presence of water vapor by the following reaction (54).

$$H_2O + O \rightarrow 2OH \qquad 2.2$$

This reduction of oxygen radicals further slows the Zeldovich mechanisms. (For further discussion on emissions, the reader is referred to Section 2.3, Emissions.)

In one study, SI combustion temperatures were reduced primarily as a result of the dilution of the charge molar and energy density by the water vapor, thus accounting for a reduction in NO_x (55). Related work included fumigation of a CI engine with similar results: reductions in nitric oxides and a slight increase in ignition delay. Water has also been introduced into CI engines by forming water-fuel emulsions. Microexplosions resulting from rapidly vaporizing water particles acted as atomizers which formed smaller and more finely dispersed fuel droplets. Observed reductions in NO_x resulted from the creation of more homogeneous combustion conditions by improved fuel distribution. In addition to reduced NO_x emissions, better combustion with less smoke resulted (53).

Water might also be expected to affect unburned hydrocarbon emissions. Strong evidence suggests that wall and midair flame quenching may be primarily responsible for the production of this pollutant. Increased amounts of water vapor may cool combustion to the extent that the quench layer is thickened and, in the limit, misfire occurs. These factors tend to increase hydrocarbon emissions.

In light of these considerations, the combustion effects of water in low-quality alcohol may be summarized as follows:

1) reductions in NO_x,
2) increases in unburned hydrocarbons,
3) increased ignition delay, and
4) misfire due to midair and wall flame quenching.

2.3 Emissions

Concern over the health and environmental effects of gaseous and solid emissions produced by automotive sector has led to more stringent control of certain combustion products. This section discusses the major pollutants emitted by Diesel engines and the possible effects of alcohol fumigation on their formation.

2.3.1 Gaseous Emissions

At the present time, three gaseous exhaust emissions are regulated by the federal government under pollution control legislation. These are nitrogen oxides, carbon monoxide, and unburned hydrocarbons. Typical concentrations of these pollutants in Diesel exhaust are shown in Table 2.4. The chemical formation of each of these exhaust pollutants follows a complex series of reaction steps. Stoichiometric calculations show that for dodecane, a pure hydrocarbon characteristic of Diesel fuel oil, the following reaction would occur:

$$C_{12}H_{26} + 18.5 O_2 + 69.6\ N_2 \rightarrow 12\ CO_2 + 13\ H_2O + 69.6\ N_2 \qquad 2.3$$

Table 2.4

Typical Diesel Emission Levels

Pollutant	Volume Concentration	Power Specific Mass (gm/bhp-hr)
Smoke	1-10 mg/ft^3	0.05-0.3
Carbon Monoxide	50-5000 ppm	2-10
Oxides of Nitrogen	200-300 ppm	4-20
Hydrocarbons	10-500 ppm	.2-3

Source: Bascom, et al. (56)

This reaction equation shows that the major pollutants are not predicted, indicating that normal combustion does not follow this simple stoichiometry. Lean and rich areas of combustion are normally present in compression ignition engines. Opposing, simultaneous, and consecutive reactions in the gas phase may compete to form products far different from those predicted in Eq. 2.3. In addition to chemical kinetics, the design of the combustion chamber and injection system has a significant effect on heat transfer and turbulent mixing. This may strongly affect combustion and the composition of the exhaust gases.

In light of the complex reactions involved in the formation of these gaseous emissions, prediction and explanation of the exhaust gas composition is, at best, difficult. However, some general trends in Diesel emissions can be explained; in addition, the effects of alcohol fumigation on the formation of these emissions can be predicted.

2.3.1.1 Hydrocarbons and Aldehydes

Unburned or partially oxidized fuel may appear in the exhaust gases as hydrocarbons or oxygenated species such as aldehydes or ketones (40). Several theories exist detailing the formation of unburned hydrocarbons; most promote the idea of wall quenching and incomplete combustion as the cause of these pollutants.

Hydrocarbon emissions in the Diesel engine are generally lower than in a comparable SI engine. This is a result of the diffusion-controlled combustion process, characteristic of CI engines; air surrounds the burning fuel charge, shielding the flame from the combustion chamber wall and effectively minimizing quenching. Incomplete combustion is reduced due to the high compression ratios of the Diesel engine which provide favorable oxidation conditions. This tends to eliminate midair-induced quench HC emissions which may occur in lean A/F ratio regions (57). Problems unique to Diesel combustion such as leaking or dripping injectors tend to increase HC emissions, however (40). Although HC emissions are generally lower for the CI compared to SI engine, the emitted compounds tend to be more photochemically reactive, toxic, and biologically active than those from a catalyst-equipped SI vehicle (40).

Fumigation of alcohols in Diesel engines introduces the effect of homogeneous as well as diffusion-controlled combustion. The aforementioned effects of wall quenching on homogeneous combustion may be expected to increase the amount of unburned hydrocarbons in the exhaust. In addition, the high latent heat of vaporization of the alcohols may tend to lower combustion temperatures, conceivably thickening the quench layer as well as increasing the occurrence of midair flame quench. To a lesser degree, higher flame speeds (which may reduce heat losses and thus effectively decrease the quench layer) as well as wider lean flammability limits would be expected to reduce hydrocarbon emissions.

In addition to unburned hydrocarbons, recent attention has been given to certain other organic emissions. These unregulated yet potentially hazardous compounds include oxygenated species such as aldehydes and ketones (40). As such, these species are not true hydrocarbons. Aldehydes are a class of organic compounds which may be formed by the oxidation of alcohols and are generally more common during the combustion of alcohol fuels. Alcohol-fueled CI engines would, therefore, be expected to produce higher concentrations of these emissions.

2.3.1.2 Carbon Monoxide

By virtue of its unthrottled operation, the CI engine runs at relatively high A/F fuel ratios compared to SI engines. Carbon monoxide emissions are typically lower in Diesel engines due to excess oxygen present during combustion. However, locally fuel-rich zones are responsible for some carbon monoxide formation. There is also some evidence that CO may be produced in quench envelopes as applied to SI combustion (58). This trend may also exist in CI combustion.

The effect of fumigated alcohol on CO emissions is not expected to be significant since overall lean combustion is maintained. However, quenching due to lower temperatures during combustion may lead to slight increases in CO production. Offsetting this result will be the tendency for CO production (normally a high temperature phenomenon) to be slowed by the lower reaction temperatures.

2.3.1.3 Oxides of Nitrogen

Oxides of nitrogen (NO_x) play a significant role in the reaction cycles of photochemical smog. Automobiles are responsible for a large share of NO_x emissions, of which nitric oxide (NO) constitutes the major portion. Although different models have been proposed for the formation of NO, most of these agree on the important effect of temperature on equilibrium kinetic parameters.

The formation of nitric oxide can be shown as

$$N_2 + O_2 \rightleftarrows 2NO \qquad 2.4$$

However, NO formation does not normally follow this kinetic route, since oxygen and nitrogen molecules do not combine in this manner (42). Rather, the Zeldovich mechanism,

$$O + N_2 \rightarrow NO + N \qquad 2.5$$

$$N + O_2 \rightarrow NO + O \qquad 2.6$$

in which a dissociated oxygen molecule attacks a nitrogen molecule, thus initiating the chain reactions 2.5 and 2.6, is responsible for most of the NO produced. Reaction 2.5 is slow due to the high activation energy ($k=1.4\times10^{14}\ \exp(-78000/R_oT)$) required; as a result, kinetic rather than equilibrium considerations are the rate-controlling factors.

NO formation usually peaks at a mixture composition slightly leaner than stoichiometric for homogeneous combustion. Although peak temperatures are reached at stoichiometric conditions, kinetic factors push the peak NO formation slightly toward the fuel lean region (54). In diffusion combustion systems of this type, the fuel burns in local, nearly stoichiometric proportions thus producing more NO than would be predicted on the basis of the apparent overall stoichiometry. However, these local mixture conditions are leaner than those normally existing in SI combustion, resulting in correspondingly lower NO emissions.

Although the final mixture temperature may be lowered as unreacted oxidizer dilutes the combustion products, NO levels remain in frozen equilibrium at high temperature concentrations. The hypothesis of peak temperature equilibrium levels is supported by many investigators who agree that the reduction reactions of NO are too slow to be significantly affected by the drop in temperature during later stages of combustion (54, 59).

In previous studies, reductions in oxides of nitrogen during alcohol dual fueling have been reported (59). These reductions were attributed to decreased peak combustion chamber temperatures resulting from the alcohol's high latent heat of vaporization and endothermic dissociation. However, at least one author reported increases in NO_x emissions during alcohol fueling (25). Increased peak pressures and possibly higher peak temperatures due to rapid combustion of the alcohol may be responsible for this trend. In addition to these effects, fumigation of alcohol may alter the mixture composition and the type of combustion (homogeneous or diffusion), thus changing the levels of nitrogen oxide concentration.

2.3.2 Particulate Emissions

Although gaseous pollutant levels found in CI engines are generally low compared to SI engines, the production of particulate matter is considerably higher. Recent attention has been given to the possibly harmful health effects which may be caused both by the solid particulate matter as well as the soluble organic fraction (SOF) of these emissions. If the Diesel engine is to assume a significant role in light-duty vehicles, processes responsible for the formation of these particulate emissions as well as the potential biological hazards associated with their presence must be assessed.

2.3.2.1 Formation

Diesel particulate emissions are on the order of 50 to 80 times that of a comparable SI engine (60). This results primarily from the diffusion type combustion typical of fuel-injected systems. Many identifiable factors alter the formation of soot, primarily through their effect on the mixing of vaporized fuel and oxygen during combustion. These include fuel injection characteristics, combustion chamber design, and combustion chamber mixture turbulence. The chemical make up of the fuel is also important; the relative content of carbon compounds as well as differences in chemical reactivity of the fuel species affect soot formation.

The formation of soot is also highly dependent upon both the local temperature and A/F ratio. Mechanisms which are generally responsible for particulate emissions, occur in fuel-rich areas and include pyrolysis as well as autothermal cracking (61). Simply stated, fuel combusted in the absence of sufficient oxygen forms soot which does not contact sufficient oxidizer until the temperature has been reduced below the level for combustion (35).

Glassman (54) identifies three stages in the soot-forming reaction:

1) nucleation, in which gas-phase reactions occur and lead to condensed-phase solid nuclei,
2) heterogeneous reactions, which occur on the nuclei surface, and
3) agglomeration and coagulation.

Oxidizing reactions compete with stage one of this soot-forming process, emphasizing the importance of the A/F ratio in the high temperature flame envelope.

Alcohol fumigation may be expected to decrease total particulate formation. This may result from physical as well as chemical differences in fuel and combustion characteristics. Alcohols burn with a nonluminous flame, indicating the lack of carbon particles present in the reaction zone. In addition to this effect, leaner combustion regions created by the homogeneous alcohol-air charge should reduce soot formation.

2.3.2.2 Biological Analysis

With the increasing number of Diesel-powered vehicles, there has been renewed interest in the potential biological health hazards posed by particulate emissions. Particulate, for biological purposes, consists of a carbon core surrounded by a soluble organic fraction (SOF) composed primarily of hydrocarbons (62). It is this soluble fraction which is suspected of containing biologically active compounds which may be inhaled by the human respiratory system.

Extensive research has been conducted dealing with the health aspects of particulate, and appears to support these concerns. The physical nature of particulate is such that it can lodge deeply in respiratory passages. At least 90% of all particulate is less than one micron in diameter; this is well within the size range that can be breathed into the lungs and deposited in pulmonary air spaces (62). Compounds of greatest concern in Diesel particulate include polycyclic aromatic hydrocarbons (PAH) which have demonstrated carcinogenicity in previous animal tests (63). Benzo(a)pyrene (B(a)P), a purified PAH,was shown to produce tumors of the respiratory tract in several animals which inhaled particulate matter containing these compounds (64).

Investigation of biological activity can be performed by using long-term animal tests or by short-term tests which assess the changes in bacteria or animal cells. Due to extensive time and monetary requirements involved in animal studies, short-term tests are frequently employed to evaluate mutagenic rather than carcinogenic events (62). Mutagenic events are those in which permanent damage to genetic "blue prints", DNA, occur. By comparison, carcinogenic events are those in which cancerous tumors result. Since genetic mutations statistically correlate with tumor formation or cancer, mutagenic events are often evaluated to determine biological hazards (62).

One mutagenicity test which is commonly used is the Ames Salmonella test. Briefly, bacteria are used which lack an essential amino acid histidine for growth. If a test compound causes DNA changes, bacteria will revert such that this critical amino acid is produced and growth will occur. By counting the number of colonies or revertants per dose of mutagenic tester compound, the mutagenic tendency of the tester compound is determined. A more detailed description of the Ames test is included in Appendix A.

By collecting and testing particulate produced during fuel oil as well as alcohol-fuel oil operation, any changes in biological activity caused by the different fuels can be measured.

Chapter 3

EXPERIMENTAL EQUIPMENT AND PROCEDURE

3.1 Introduction

An experimental program was completed to determine the effects of aqueous alcohol fumigation on engine efficiency and emissions, in accordance with the objectives of this study listed in Section 1.2. This chapter describes the equipment, instrumentation, and experimental procedures used in the completion of this work.

3.2 Engine Set-up

A fully instrumented Diesel engine served to generate experimental data for the purpose of evaluating the effects of alcohol fumigation. The associated engine apparatus permitted speed and load control, as well as evaluation of combustion events related to injection timing and cylinder pressure rise. Detailed descriptions of the equipment are now presented.

3.2.1 Engine and Dynamometer

An AVCO Lycoming Bernard single-cylinder, direct injected Diesel engine was used in this study. The engine chosen was a four-cycle, air-cooled model capable of producing 6 brake horsepower (bhp) at full load and 3000 revolutions per minute (RPM). Table 3.1 lists the pertinent engine and injection system specifications.

The engine was coupled directly to a Westinghouse cradled electric dynamometer which acted both as a motor to start the engine and as an absorbing generator to load the engine during testing. The dynamometer was a closed-loop feedback type which maintained a constant, operator

Table 3.1

Engine Specifications

Bore	3.00 in.
Stroke	3.0625 in.
Displacement	21.7 cubic in.
Compression Ratio	18:1
Intake Valve Specifications	
Diameter	1.281 in.
Opens (Degrees Crank Angle)	19.0° BTDC
Closes (Degrees Crank Angle)	35.0° ABDC
Exhaust Valve Specifications	
Diameter	1.031 in.
Opens (Degrees Crank Angle)	49.0° BBDC
Closes (Degrees Crank Angle)	5.0° ATDC
Injection Timing	27° BTDC
Rated Power (Continuous Output)	6 bhp @ 3000 RPM

selected speed, independent of engine performance. Torque and load measurements were made from a scale which had a resolution of 0.10 pounds or 0.10 foot-pound-force (ft-lb$_f$) torque.

3.2.2 Needle Lift and Crankshaft Phasing Instrumentation

The stock fuel injector body (Bosch #J3671) was modified to permit determination of fuel injection timing. Needle lift was detected by a KAMAN Measuring Systems proximity measuring unit which incorporated a noncontacting inductive sensor and a solid state signal conditioning module. The inductive sensor featured 10 microinch resolution and 10 kilohertz (kHz) frequency response.

A magnetic inductance pickup manufactured by AIRPAX, North American Philips Controls Corporation, sensed the location of a top dead center (TDC) indicator affixed to the crankshaft flange. This unit furnished information regarding the crankshaft orientation.

Outputs from the needle lift and magnetic pickup sensors were displayed on a Nicolet Instrument Corporation Explorer III digital oscilloscope (Figure 3.1). Comparison of the two signals permitted determination of fuel injection timing relative to piston displacement from TDC.

3.2.3 Combustion Pressure Measurement

Cylinder pressure measurements were made using two quartz transducers mounted in the cylinder head. One transducer (Kistler Model 601B) was mounted flush with the combustion chamber, and the second transducer (Kistler Model 6031) communicated with the main chamber through a small connecting passageway in the cylinder head. No damping

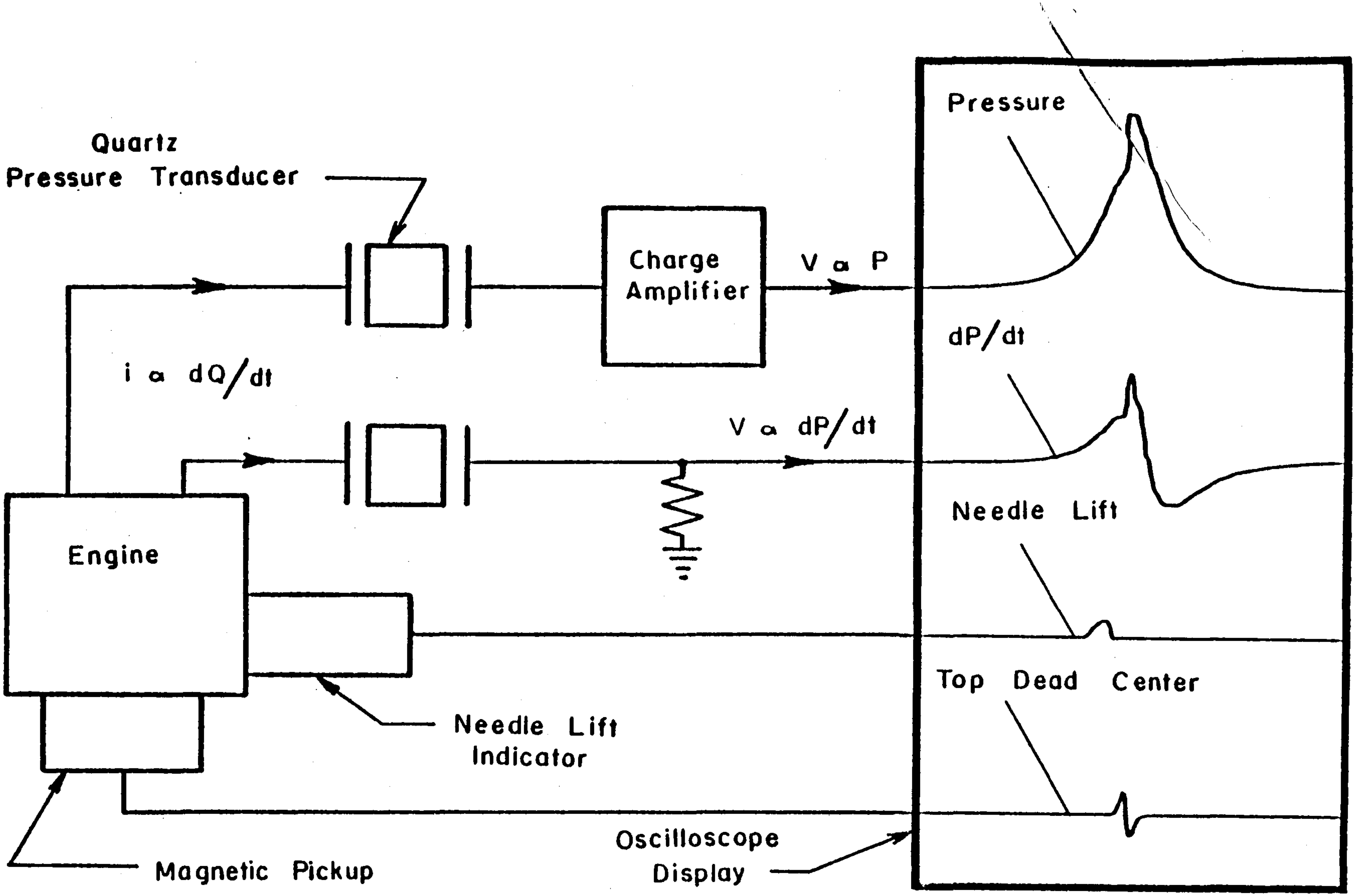

Fig. 3.1 - Combustion Pressure, Rate of Pressure and Injection Timing Instrumentation - Characteristic Output Signals

or frequency effects created by the passageway were detected when signals from the two transducers were compared.

Each transducer acted on the piezoelectric principle. The surfaces of the quartz crystals became electrically charged when stressed mechanically, in this case by pressure forces. This changing charge induced a current which was proportional to the rate of pressure change. By coupling the surface-mounted pressure transducer to a charge amplifier, a voltage signal proportional to cylinder pressure was obtained. In this study, a KIAG Swiss Model 5002 dual charge amplifier massaged the transducer output and delivered an output signal of 1 volt per 100 psi measured in the combustion chamber.

The current from the remote mounted transducer was shunted to ground through a high impedance resistor. The resulting signal, a voltage proportional to rate of pressure change, was evaluated as a means of knock quantification. (The knock quantification system is described in more detail in Appendix B.) The pressure transducer circuits and characteristic output signals are shown in Fig. 3.1.

3.3 Air-Fuel Induction Management

Careful control and measurement of air and fuel flows into the engine were maintained throughout the study. This section describes the hardware utilized in the management of fuel oil, alcohol and inlet air flows.

3.3.1 Diesel Fuel Oil System

The quantity of fuel oil injected per combustion stroke was regulated through adjustment of a graduated rack control. Actual fuel flow was measured by means of a calibrated rotameter (Brooks Tube

R-2-25A) located upstream of the injection pump and return fuel line. During initial testing, it was found that rotameter calibration was sensitive to temperature-induced changes in density and viscosity of the fuel oil. In order to obtain accurate flow measurements, the fuel oil was passed through a constant temperature water bath adjusted to maintain a fuel oil temperature of 85°F at the rotameter inlet. The fuel oil storage tank was pressurized with nitrogen to hold a 12 psig fuel pressure at the injection pump inlet port. Details of this system are presented in Fig. 3.2.

3.3.2 Alcohol System

Alcohol was fumigated into the intake air charge and entered the engine as a vapor or mist dependent upon the degree of vaporization which occurred. A Spraying Systems Inc. 1/4 J Series air atomization nozzle was used for fumigation. Secondary air was supplied at a constant 10 psig to the nozzle where it mixed with pressurized alcohol (0-10 psig) to form a finely atomized and distributed spray.

The nozzle was located in a 4-inch I.D. steel tube which formed the air circuit for the flow of primary inlet air to the engine. Positioned approximately 4 feet ahead of the inlet manifold, the nozzle delivered a continuous spray of atomized alcohol droplets in the direction of the primary inlet air flow. Alcohol was stored under pressure regulated nitrogen in a stainless steel tank at approximately 15 psig. The flow of alcohol to the engine was controlled by varying the liquid pressure of the alcohol to the atomization nozzle. The liquid pressure was regulated through two flow control valves located immediately after calibrated rotameters (Matheson Co. Inc. No. 602 and

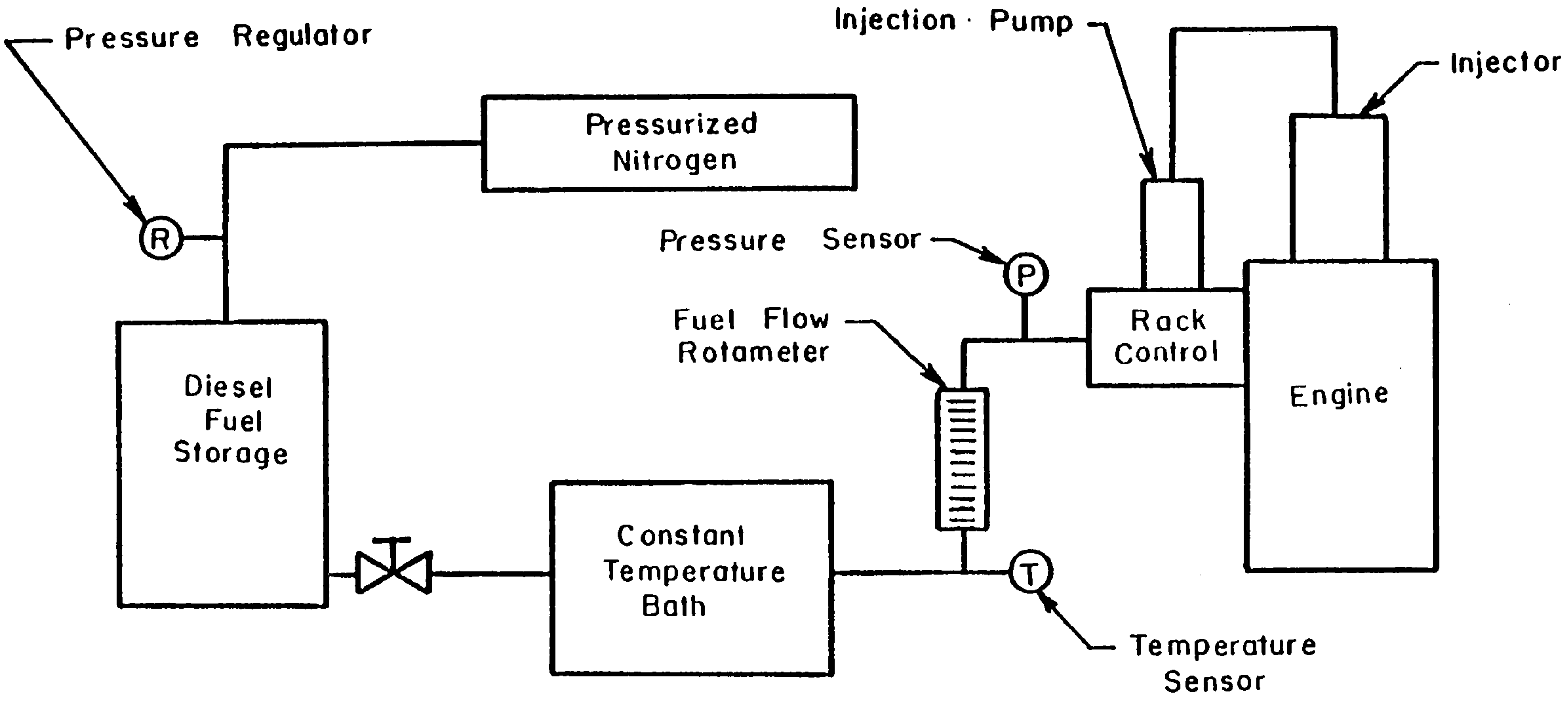

Fig. 3.2 - Diesel Fuel Oil System

603) which measured the alcohol flow. The flow of secondary air was also measured through a calibrated rotameter (Brooks Tube R-6-15-B). The alcohol fuel system and the primary inlet air system (described more thoroughly in the next section) are presented in Fig. 3.3.

3.3.3 Primary Inlet Air System

Figure 3.3 shows a schematic of the primary inlet air and alcohol induction systems which were used in this study. Primary inlet air was dehumidified through an ice bath and subsequently passed through a charcoal filter. A regulator maintained a constant 20 psig pressure in the primary air rotameter (Fischer & Porter Precision Bore Flowrator) used to measure the air flow. Two flow control valves were used to hold manifold pressure at standard test pressure (29.38 in. Hg absolute).

A large plenum chamber located upstream of the engine inlet contained thermostatically controlled heating elements which maintained a steady primary air temperature of 85°F ahead of the alcohol fumigation nozzle.

3.4 Temperature Measurement

Temperature measurements were made using chromel-alumel thermocouples mounted at relevant locations. A Leeds and Northrup balancing potentiometer, calibrated to read directly in degrees Fahrenheit, was used to balance the electromotive force created by the selected thermocouple. The following temperatures were monitored with thermocouples throughout the investigation:

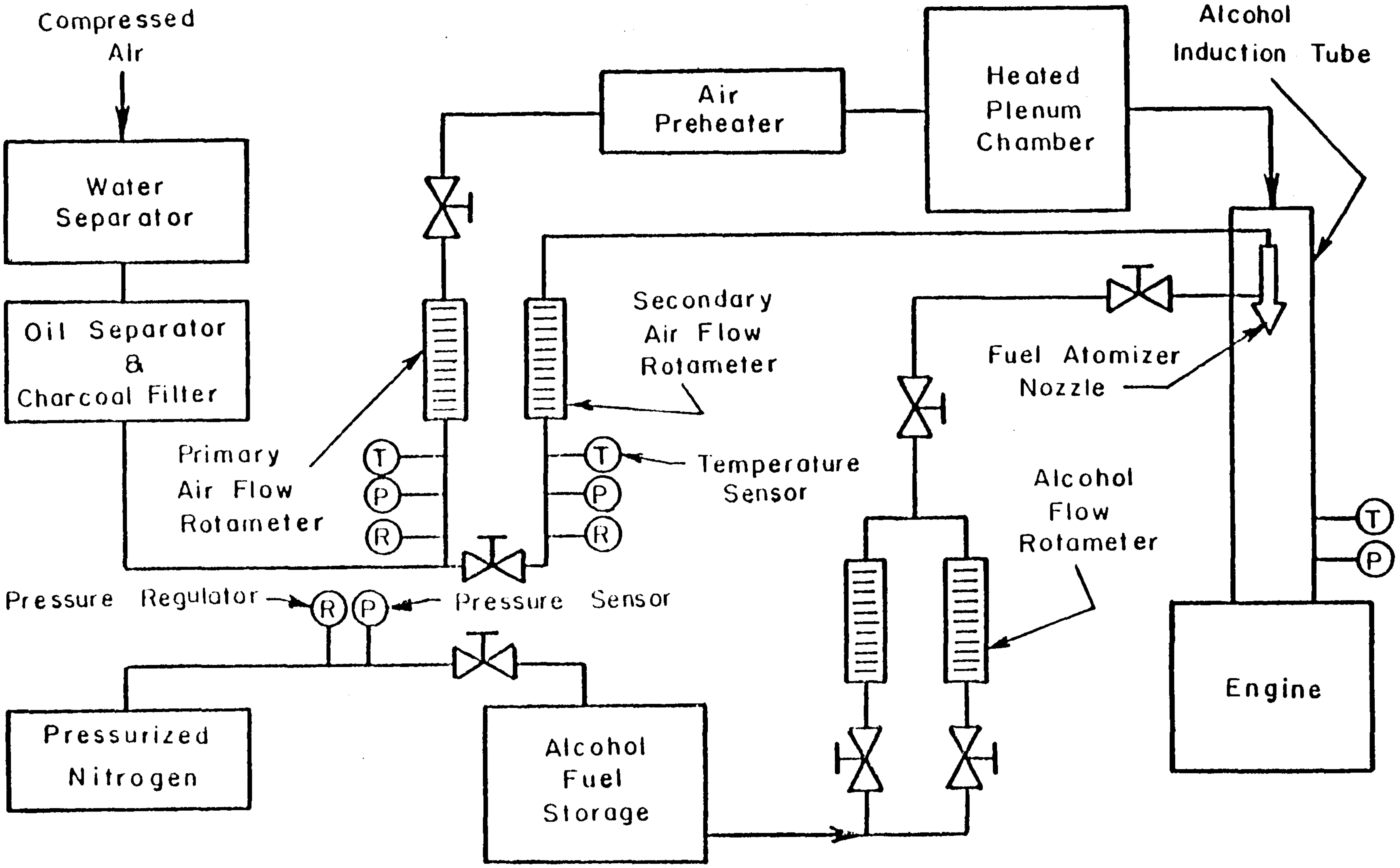

Fig. 3.3 - Alcohol Fumigation and Primary Inlet Air System

1) Oil (3 locations)
2) Exhaust Gases
3) Cylinder Head
4) Inlet Air
5) Cooling Air
6) Plenum Tank Air
7) Alcohol Fuel
8) Particulate Filter

The primary air and Diesel fuel oil flow temperatures were measured using mercury-filled glass tube thermometers.

3.5 Gas Phase Emissions Analysis

The exhaust gases were analyzed for several constituents: CO_2, CO, O_2, NO_x, and unburned hydrocarbons (HC). A representative exhaust sample was extracted continuously from the main exhaust flow through a stainless steel probe located at a point downstream of a large mixing tank. The exhaust extraction system, shown in Fig. 3.4, was also used for particulate emissions collection and is discussed in more detail in Section 3.6, Particulate Emissions Collection. The exhaust gases were then transferred through a Teflon sample line to gas phase instruments for analysis.

Prior to entering these analyzers, the exhaust gases were filtered to remove particulate matter and cooled to condense any vaporized water (Fig. 3.5). Pressure regulators and flow control valves governed the flow rates of span and sample gases through the instruments. Rotameters located within the instruments and on the control panel were used to verify that the correct gas flows passed through each respective analyzer.

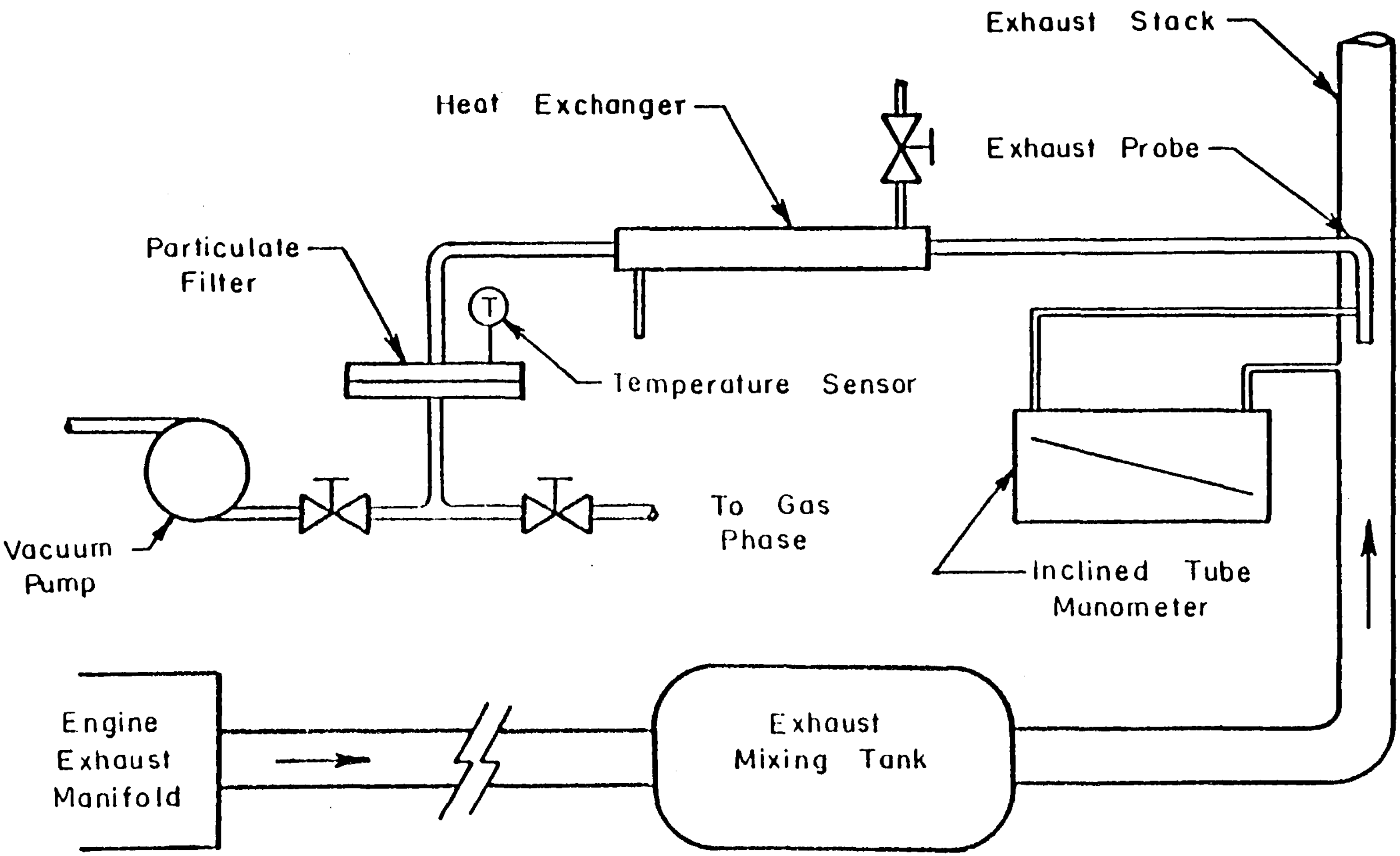

Fig. 3.4 - Diesel Exhaust Sampling and Particulate Collection System

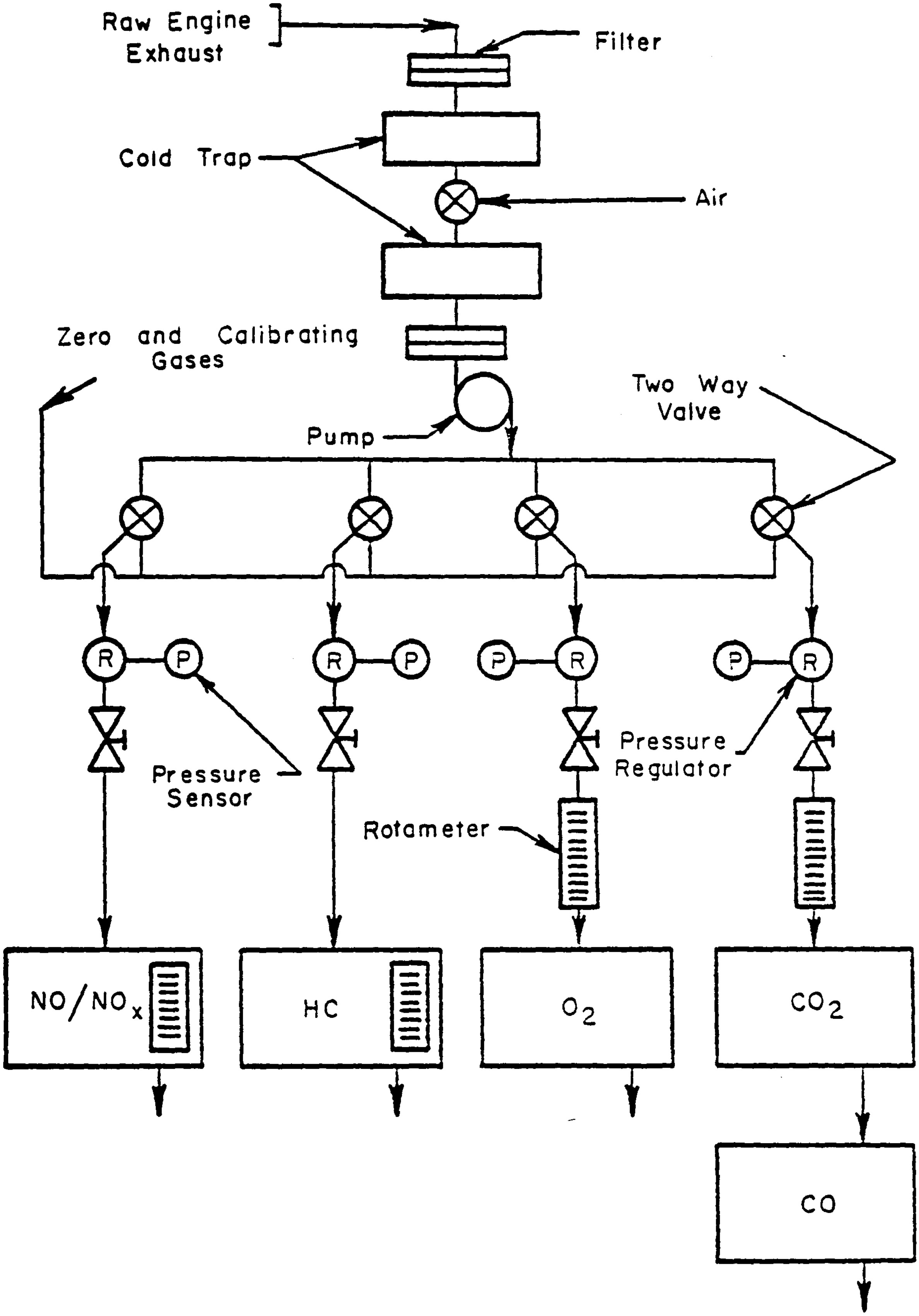

Fig. 3.5.- Gaseous Exhaust Analysis System

Oxides of nitrogen were measured using a Beckman Model 955 heated, chemiluminescent analyzer. The instrument was calibrated on a 550 ppm NO span gas. Detection of CO and CO_2 emissions was made using Beckman Model 864 and Model IR-15A infrared analyzers. These instruments were calibrated on 418 ppm CO and 14.9% CO_2 span gases, respectively. Oxygen was measured by a Beckman Model 741 analyzer; this instrument incorporated an amperometric sensor and was calibrated on high purity air. Unburned hydrocarbons, measured as CH_4, were detected by a Beckman Model 109 unheated flame ionization detector. A certified calibration gas of 530 ppm methane concentration was used for calibration of this unit.

3.6 Particulate Emissions Collection

Particulate exhaust emissions were collected on Teflon-coated, glass fiber filters. The filters, manufactured by Pallflex Products Corporation (Type T60A20), measured 142 mm in diameter and were held in a stainless steel holder located as shown in Fig. 3.4. The exhaust sample was extracted from the center of a 1.610 inch I.D. exhaust stack using a 0.420 inch I.D. stainless steel probe bent at a 90° angle and oriented parallel to the exhaust gas flow. The probe was positioned at a point greater than 10 pipe diameters beyond a stainless steel mixing tank. The Reynolds Number of the exhaust gases at the probe location indicated turbulent flow under all operating conditions; therefore, probe location was not as critical as it may have been had the flow profile been laminar.

In order to avoid mass discrimination and thus extract a representative sample, an inclined tube manometer (Dwyer Manufacturing Co.) was included in the collection system to monitor the relative velocities of the sample and main exhaust flows. When gas velocities were matched in both the exhaust stack and the sampling probe (i.e., isokinetic sampling), the overall pressure differential recorded by the manometer was zero, as predicted by the well-known Bernoulli relationship. Since the chemical nature of the collected particulate could also be affected by high sampling temperatures, a water-cooled heat exchanger was employed to limit all sampling temperatures to less than 125°F.

3.7 Experimental Procedure

The intent of this study was to draw some conclusions concerning the effects of aqueous alcohol fumigation on engine performance and emissions. To this end, a test program was outlined which provided experimental data to offer insight into these areas of interest. An outline of the test program as defined by rack setting, engine speed, and percent alcohol substitution is presented in Table 3.2.

Full rack was determined by adjusting the fuel flow to obtain an output of 4.5 brake horsepower corrected to standard conditions at 2800 RPM. With the rack locked at this position, the engine was loaded such that the speed dropped to 2400, then 1800 RPM. The fuel flow was recorded, and the corrected brake horsepower (BHP_c) was calculated at each full rack condition. Third and 2/3 rack settings were defined as those fuel flows at which the engine developed one-third and two-thirds, respectively, of the full rack BHP_c at each speed. Results obtained in these baseline tests were used to generate a test matrix in which each test cell defined a particular rate of energy input.

Table 3.2

Test Program

Test Series	Rack	RPM	Alcohol Fuel	Alcohol Proof	% Alcohol Subst.
1	1/3 2/3 Full	2800	–	–	0 (Baseline Only)
2	1/3 2/3 Full	2400	–	–	0 (Baseline Only)
3	1/3 2/3 Full	1800	–	–	0 (Baseline Only)
4	1/3 2/3 Full	2400	Ethanol	200	0 to Misfire Limit
5	1/3 2/3 Full	2400	Ethanol	180	0 to Misfire Limit
6	1/3 2/3 Full	2400	Ethanol	160	0 to Misfire Limit
7	1/3 2/3 Full	2400	Ethanol	140	0 to Misfire Limit
8	1/3 2/3 Full	2400	Methanol	200	0 to Misfire Limit
9	1/3 2/3 Full	2400	Methanol	160	0 to Misfire Limit

In completing the remainder of the test program, a rigid procedure was followed in an attempt to prevent any intervening variables from biasing the experimental results. Before each test series, the desired proof alcohol was formed by mixing 200 proof alcohol with distilled water. All alcohol rotameters were recalibrated using this particular alcohol blend. Engine preparation included cleaning all deposits from the cylinder head, piston crown, combustion cup, and injector nozzle. The valves were lapped, and the engine reassembled such that the piston crown-to-cylinder head clearance was held constant at .032±.001 inches.

An attempt was made throughout the study to document any acceleration in wear rates resulting from alcohol fueling. To this end, friction horsepower (fhp) was measured periodically during testing. Cylinder wear rates were also determined during engine tear down by measuring the cylinder bore diameter at several locations.

Actual data collection started by operating the engine at 2/3 rack, baseline fuel, 2400 RPM for approximately 30 minutes until the oil temperature reached 145±5°F. The engine oil was maintained at this temperature for the duration of the test run.

At all alcohol-fueled test points the total energy input rate to the engine was maintained at the value determined in the baseline testing. At each 2400 RPM test condition, performance and emissions data were gathered for the baseline (Diesel fuel only) condition. An incremental reduction in Diesel fuel flow was then accompanied by an energy equivalent increase in alcohol flow such that the total energy input rate to the engine remained constant. The engine was allowed to run at each condition until steady-state conditions were reached; at this point performance and emissions data were recorded. Alcohol was substituted for fuel oil in 10% energy increments until the engine would not run due to combustion quenching. Where necessary, the final increment of alcohol substitution was reduced to 5% by energy to avoid engine misfire.

In addition to gaseous emissions and performance data, exhaust particulate was collected at specific test points for further biological analysis. As for other tests, particulate collection proceeded only after the engine had reached steady-state conditions at a particular operating condition.

Chapter 4

PRESENTATION OF EXPERIMENTAL RESULTS

4.1 Introduction

An experimental program using the equipment and procedures discussed in Chapter 3 and outlined in Table 3.2 was completed to determine the effects of alcohol fumigation on engine emissions and performance. The results of this experimental work are presented in this chapter. Raw data, collected during testing, was reduced to meaningful parameters using an IBM 370 computer system and an Apple II microprocessor. A complete listing of raw and reduced data is presented in Appendix C.

Six fuel combinations of alcohol and water, consisting of four ethanol and two methanol proofs, were used to complete this work. The ethanol data have been plotted separately. For comparison purposes, the methanol data and corresponding-proof ethanol data have been presented together. This data will provide a basis for analysis of changes in engine combustion, efficiency, and emissions during alcohol fueling.

4.2 Engine Performance

Any feasible alternative fuel for automotive usage must offer attractive performance and emissions characteristics compared to conventional hydrocarbon fuels. Damage to critical engine components resulting from changes in fuel type are also of primary concern.

In light of these considerations, several areas of engine performance were examined for changes caused by alcohol fumigation.

These areas of interest included thermal efficiency, ignition delay, combustion intensity, and to a lesser degree, engine wear.

4.2.1 Baseline Testing

In order to provide a data base for comparison during alcohol substitution, a thorough investigation of engine performance on baseline Diesel fuel oil was first completed. Table 4.1 lists the characteristics of the baseline fuel and lubricating oil used in this study. The power output of the engine at 1800, 2400, and 2800 RPM was measured and is shown in Fig. 4.1. During subsequent testing, engine performance was first gauged against these baseline values to verify that all systems were operating properly.

As described in Section 3.5, a test matrix was established during baseline testing which defined the rate of energy input to the engine. This matrix (Table 4.2) specifies, according to rack and engine speed, the energy input rate to be held constant during alcohol substitution.

The measured lower heating values of the Diesel fuel oil and alcohol were used to calculate the correct flow rates of each fuel; flow rates were established to control alcohol substitution to the engine in 10% energy increments.

4.2.2 Engine Efficiency

The thermal efficiency (η_{th}) of an engine can be defined as the fraction of fuel energy supplied to the engine which is converted into useful work:

$$\eta_{th} = \frac{\text{engine work output rate}}{\text{fuel energy input rate}} \qquad 4.1$$

The thermal efficiency results, based on corrected brake horsepower (BHP_c) and fuel lower heating value, are presented in Figs. 4.2 and 4.3.

Table 4.1

Baseline Fuel and Lubricating Oil Specifications

PROPERTIES OF BASELINE TEST FUEL

Fuel Type: MILF 46162 A Grade 2 Diesel

Physical & Chemical Properties	
Gravity (°API)	35.9
Flash Point (°F)	158.0
Pour Point (°F)	-10.0
Cloud Point (°F)	0.0
Viscosity (SUS) @ 100°F	34.2
Cetane No. (calculated)	47.5
Total Sulfur (Wt.%)	.549
Aromatics (%)	36.5

Distillation Properties	
Initial Boiling Point (°F)	376.
10%	430.
50%	490.
90%	575.
End Point (°F)	627.
Recovery (%)	99.

PROPERTIES OF TEST ENGINE LUBRICATING OIL

Oil Type: Shell Rotella T Premium Multipurpose HD

Physical & Chemical Properties	
Saybolt Viscosity @100°F (SSU)	560.0
Saybolt Viscosity @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 Specifications	
Meets	MIL-L-2104C
Exceeds	MIL-L-46152 MIL-L-2104B
API Classification	CD,SE

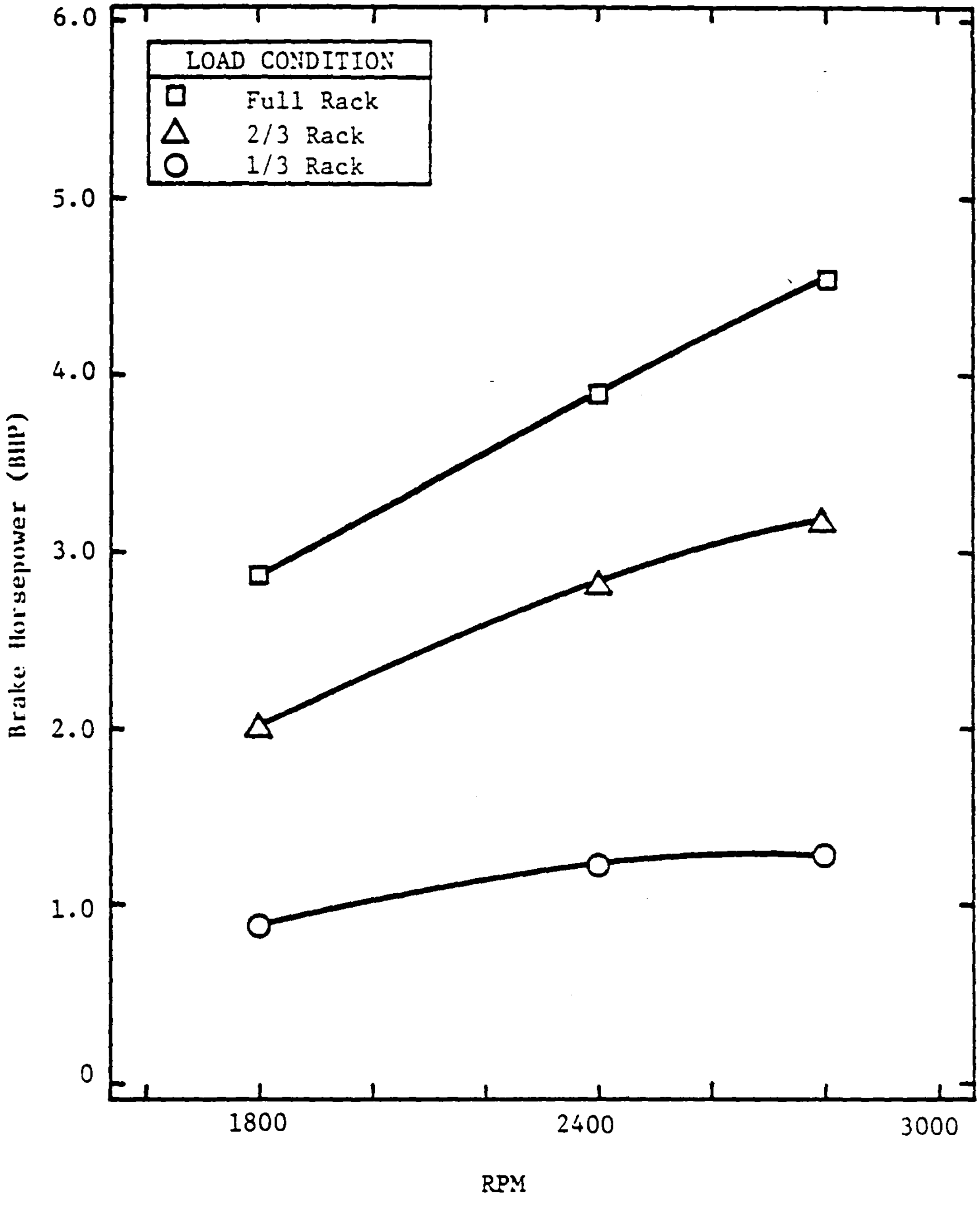

Fig. 4.1 - Brake Horsepower as a Function of Engine Speed at Baseline Operating Conditions

Table 4.2

Baseline Data Matrix for AVCO-Bernard W51 Single-Cylinder DI Diesel Engine

RPM Rack	1800	2400	2800
Full	2.92 59.49 .559 31,304.	3.94 60.26 .572 43,317	4.50 58.94 .613 53,127.
2/3	2.08 42.35 .577 23,200.	2.84 43.50 .600 32,643.	3.30 43.24 .638 40,417.
1/3	.91 18.66 .847 14,885.	1.35 20.69 .847 22,027.	1.41 18.41 1.01 27,226.

Data in each matrix cell organized as follows:

BHP_c (horsepower)
BMEP (psi)
BSFC (lbm fuel/bhp-hr)
Energy input rate (Btu/hr)

Performance data corrected to
<u>Standard Test Conditions</u>
T=545°R (85°F)
P=29.38 in. Hg.

Full Rack Test Horsepower: 4.5 BHP_c @ 2800 rpm

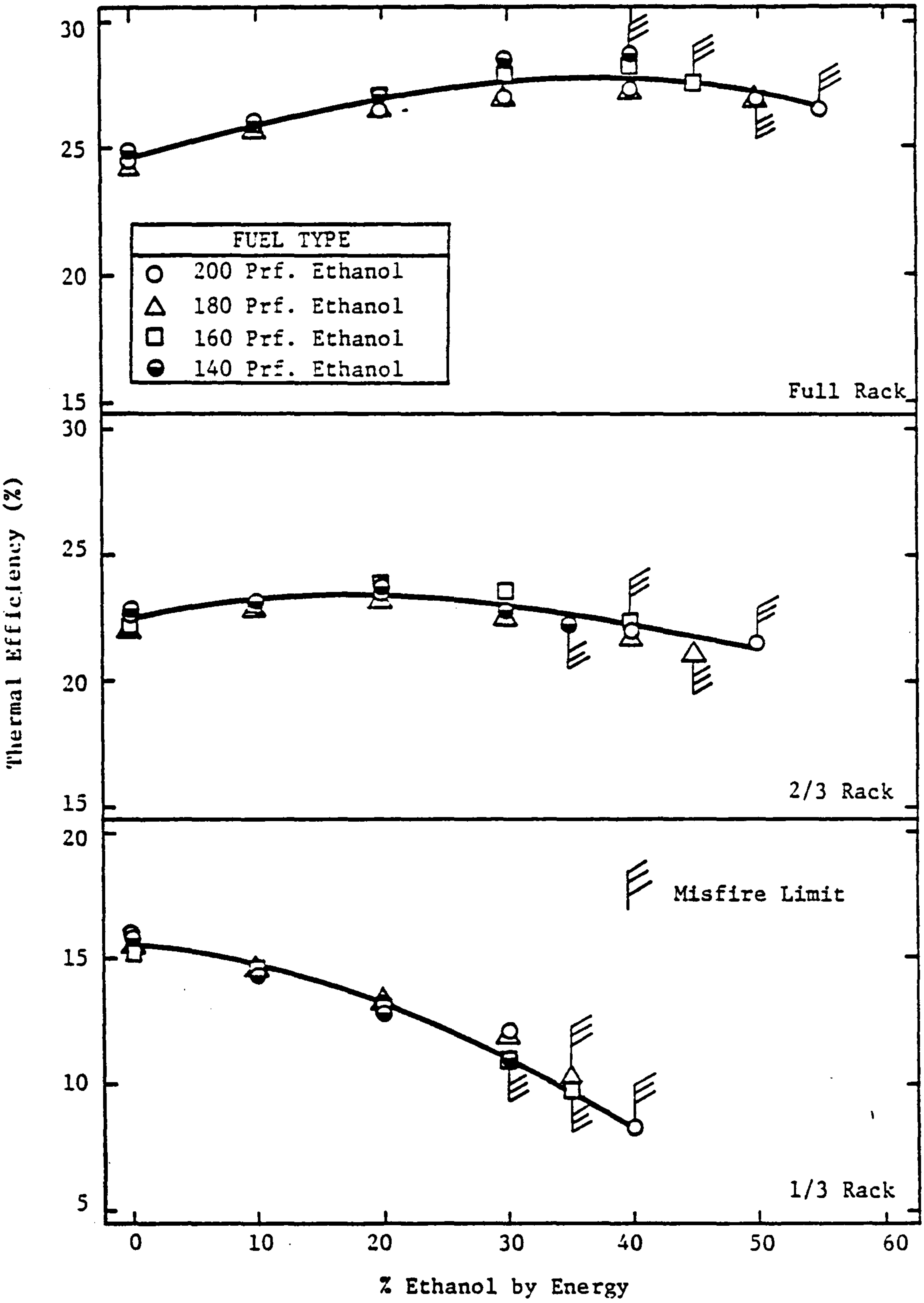

Fig. 4.2 - Thermal Efficiency as a Function of Fumigated Ethanol at 2400 RPM

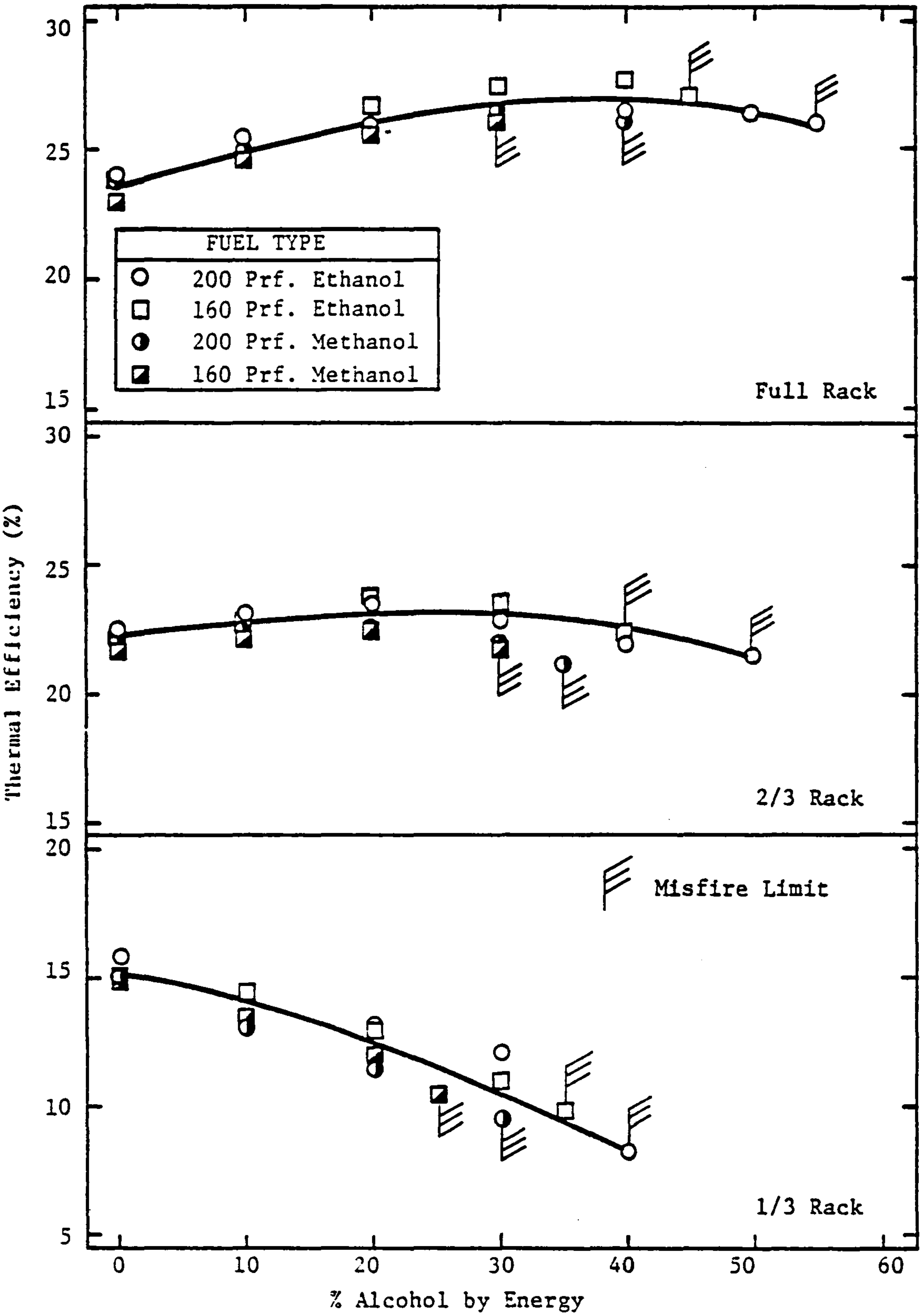

Fig. 4.3 - Comparison of Thermal Efficiencies for Ethanol and Methanol Fumigants at 2400 RPM

Examination of Eq. 4.1 reveals that by holding the energy input rate constant for each specific test condition, the thermal efficiency curves also represent the trend in power output for the engine. Under all operating conditions, alcohol was substituted until engine misfire occurred; the last data point for each fuel and test condition defines the maximum possible substitution of each proof alcohol fuel.

At the 1/3 rack setting, thermal efficiency dropped off significantly for all fuels and test conditions. The 2/3 rack setting initially exhibited a slight increase in thermal efficiency for all fuels tested; however, at the point of engine misfire, efficiency was near its baseline value. Slightly larger thermal efficiency improvements were experienced at the full rack condition and were maintained to the misfire limit.

General trends noted here include the rising percentages of alcohol which could be substituted for Diesel fuel oil as rack setting (load) was increased. In addition, the percentage of water contained in the fuel had no apparent effect on performance; however, combustion quenching occurred earlier for lower proof fuels. Comparison of ethanol and methanol fuels in Fig. 4.3 reveals that ethanol performance was slightly higher at the low rack settings; differences were insignificant as rack setting increased.

4.2.3 Ignition Delay and Combustion Intensity

The physical and chemical differences in fuel structure of alcohols and fuel oil (see Section 2.1, Fuel Characteristics) lead to a combination of changes in the combustion process. The combustion parameters investigated in this study were those largely responsible for the level of mechanical stress placed on the engine.

Maximum pressure and rate of pressure rise (dp/dt) are believed to be two of the more critical factors affecting engine stress. The peak pressure data are presented in Figs. 4.4 and 4.5. The scatter observed in these data may be due to the engine cycle-by-cycle variations. The lack of direct engine-to-computer interfacing prevented analysis of greater numbers of combustion cycles; it is assumed that larger data samples would have minimized scatter. Nevertheless, an obvious trend in peak pressure was observed as alcohol substitution increased.

Under all alcohol-fueled conditions, peak cylinder pressure increased, followed by a decrease near the misfire limit. The maximum percentage increase in peak pressure was observed to occur at the full rack setting and decrease correspondingly with rack setting. No consistent effect of alcohol quality on peak pressure is shown for ethanol or methanol (see Fig. 4.5).

The trend in rate of pressure rise data shown in Figs. 4.6 and 4.7 is similar to that for peak pressure, as might be expected. Again, the rate of pressure increased initially, then dropped off with alcohol addition. Ignition delay, a factor associated closely with rate of pressure rise in Diesel combustion, is shown in Figs. 4.8 and 4.9. An increase in ignition delay occurred as alcohol was substituted for fuel oil. A dependence of ignition delay on water content of the fuel is also indicated by the test data; delay generally increased with

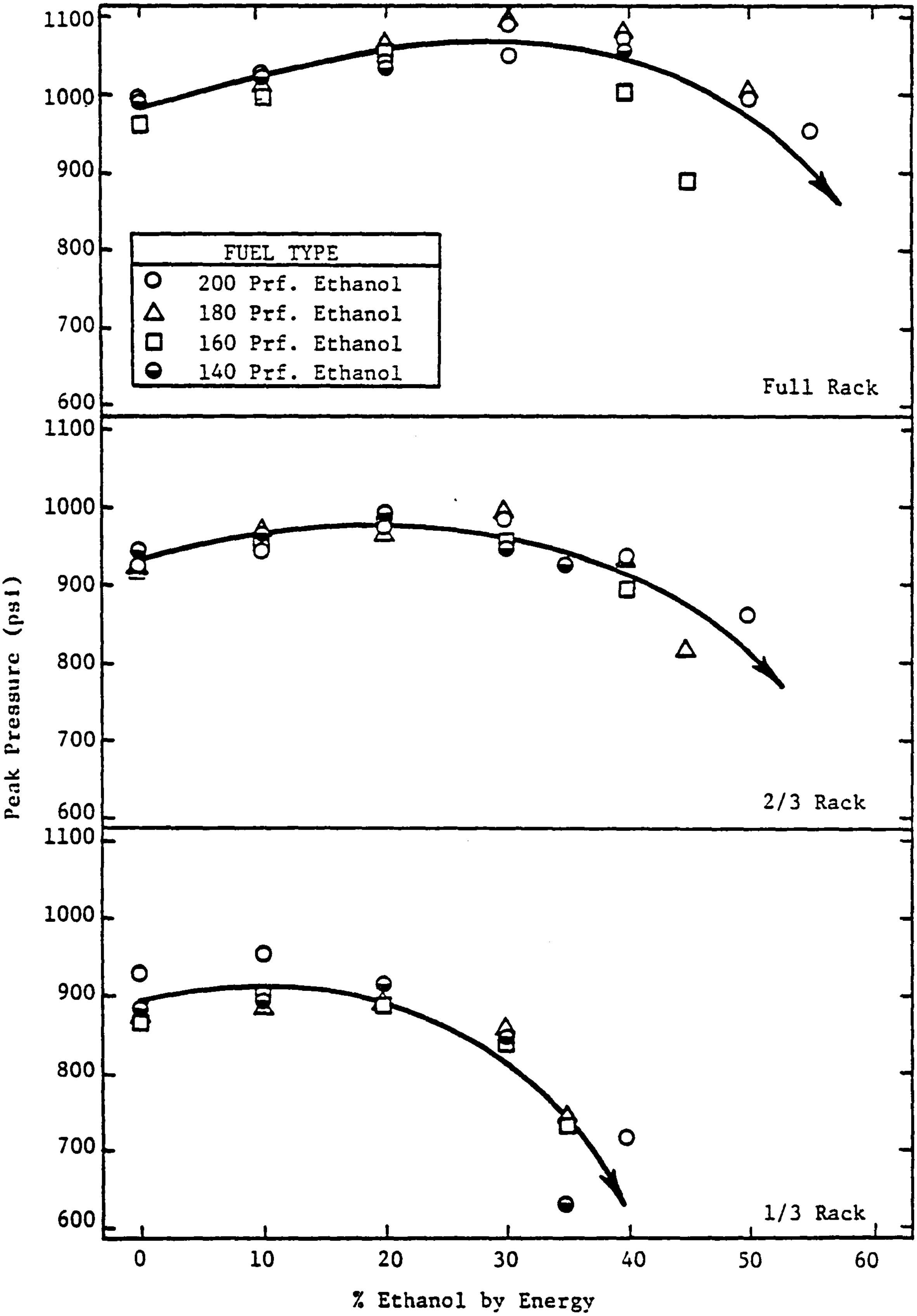

Fig. 4.4 - Peak Pressure as a Function of Fumigated Ethanol at 2400 RPM

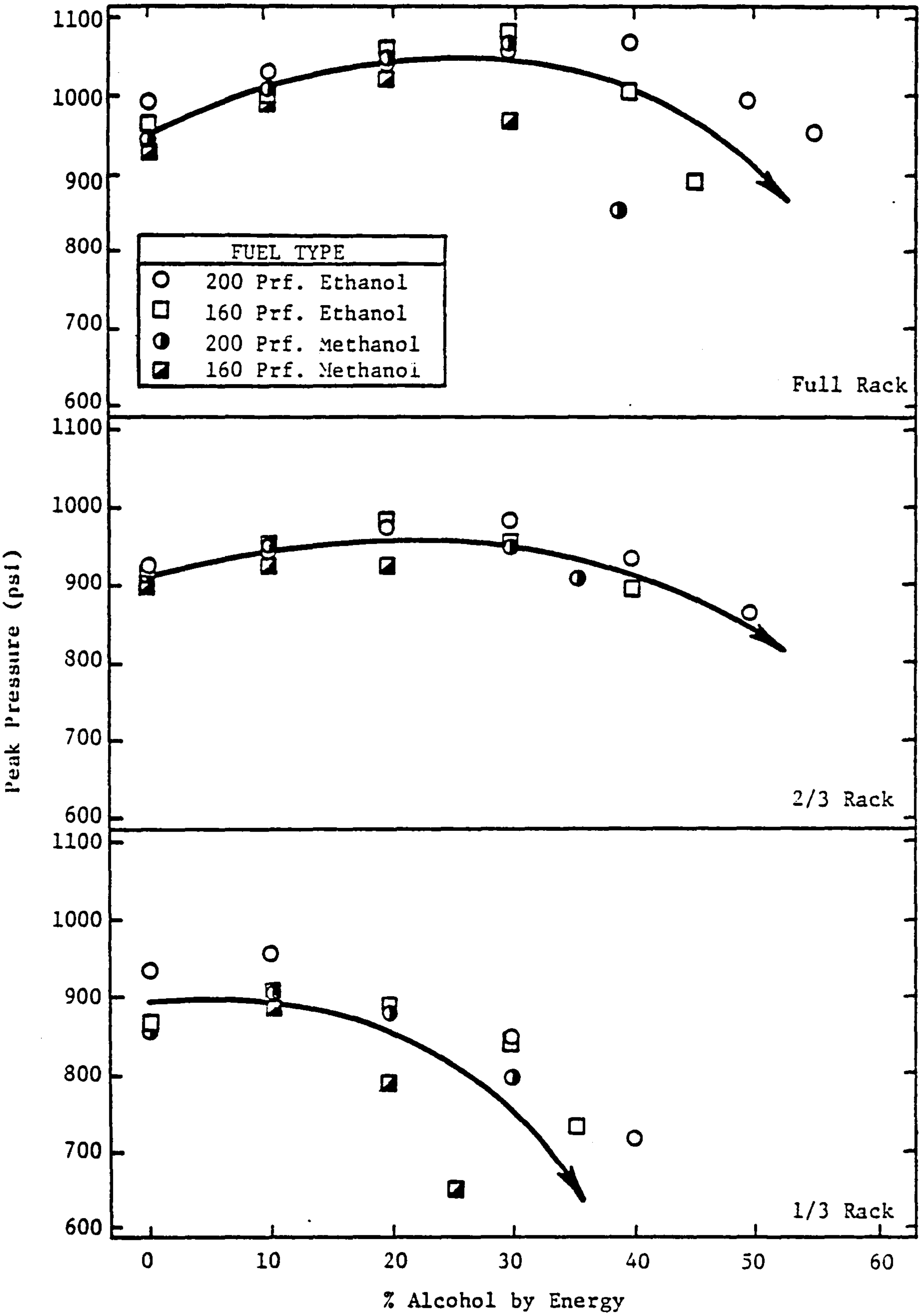

Fig. 4.5 -. Comparison of Peak Pressures for Ethanol and Methanol Fumigants at 2400 RPM

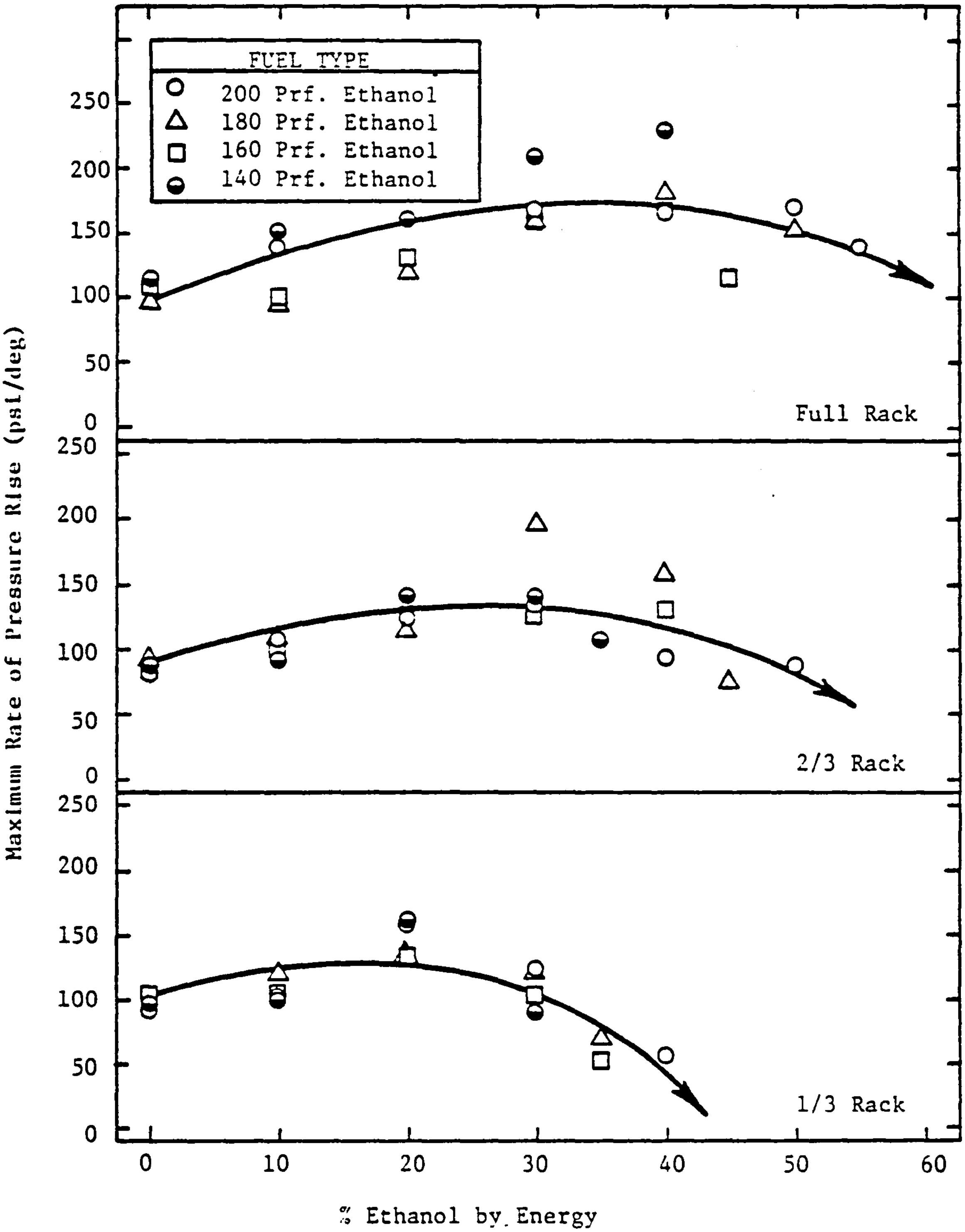

Fig. 4.6 - Maximum Rate of Pressure Rise as a Function of Fumigated Ethanol at 2400 RPM

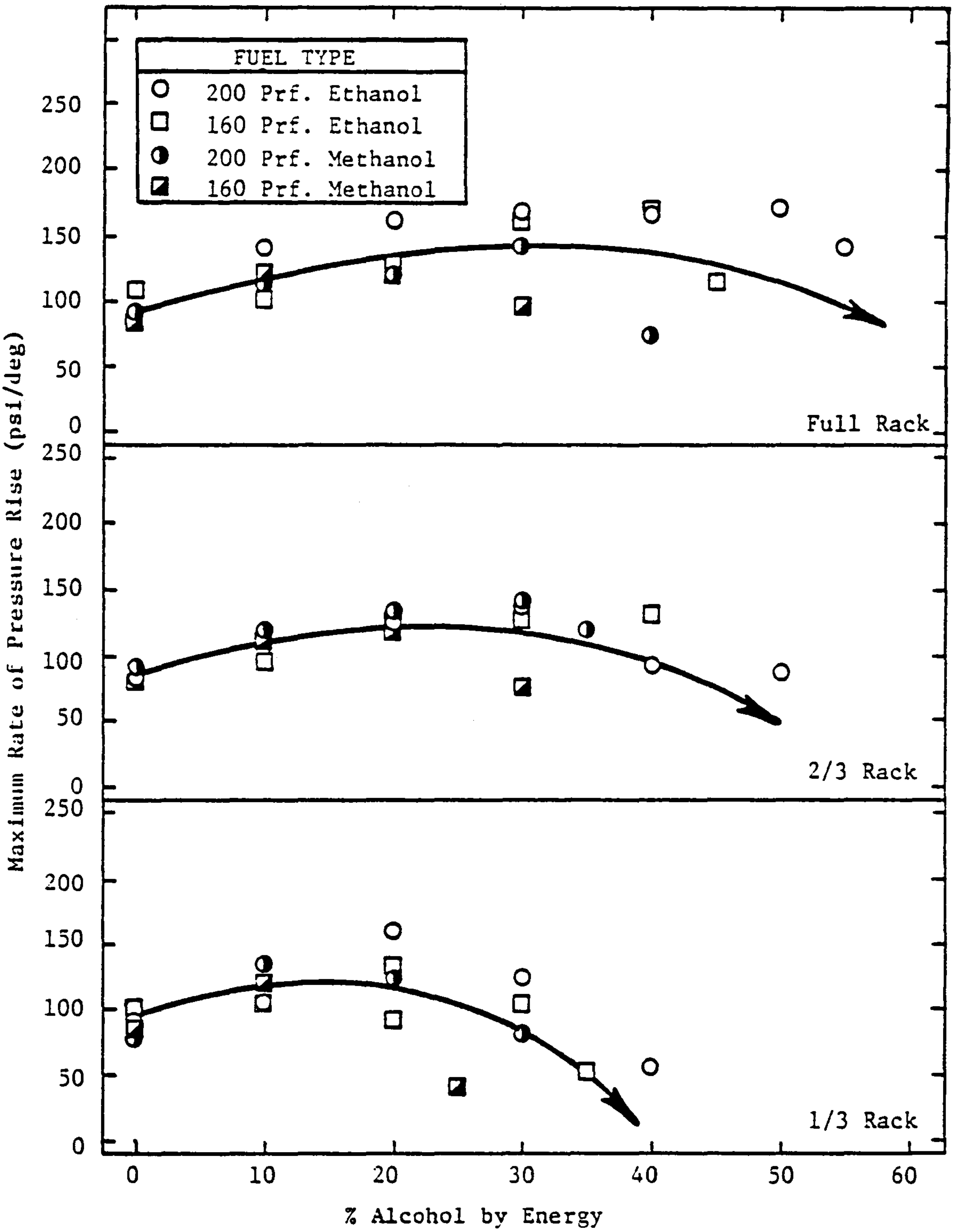

Fig. 4.7 - Comparison of Maximum Rates of Pressure Rise for Fumigated Ethanol and Methanol at 2400 RPM

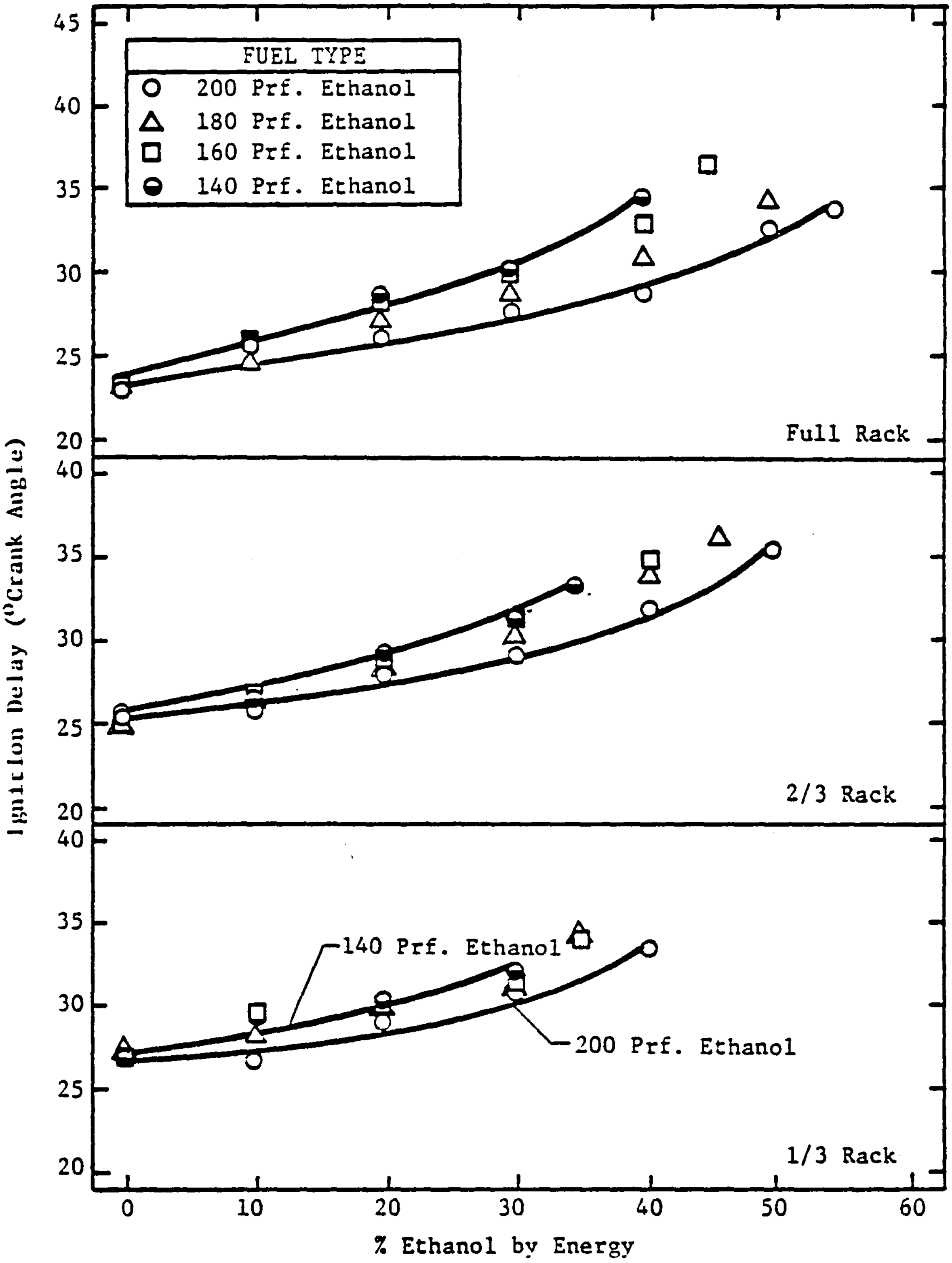

Fig. 4.8 - Ignition Delay as a Function of Fumigated Ethanol at 2400 RPM

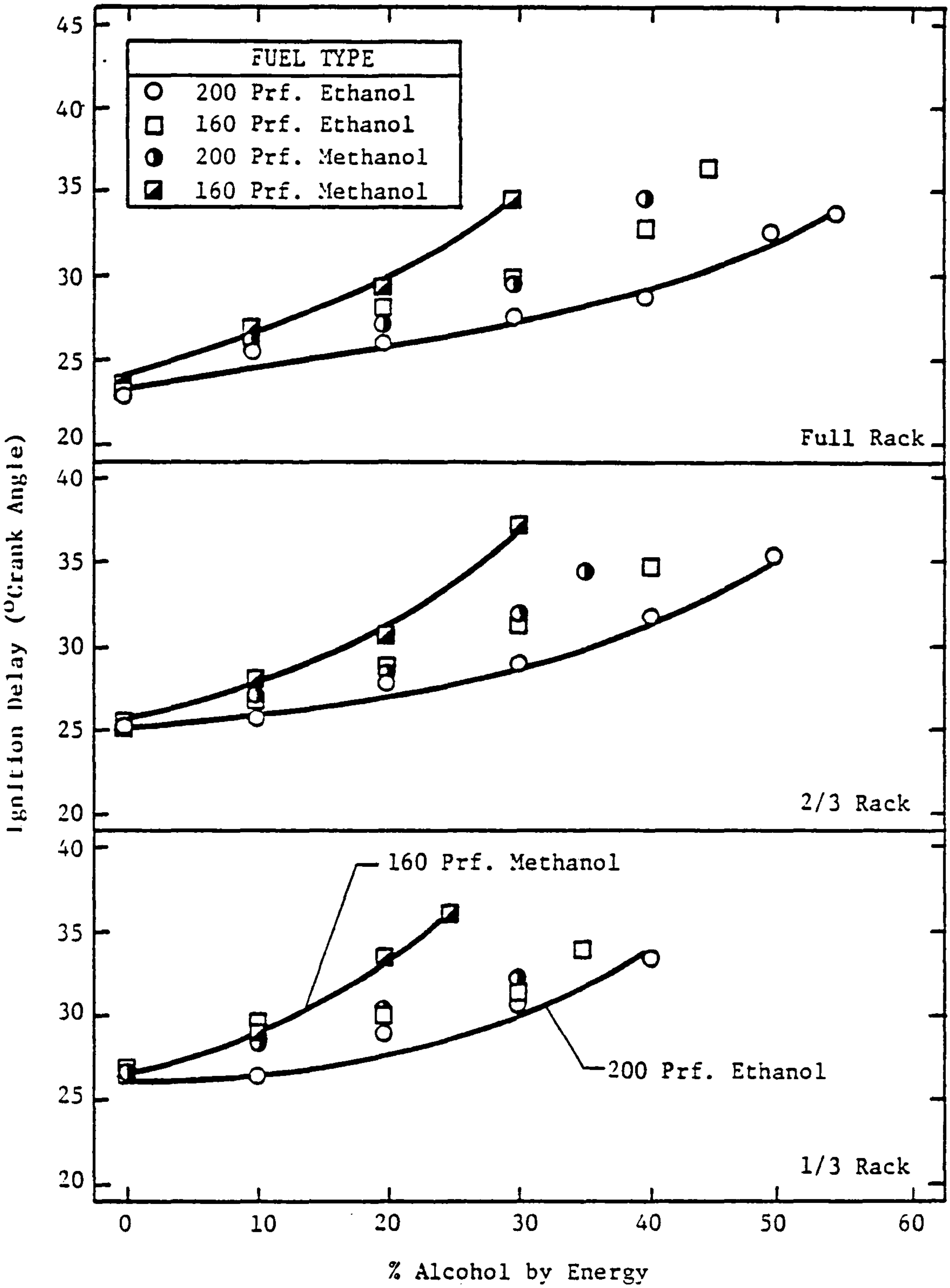

Fig. 4.9 - Comparison of Ignition Delays for Ethanol and Methanol Fumigants at 2400 RPM

decrease in alcohol quality, particularly at conditions where water volume was significant (i.e., higher rack settings and alcohol substitution points).

Comparison of methanol and ethanol fuels in Fig. 4.9 indicates that the ignition delay of 200 proof methanol was similar to that for the 160 proof ethanol. The ignition delay of 160 proof methanol was significantly longer than for similar proof ethanol. A further correlation between ignition delay and maximum percent alcohol substitution is shown in Figs. 4.8 and 4.9; in general, alcohols exhibiting the longest ignition delays caused misfire at severely reduced alcohol substitution levels.

Pressure traces recorded during combustion of each fuel are shown in Figs. 4.10-4.15. The fraction of fuel energy supplied by ethanol (E) or methanol (M) is noted for each curve. The peak pressure, rate of pressure, and ignition delay data presented earlier were measured from similar recorded events. Qualitative examination of these various pressure curves provides an additional insight to the effects of alcohol and water fumigation on engine combustion.

4.2.4 Engine Wear

Checks for upper-cylinder wear were made periodically during engine tear down. The cylinder was measured and visually inspected for signs of scoring or pitting. During the duration of this study, no abnormal rates of wear were recorded. Carbon deposits were observed to form on the nozzle and piston crown; however, the formation of these deposits did not seem to be affected by alcohol quality.

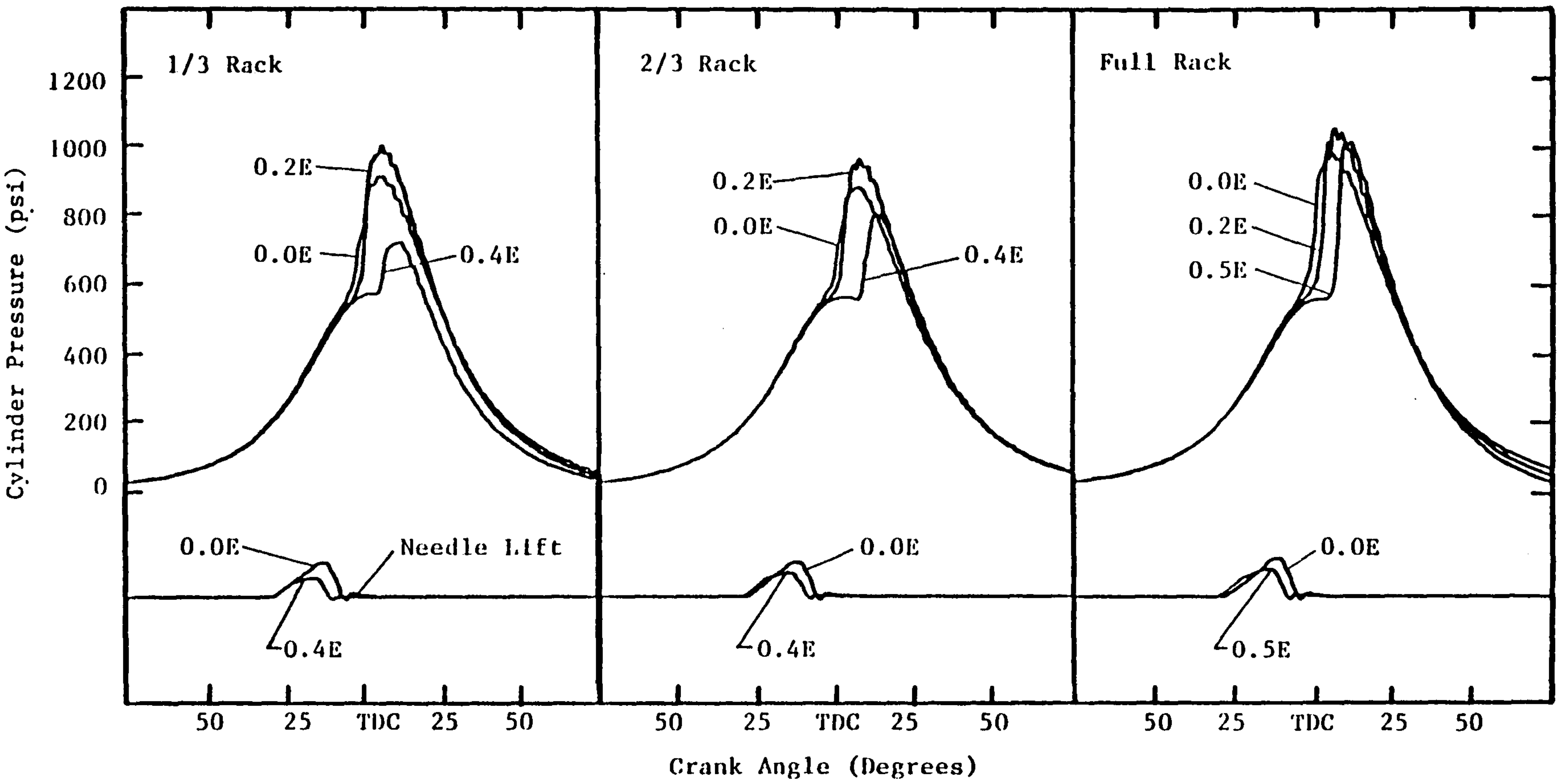

Fig. 4.10 - Comparison of Characteristic Pressure and Needle Lift Histories for Baseline (0.0E) and Various 200 Proof Ethanol Fumigated Conditions at 2400 RPM

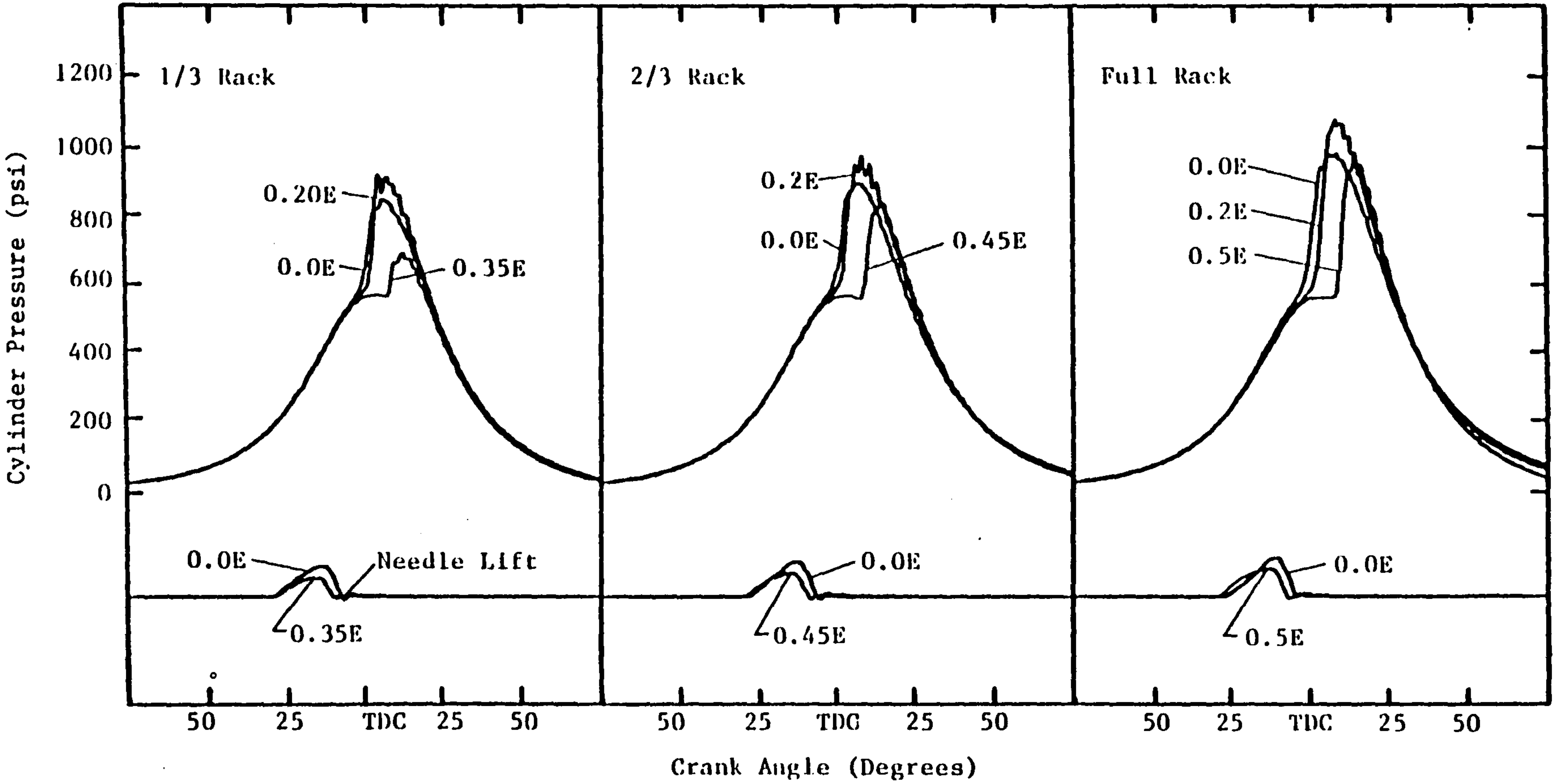

Fig. 4.11 - Comparison of Characteristic Pressure and Needle Lift Histories for Baseline (0.0E) and Various 180 Proof Ethanol Fumigated Conditions at 2400 RPM

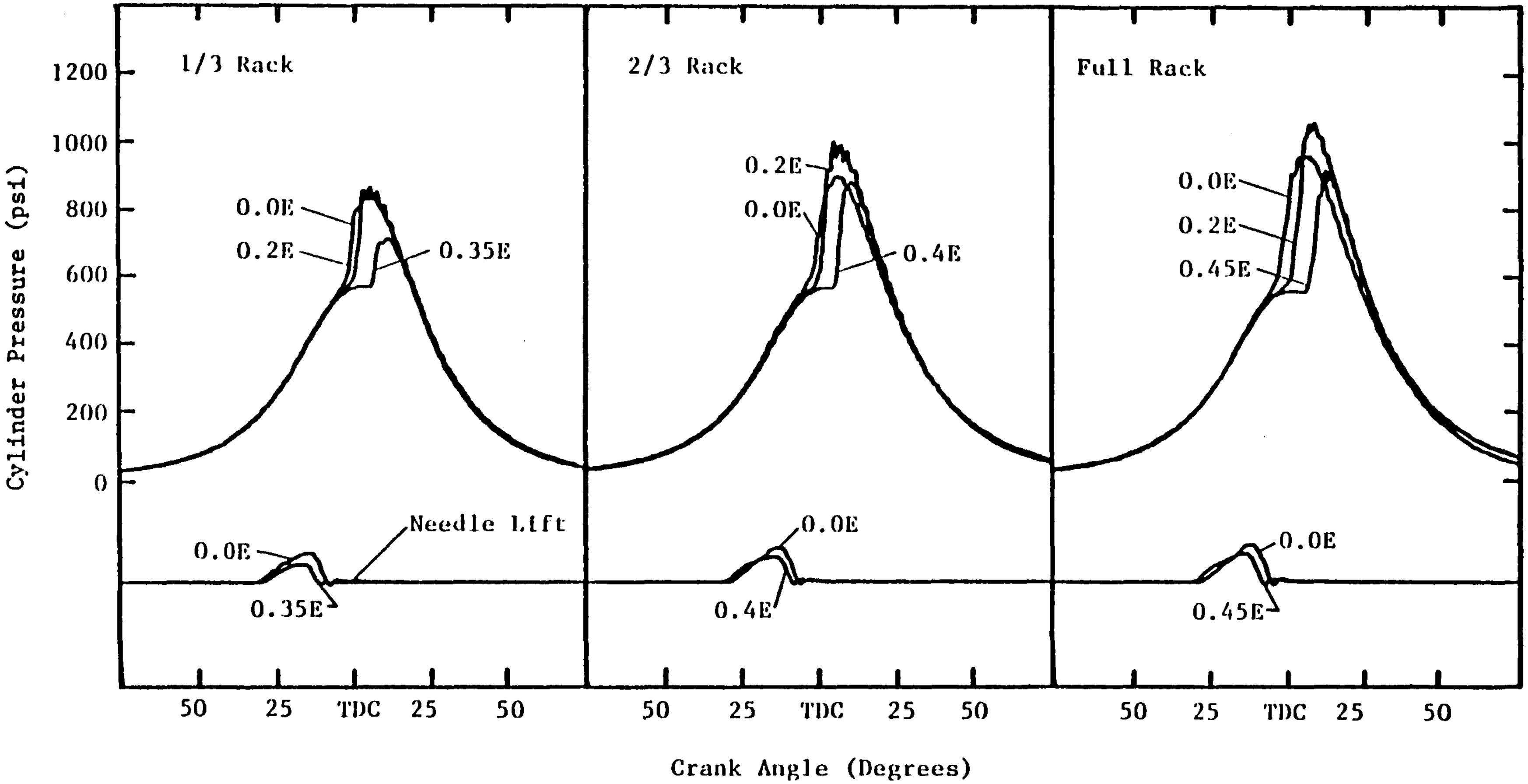

Fig. 4.12 - Comparison of Characteristic Pressure and Needle Lift Histories for Baseline (0.0E) and Various 160 Proof Ethanol Fumigated Conditions at 2400 RPM

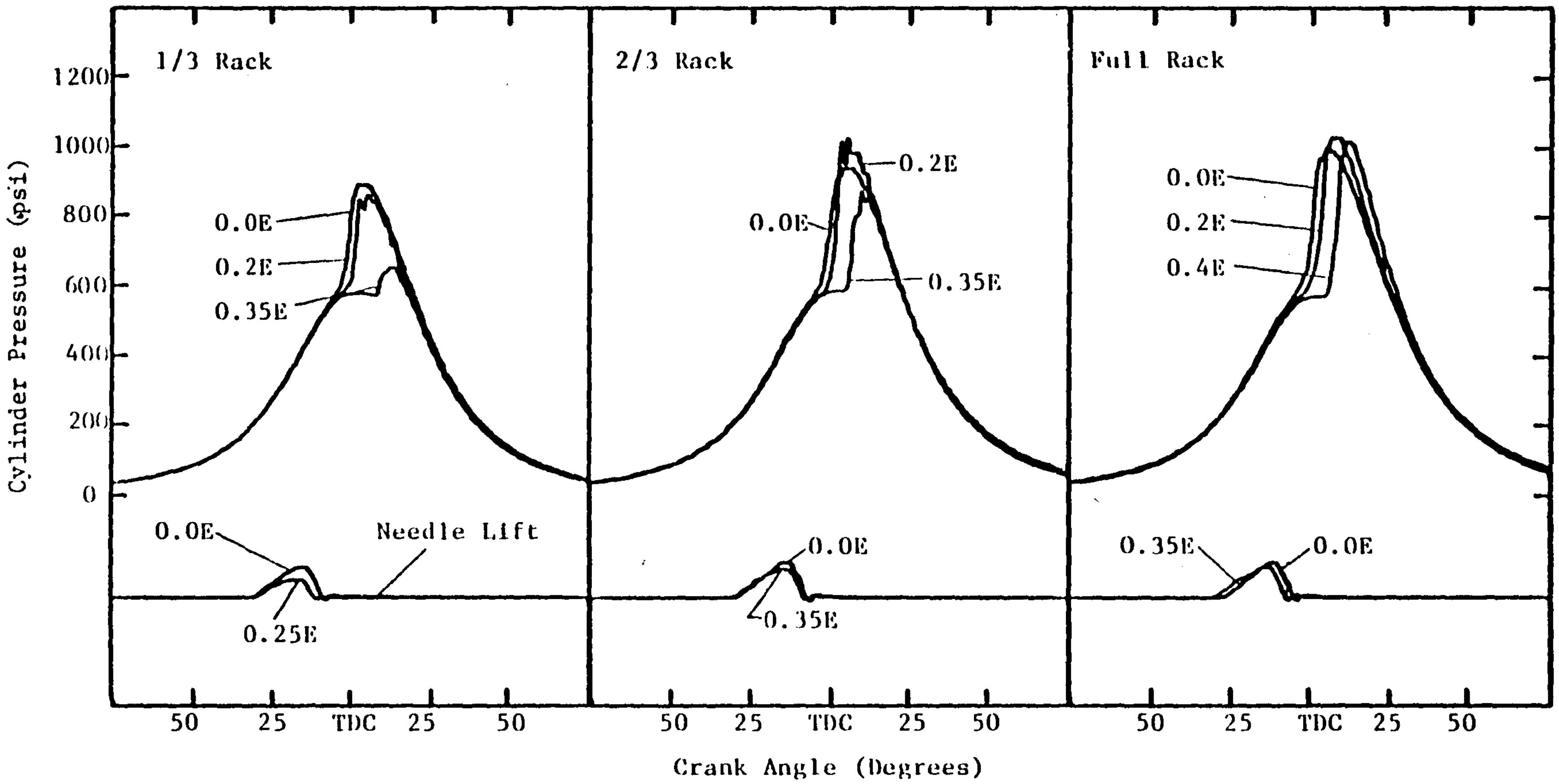

Figure 4.13 - Comparison of Characteristic Pressure and Needle Life Histories for Baseline (0.0E) and Various 140 Proof Ethanol Fumigated Conditions at 2400 RPM

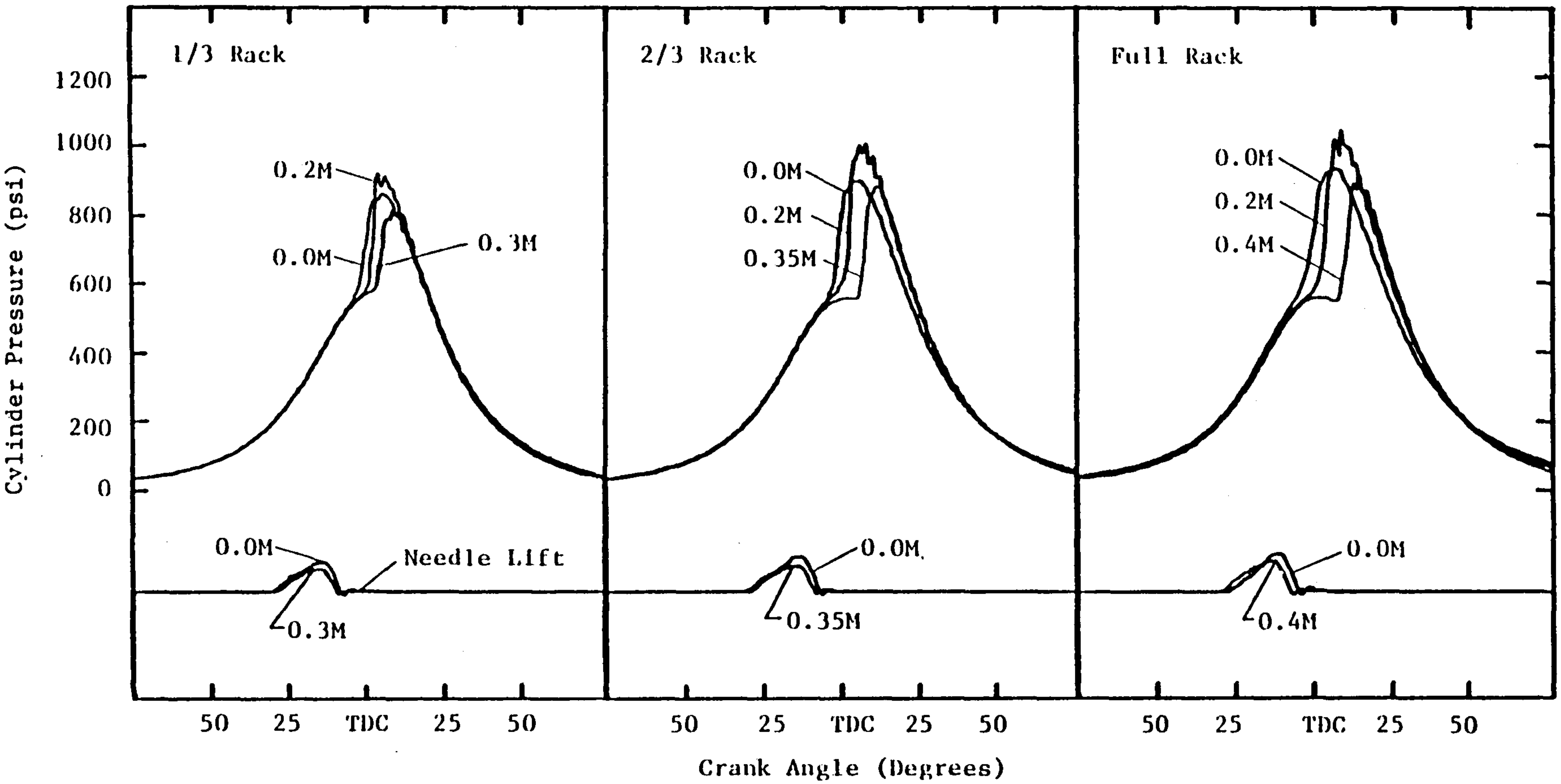

Figure 4.14 - Comparison of Characteristic Pressure and Needle Lift Histories for Baseline (0.0M) and Various 200 Proof Methanol Fumigated Conditions at 2400 RPM

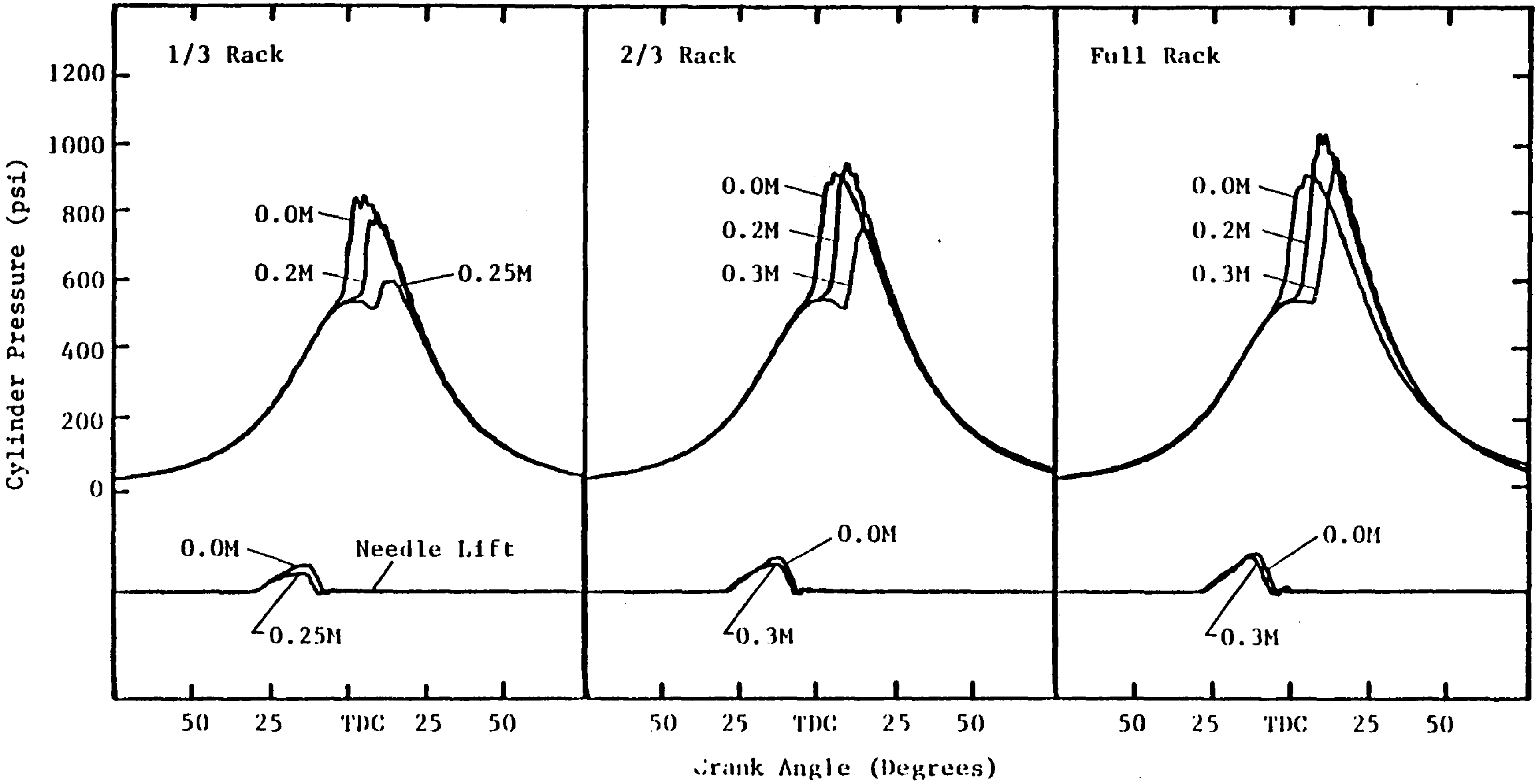

Figure 4.15 - Comparison of Characteristic Pressure and Needle Lift Histories for Baseline (0.0M) and Various 160 Proof Methanol Fumigated Conditions at 2400 RPM

4.3 Gas Phase Emissions

During all segments of the test program, exhaust gases were analyzed for CO_2, CO, O_2, NO_x and unburned HC. Of major concern in this work were gaseous emissions currently regulated under federal emissions standards: CO, HC, and NO_x. Oxides of nitrogen and unburned HC are of particular interest due to their role in the formation of photochemical smog. Carbon monoxide, although of generally low concentration in Diesel engine exhaust, is highly toxic when inhaled in even small quantities.

Emissions data are presented on an indicated power-specific mass-flow rate basis (gm/ihp-hr). This treatment of the data is necessitated by the fact that under certain operating conditions, engine parameters in terms of power and exhaust volume flow rate may change independently. The emissions parameter of interest is the mass of pollutant per unit of energy delivered.

4.3.1 Hydrocarbons

The trend in hydrocarbon emissions is shown in Figs. 4.16 and 4.17. Before discussing the data, several factors should be considered. In this work, an unheated flame ionization detector was used to measure unburned HC; error may be introduced by the effect of particulate absorption and hydrocarbon condensation in Diesel combustion studies. In addition to this problem, disparities in the results may be created by measurement error in the HC analyzer. Canton (10) reported that the presence of methanol and oxygen in the sample may slow and ultimately

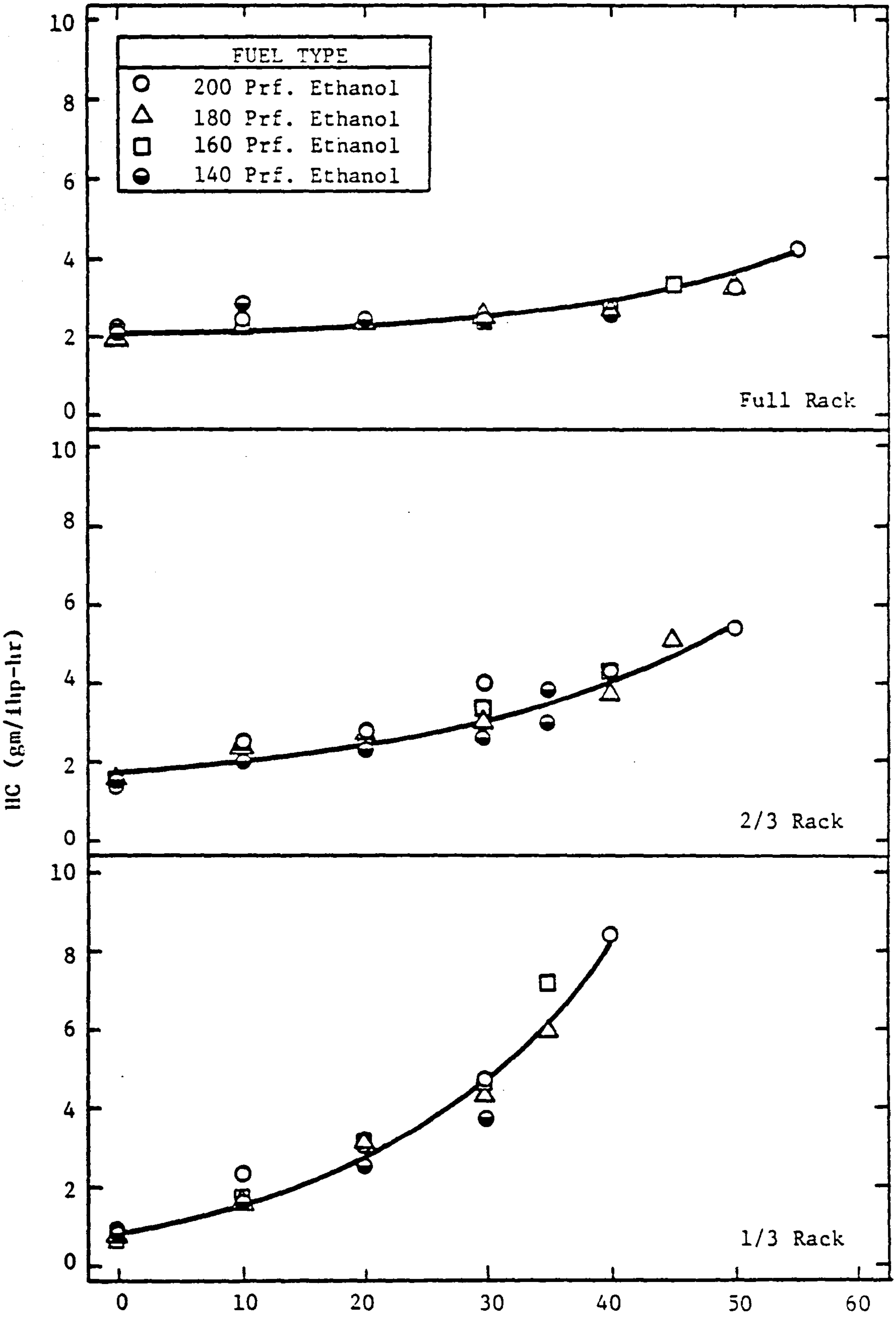

Figure 4.16 - Unburned Hydrocarbon Emission as a Function of Fumigated Ethanol at 2400 RPM

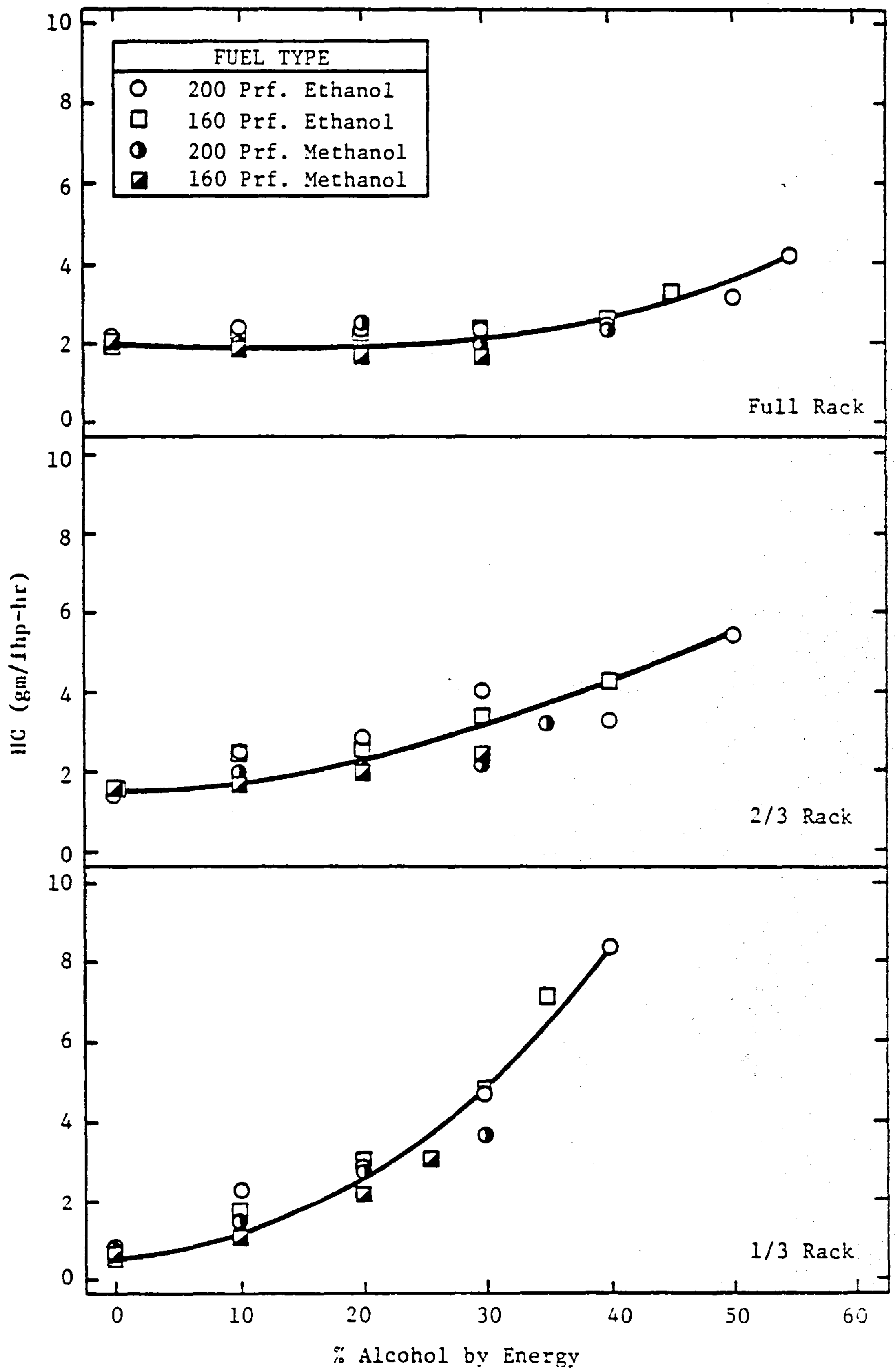

Figure 4.17 - Comparison of Unburned Hydrocarbon Emission for Ethanol and Methanol Fumigants at 2400 RPM

lower instrument response. Unburned alcohols, as well as the formation of aldehydes (not considered in this study), may also have affected HC measurement.

In view of these problems, HC data is presented only as a guideline to indicate the general trends in the emission of this pollutant under different operating conditions. The moderate data scatter is well within the experimental capabilities of the analyzer used in this work.

General trends in HC emissions are evident with regard to rack setting and percent alcohol substitution. An increase in HC emissions was observed for all three rack settings as alcohol was introduced. The relative increase in HC emissions was more significant at lower rack settings; an increase of over 8 fold at the 1/3 rack setting was measured near the misfire limit. Similar rates of HC increase were noted for other alcohols. Although percent alcohol substitution and rack setting were important factors affecting HC emission, alcohol quality had no obvious effect. This is evidenced by the narrow data point band width for most operating conditions.

4.3.2 Carbon Monoxide

The effect of alcohol fumigation on engine CO emission is shown in Figs. 4.18 and 4.19. A strong rack dependence was observed for this pollutant. At the 1/3 rack setting, a significant rise in CO emissions resulted as the percentage of alcohol was increased. This same trend was noted at the 2/3 rack condition; however, the increase in CO was less pronounced. At the full rack condition, CO emissions showed only a slight increase up to the point of 25% alcohol substitution, followed by a gentle decline to near the baseline value.

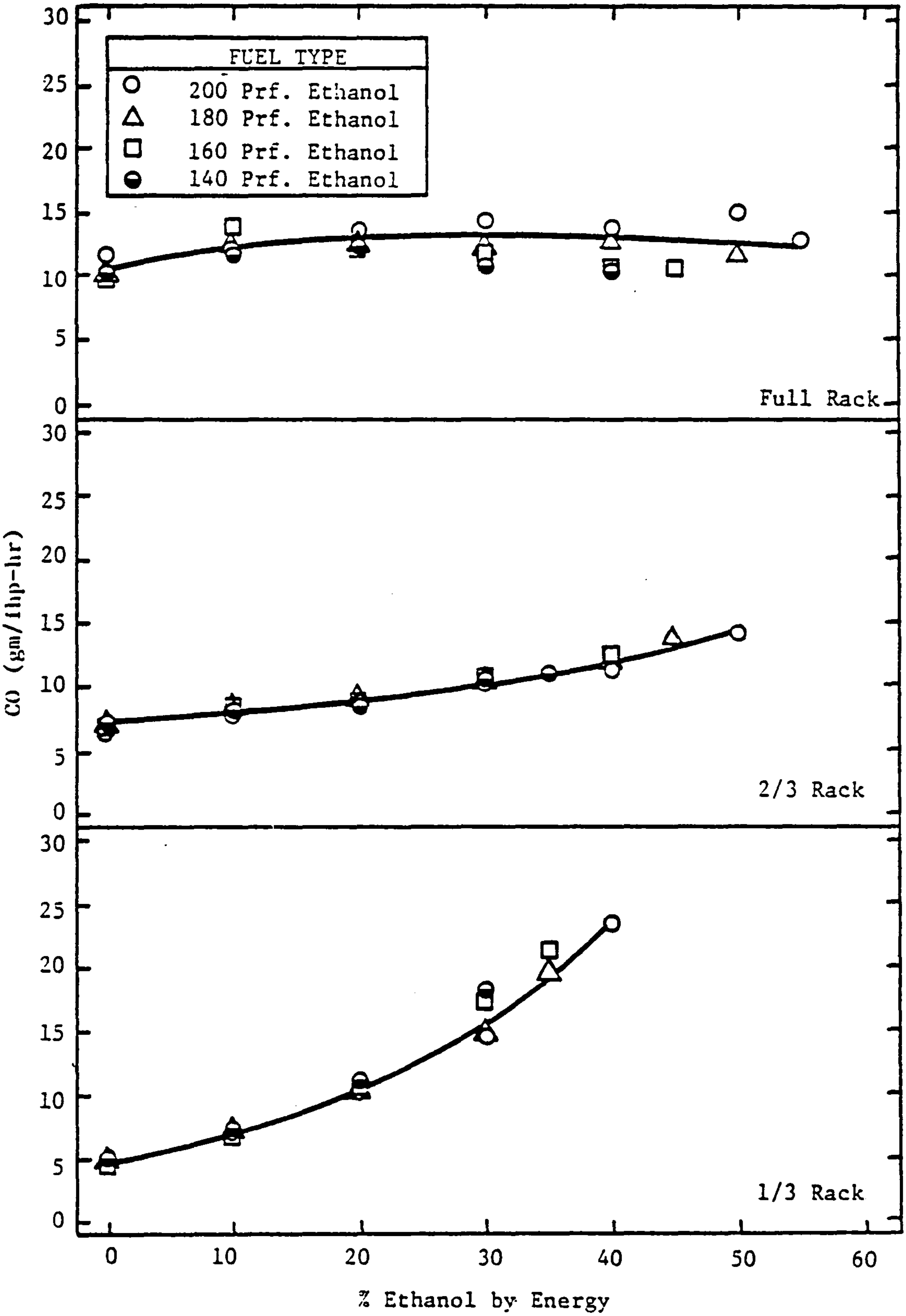

Figure 4.18 - Carbon Monoxide Emission as a Function of Fumigated Ethanol at 2400 RPM

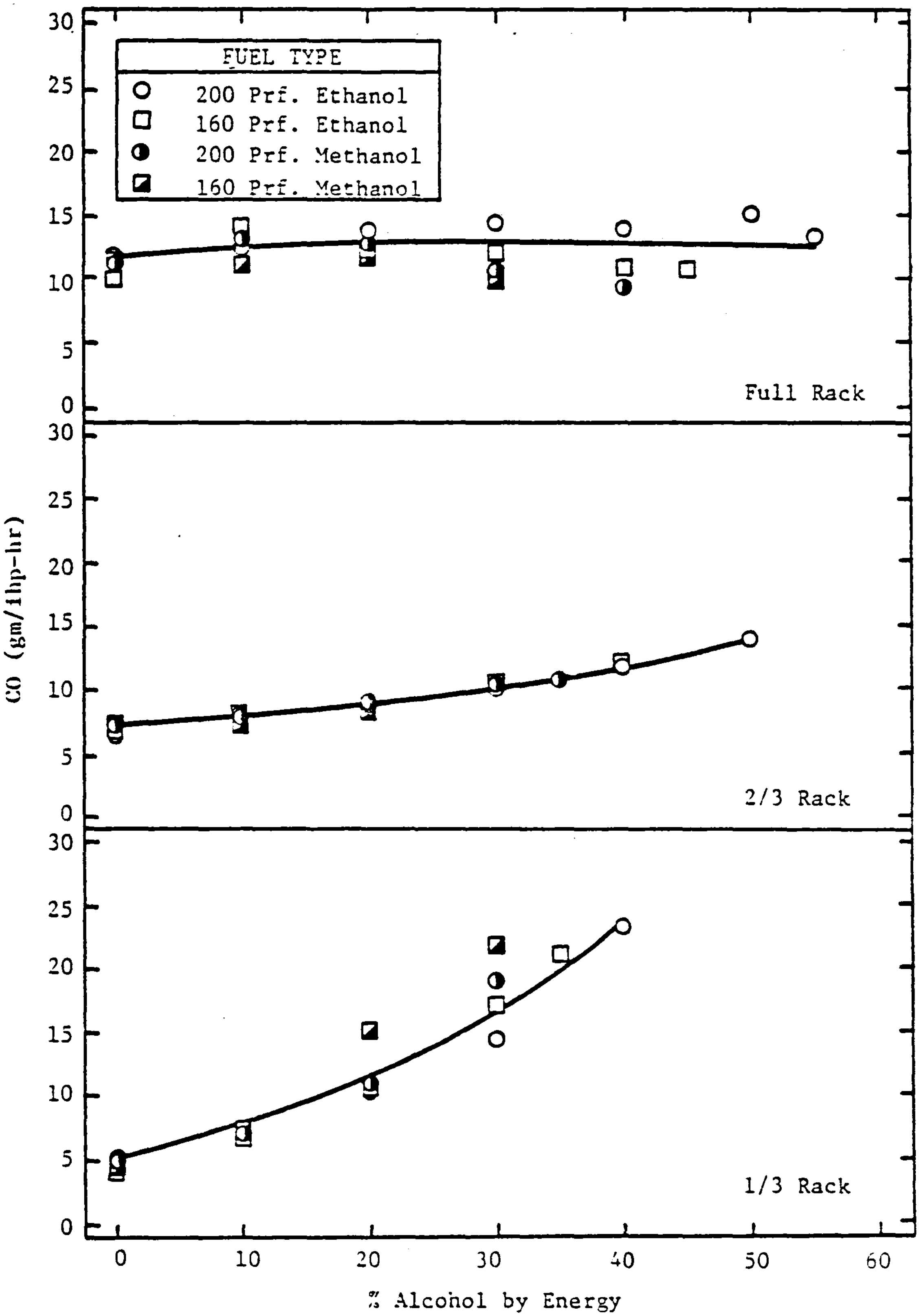

Figure 4.19 - Comparison of Carbon Monoxide Emission for Ethanol and Methanol Fumigants at 2400 RPM

The fuel type and water content did not significantly affect CO emissions. However, the data tended to scatter as the misfire limit was approached for each fuel.

4.3.3 Oxides of Nitrogen

Oxides of nitrogen (NO_x) results are presented in Figs. 4.20 and 4.21. NO_x emissions were observed to be highly dependent upon the water content of the alcohol. Figure 4.20 points out this strong effect for various proofs of ethanol. As water content increased, the exhaust concentration of NO_x was observed to decline. Wet methanol (160 proof) showed significant reductions in NO_x formation, especially as the amount of fumigated alcohol exceeded 15%.

General trends in NO_x emissions also changed with rack setting. At the 1/3 rack condition, NO_x declined as alcohol was substituted. The 2/3 rack condition showed that a slight increase followed by an equivalent decrease in NO_x resulted as increasing quantities of alcohol were fumigated. In most cases, NO_x showed moderate increases at the full rack condition; however, this again was dependent upon the water content of the fuel.

4.4 Solid Phase Emissions

Particulate emissions, currently creating widespread concern regarding their potentially harmful effect on biological systems, were collected and analyzed. Data and test results indicating the dependence of rate of formation and biological activity of these emissions on baseline and ethanol fuel are now presented.

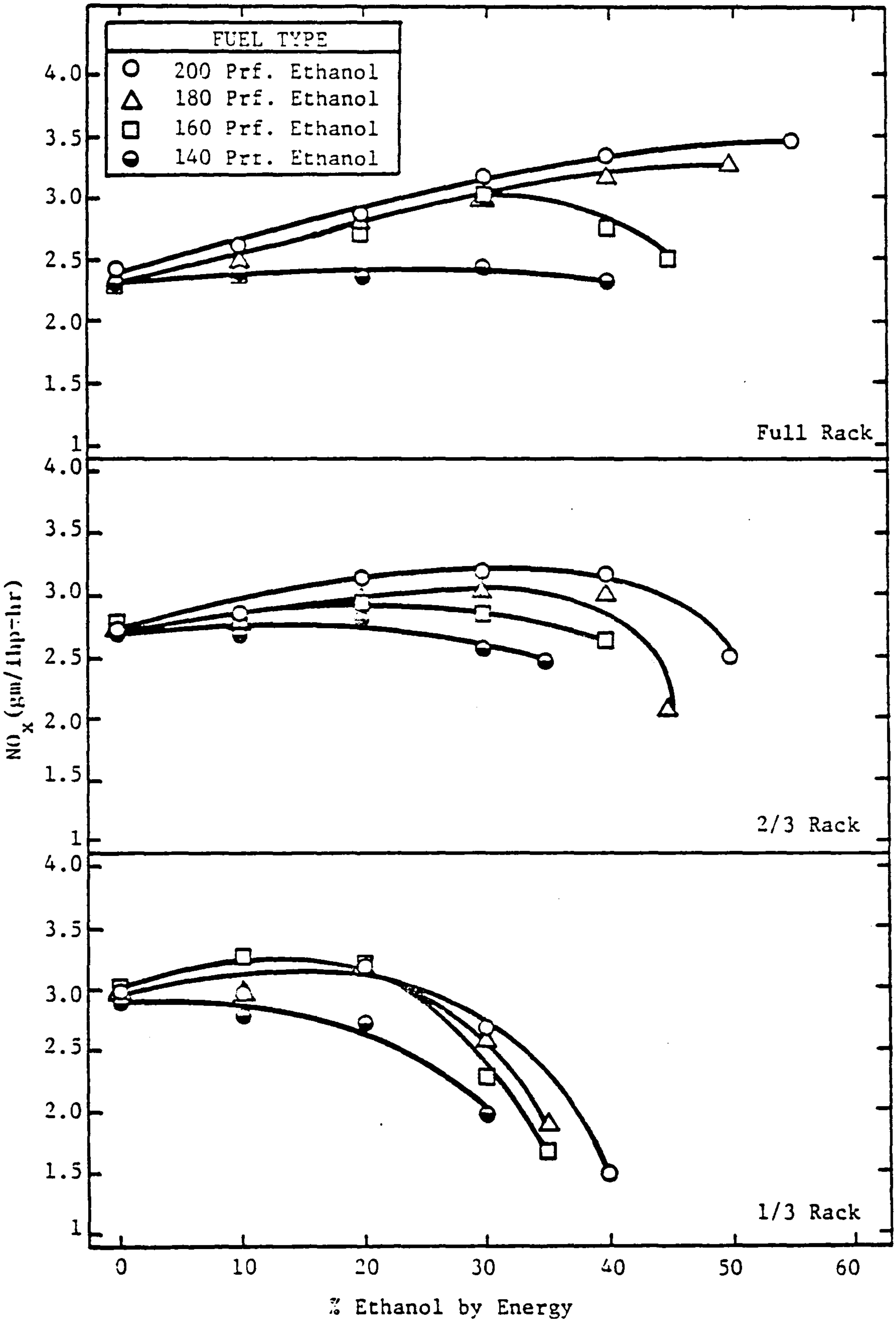

Fig. 4.20 - Oxides of Nitrogen Emission as a Function of Fumigated Ethanol at 2400 RPM

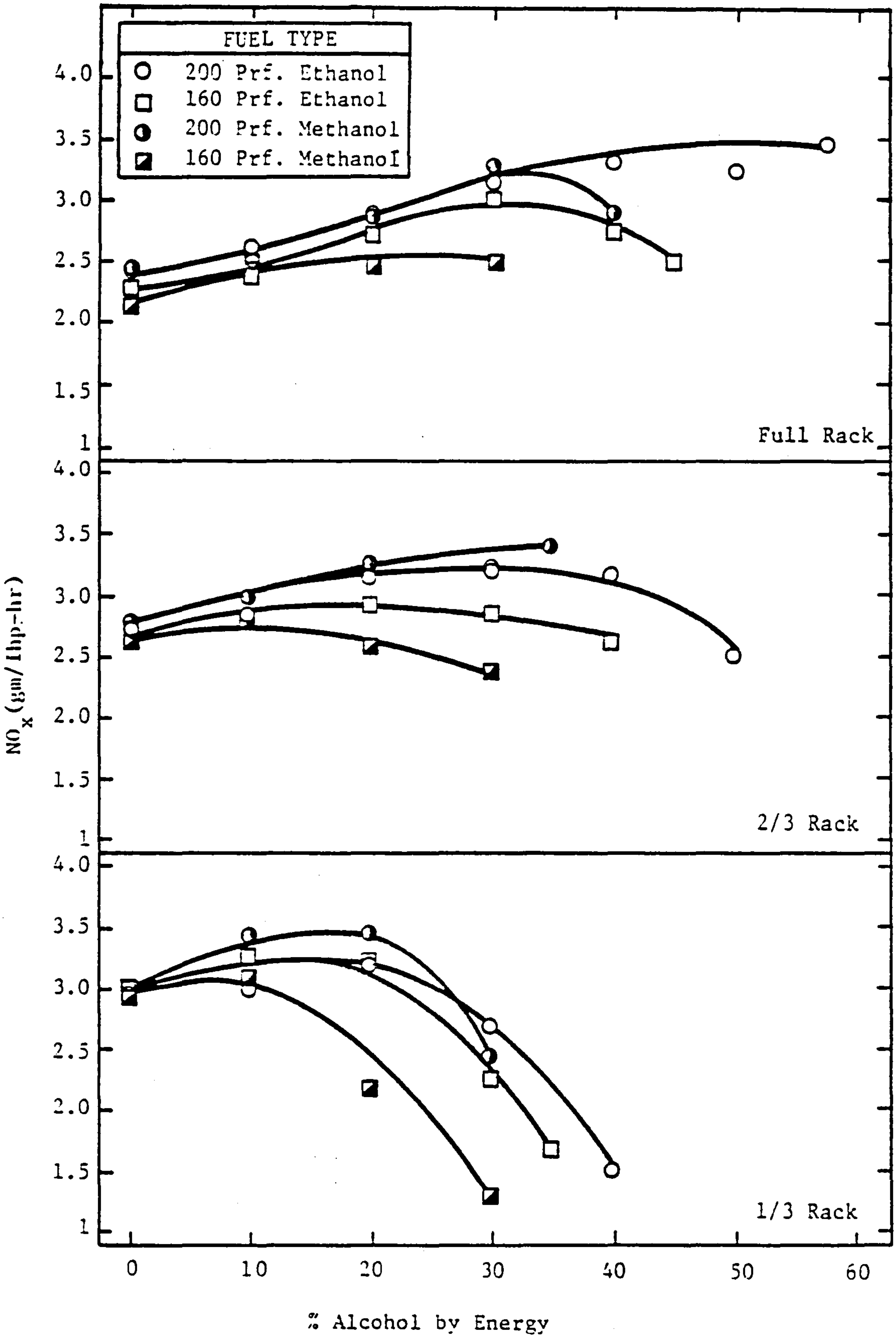

Figure 4.21 - Comparison of Oxides of Nitrogen Emission for Ethanol and Methanol Fumigants at 2400 RPM

4.4.1 Loading Rate

The mass loading rate of particulate emissions (gm/min.) was observed to decrease as alcohol replaced the baseline fuel (Table 4.2). Reductions of more than 70% of the baseline value occurred at some operating conditions.

A trend of increased particulate loading was noted with an increase in water content of the fuel. Very limited particulate analysis for methanol-fumigated conditions was performed in this work. As a result, comparison of biological enhancement and mass loading rates for ethanol and methanol is precluded.

4.4.2 Biological Assay

The biological activity of the Diesel particulate as measured by the Ames *Salmonella typhimurium* test is listed in Table 4.2. Both the raw particulate as well as the soluble organic fraction (SOF) for various racks, proofs, and percentages of ethanol were analyzed. Specific emphasis was placed upon performing a complete biological analysis of particulate emissions at the 2/3 rack condition.

Results indicate that the biological activity of the raw particulate was enhanced as ethanol was substituted for fuel oil. Activity is seen to increase in a manner proportional to percentage of ethanol fumigated. Tests performed on the SOF were somewhat less conclusive; experimental error introduced in the SOF extraction process may have been responsible for scattered results.

Table 4.3 - Summary of Particulate Data

Rack / Fuel	1/3		2/3			Full		
200 Prf. Ethanol	22027. * 0. 3.06 41.7 1.12 8.9 ± .5	22027. 20. 2.68 61.9 - 4.6 ± .4	32643. 0. 7.21 20.0 .37 6.7 ± .7	32643. 20. 4.97 33.6 1.6 19.8 ± 3.5	32643. 40. 2.0 57.0 - 6.9 ± 1.1	43317. 0. 14.62 9.9 .11 5.5 ± 1.0	43317. 20. 10.18 7.3 .17 18.1	43317. 40. 5.14 23.1 .77 18.7 ± 4.5
180 Prf. Ethanol	↓		↓	32643. 20. 5.07 28.1 - 21.7 ± 3.6		↓		
160 Prf. Ethanol	↓		↓	32643. 20. 5.69 53.9 - 10.1 ± 1.25		↓		
140 Prf. Ethanol	↓		↓	32643. 20. 5.38 43.1 - 6.9		↓		

*Data in each block is tabulated as follows:
Total fuel energy input - Btu/hr
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

Chapter 5

DISCUSSION OF RESULTS, SUMMARY AND CONCLUSIONS

5.1 Introduction

Data showing the effect of aqueous alcohol fumigation on Diesel engine performance and emissions were presented in Chapter 4. Interpretation and analysis of these experimental results, aided by the theoretical considerations presented in Chapter 2, are now reported.

Conclusions drawn from the experimental findings and recommendations for further work complete the study.

5.2 Thermal Efficiency

Thermal efficiency, a measure of an engine's effectiveness in producing shaft work, showed slight but consistent trends during alcohol fueling at various rack settings. Efficiency calculations were based upon BHP_c and the liquid fuel, lower constant pressure heating values of the Diesel and alcohol fuels.

Slight gains in thermal efficiency at the 2/3 and full rack settings with increased alcohol substitution are attributed to several factors. Increased ignition delays and large quantities of vaporized alcohol (inherent in fumigation) coupled to create rapid, nearly constant volume combustion near top dead center (TDC) - a more efficient process than typical Diesel combustion (Fig. 5.1). Peak pressures were possibly increased by the rapid heat release and by the formation of more moles of products during alcohol combustion. Rapid rates of energy release may also have reduced heat loss from the engine.

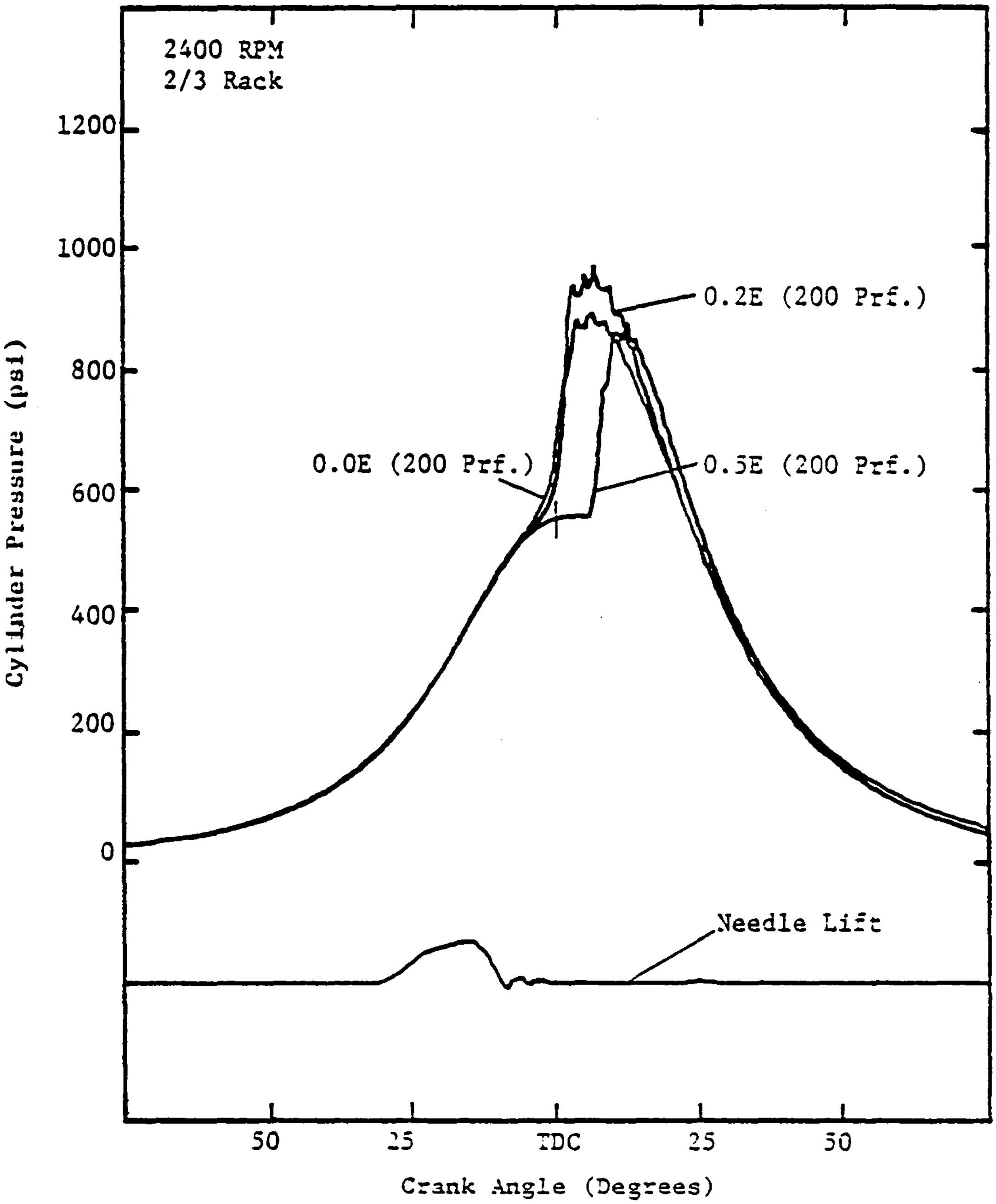

Figure 5.1 - Comparison of Pressure Histories for 200 Proof Ethanol Fumigant

Although inlet air temperatures were depressed by the high latent heat of the vaporizing alcohol, volumetric efficiency (η_v) decreased slightly. This reduction in η_v resulted from vaporized alcohol displacing air during the intake stroke; a less dense air charge existed in the cylinder, possibly deteriorating combustion. Increased CO and unburned HC emissions seem to confirm that more complete combustion was not responsible for improved thermal efficiency.

Thermal efficiency was observed to decrease at light load (1/3 rack setting). Examination of the characteristic 1/3 rack pressure traces and ignition delay data presented in Chapter 4 provides some explanation for this decrease. At light load, the lower total energy release and proportionately larger amount of fuel energy leaving the cylinder through heat transfer cause overall cooler conditions in the combustion chamber. Therefore, less energy is available to heat and vaporize the alcohol charge; the quench layer is thickened and combustion deteriorates rapidly. These conditions were evidenced by the rapid increase in CO and HC emissions for the low rack settings.

A general trend noted in the thermal efficiency data is the reduction in maximum possible alcohol substitution with lower rack setting and higher water content; misfire due to combustion quenching was enhanced by the high heats of vaporization. The relatively higher latent heat of methanol compared to ethanol created combustion conditions that were significantly different - combustion quenching occurred at a much lower alcohol substitution quantity. This same effect was also observed as the amount of water in the fumigated alcohol was increased. These trends are illustrated in Fig. 5.2.

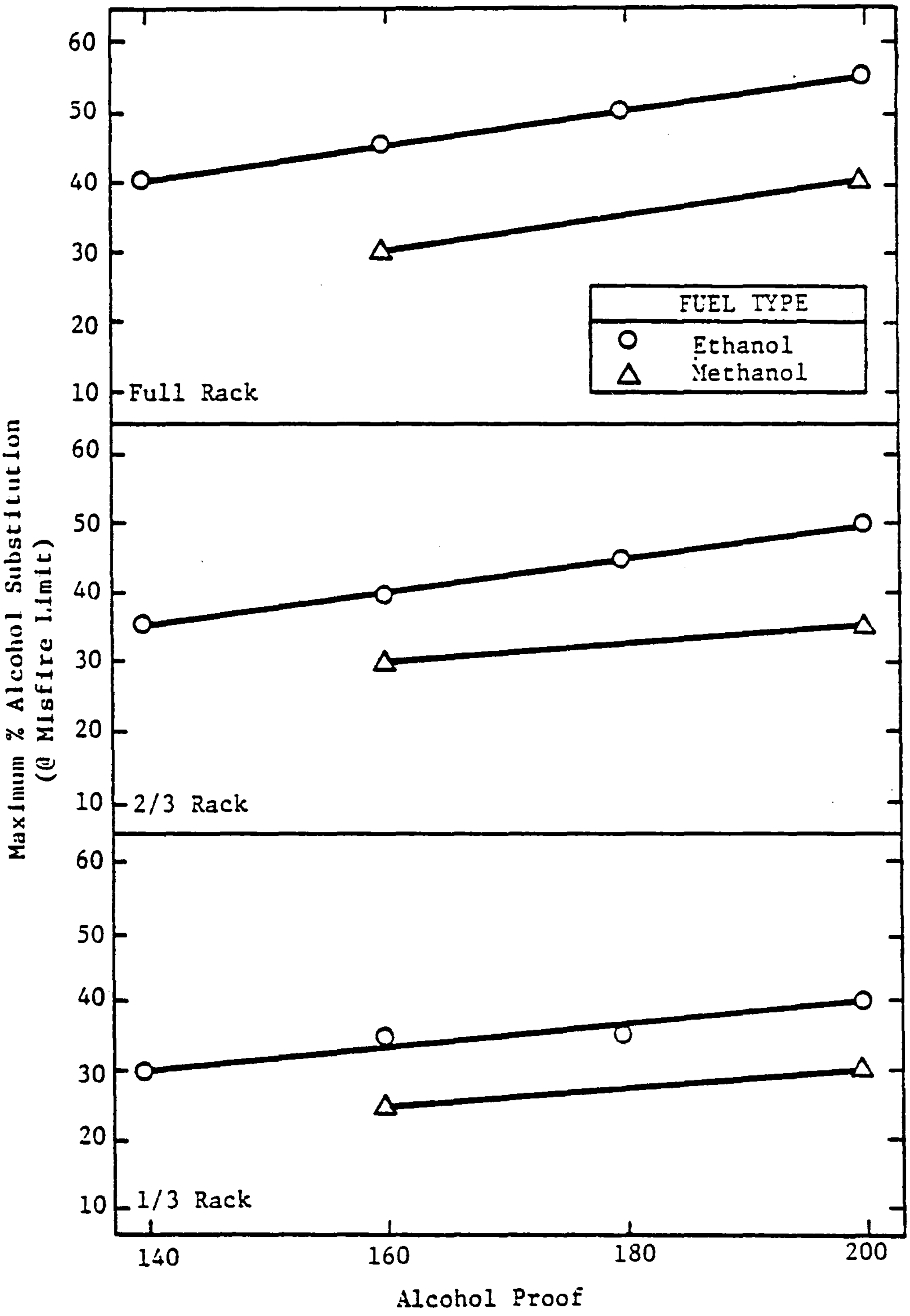

Figure 5.2 - Maximum Percent Alcohol Substitution as a Function of Alcohol Proof at 2400 RPM

5.3 Combustion Intensity

Factors affecting the severity of Diesel combustion have already been discussed (Sec. 2.2); however, before evaluating the experimental results, a brief review of these factors will be presented.

Knock, the audible manifestation of rapid rates of cylinder pressure rise, occurs early in the Diesel combustion cycle. The initial energy release during autoignition is generally responsible for creating strong pressure pulses which strike the cylinder walls. The strength of these pulses is controlled by the quantity of fuel which ignites almost simultaneously as combustion begins. Fuels exhibiting long ignition delays, such as alcohols, permit larger quantities of fuel to accumulate and vaporize preceding ignition; as a result, combustion usually begins with a large release of energy, creating engine knock. Severe knock is undesirable in Diesel engine combustion due to the mechanical stresses created by the associated rapid pressure rise and high peak pressure.

The peak pressure and rate of pressure rise data presented in Figs. 4.6-4.9 show that increases in both of these combustion parameters occurred during initial alcohol substitution. Aural and quantitative measurement of knock (see Appendix B) confirmed that combustion intensity increased at these alcohol-fueled conditions.

Two characteristics of alcohol fumigation are responsible for these observed increases in combustion intensity:

1) increased ignition delay resulting from the charge cooling of the vaporizing alcohol, and

2) the presence of a vaporized, homogeneous alcohol fuel charge which ignites immediately as combustion starts.

The effect of both of these factors can be noted in Fig. 5.1. Constant volume combustion near TDC occurred as high flame speeds enhanced combustion in the alcohol fuel charge. Correspondingly higher rates of pressure rise and peak pressure resulted.

However, peak pressure and rates of pressure rise declined below baseline values as the misfire limit was approached; a significant reduction in combustion noise accompanied these events. Autoignition delayed until well after TDC was responsible for the observed reduction of combustion severity. Combustion of this type is also shown in Fig. 5.1. The effect of water in the alcohol fuel was most evidenced by its extension of ignition delay - a result of increased charge cooling. Combustion quenching posed the limiting constraint on maximum alcohol substitution as autoignition occurred progressively later in the expansion stroke. Figure 5.3 illustrates more clearly the extended ignition delay which occurred during combustion of lower quality alcohols. The increased ignition delay experienced during methanol compared to ethanol fumigation is seen in Fig. 5.4.

5.4 Emissions

Both gaseous and solid emissions were collected and analyzed during engine operation. Equivalence ratio $[(F/A)_{act.}|(F/A)_{stoich.}]$ is known to have a significant effect on particulate and gaseous emissions production. The measured A/F ratio for anhydrous methanol and ethanol (Fig. 5.5) was observed to decrease as alcohol substitution increased. Similar trends were noted for the lower proof alcohols. The lower

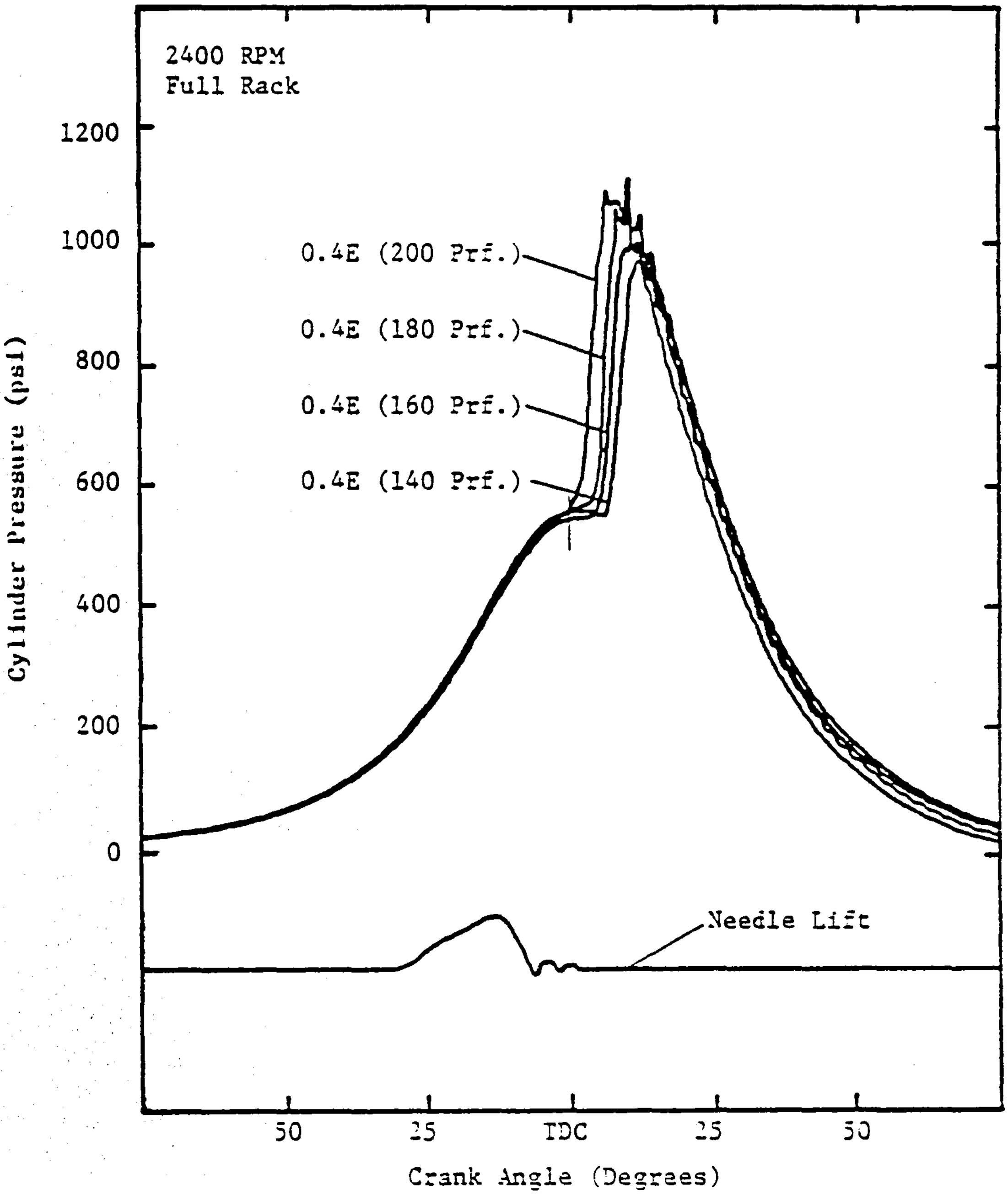

Figure 5.3 - Comparison of Pressure Histories for
Various Proof Ethanol Fumigants

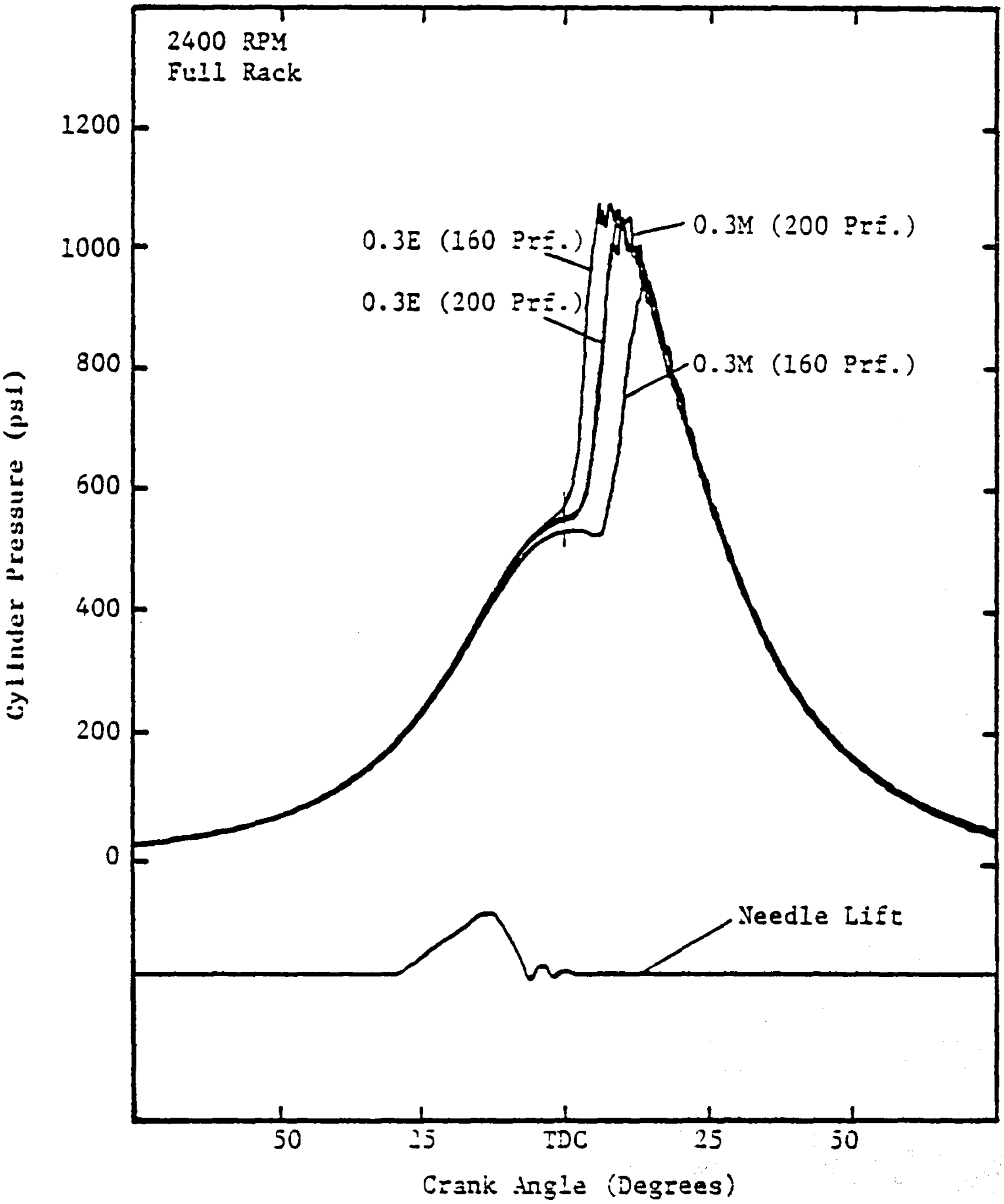

Figure 5.4 - Comparison of Pressure Histories for Various Proof Ethanol and Methanol Fumigants

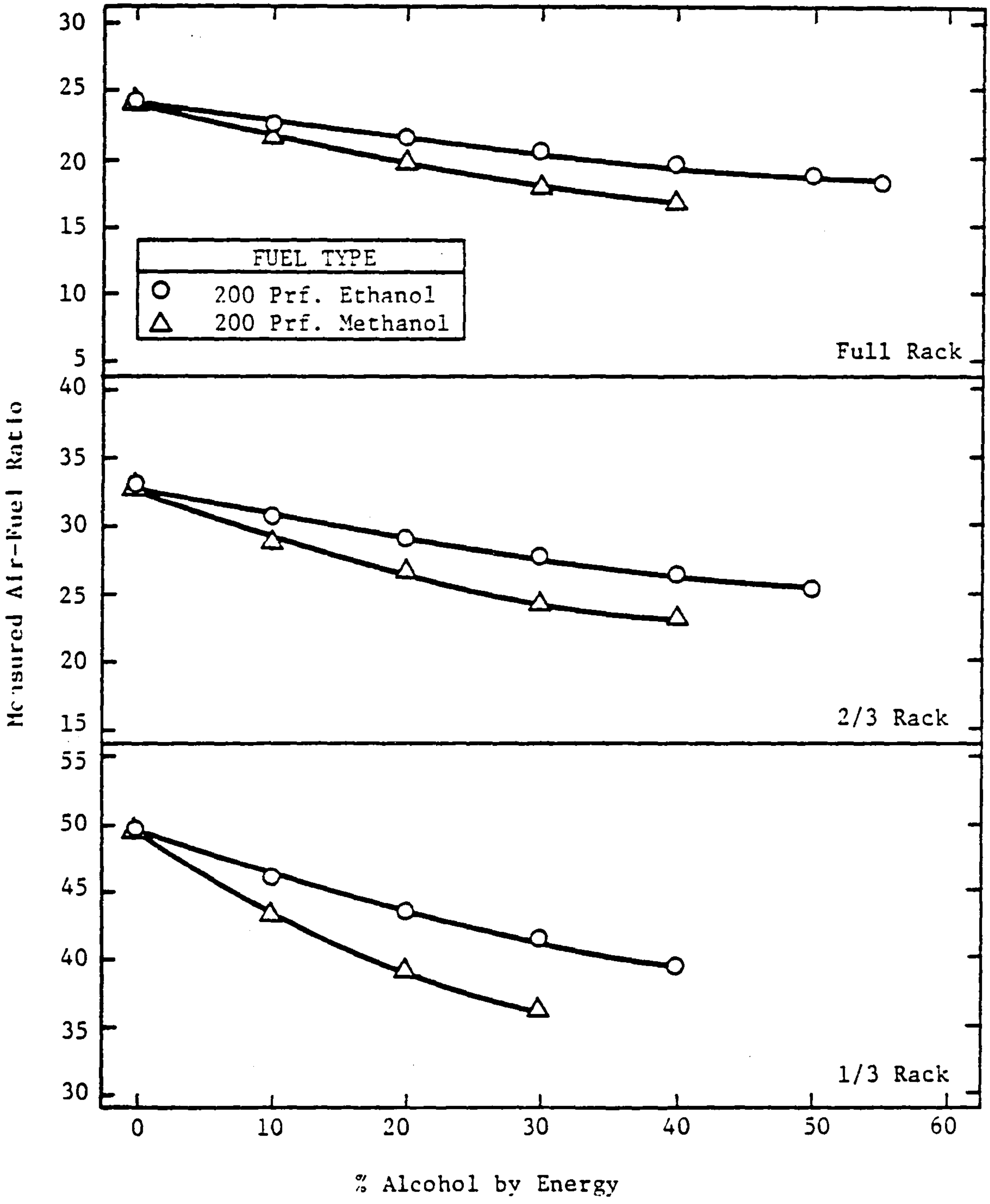

Figure 5.5 - Comparison of Measured Air-Fuel Ratios for Ethanol and Methanol Fumigants at 2400 RPM

heating value of the alcohol necessitated a larger mass substitution of fuel, accounting for the observed reduction. However, the stoichiometric A/F ratio also decreased due to bound oxygen and the higher hydrogen to carbon ratio of the alcohol molecule. As a result, the equivalence ratio remained nearly constant (Fig. 5.6).

From this brief analysis, it appears that factors other than mixture composition were responsible for the changes in pollutant formation. Since diffusive-type combustion of the fuel oil as well as homogeneous alcohol combustion existed, emission characteristics of both Diesel and SI engines would be predicted. A more thorough analysis of the factors responsible for emissions changes is now presented.

5.4.1 Gaseous Emissions

Federal standards currently regulate the emission of three harmful gaseous pollutants: CO, NO_x, and unburned HC. Fumigated alcohol significantly altered the levels of each of these pollutants produced under varied operating conditions.

Carbon monoxide emissions increased with alcohol substitution at the 1/3 and 2/3 rack settings, but remained fairly constant at the full rack operating condition. An obvious rack (load) dependency is indicated by the data.

Carbon monoxide is produced as a result of incomplete oxidation of the fuel. Factors causing combustion deteorioration (such as high latent heats of vaporization) could be responsible for increased CO production. The rack dependency of this emission indicates that combustion chamber temperatures may have had a significant effect. A

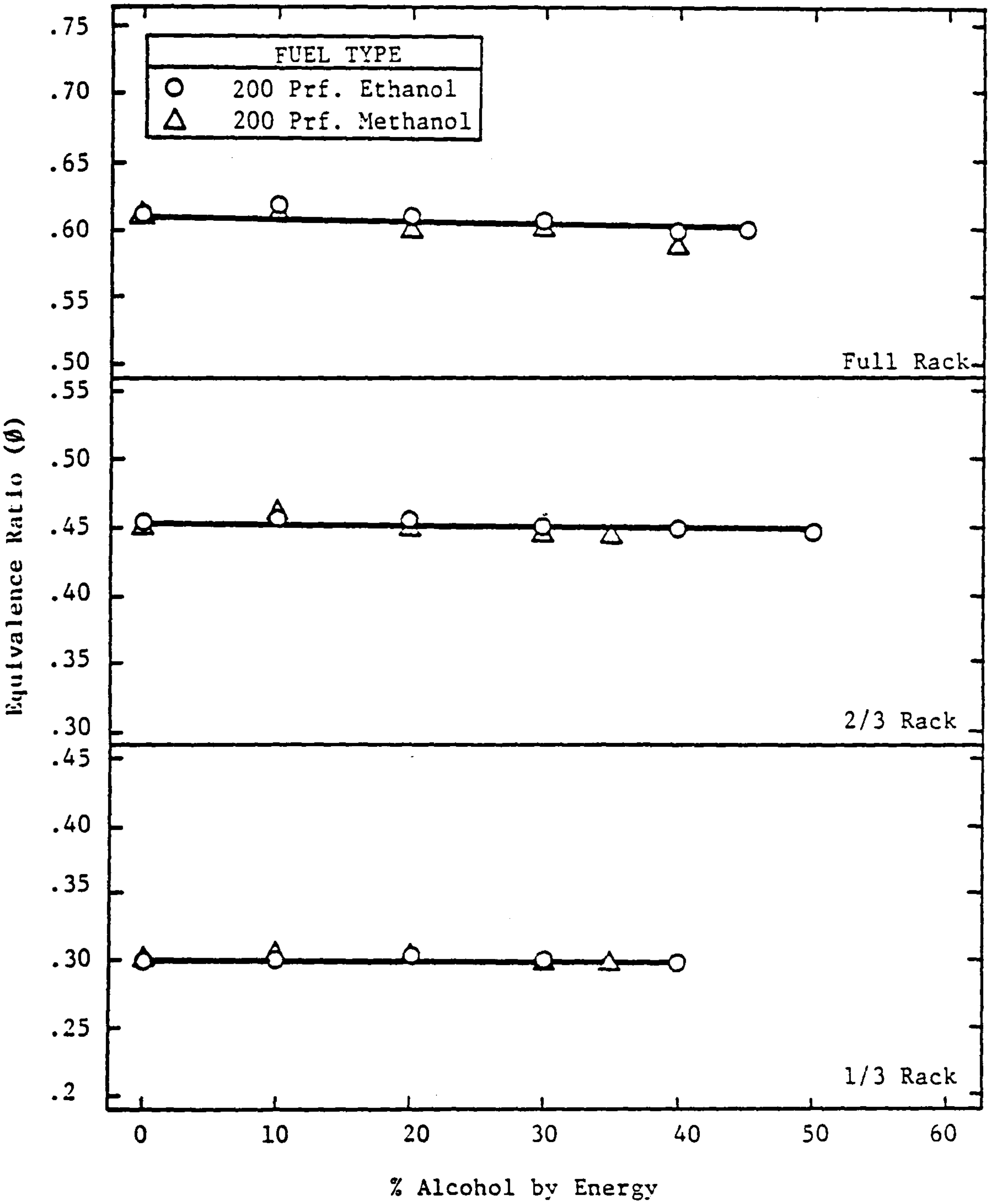

Figure 5.6 - Equivalence Ratio for Ethanol and Methanol Fumigants at 2400 RPM

thickened quench layer created by the cooling effect of vaporizing alcohol could have played a major role in the increased CO production.

As rack setting (and combustion temperature) increased, better air utilization due to the presence of a homogeneous alcohol charge may have lowered CO emissions. This effect, combined with higher combustion temperatures, would tend to minimize the increase in CO emission. At the full rack setting, CO emissions remained constant or decreased slightly; improved air utilization and a smaller quench effect apparently dominated at this condition.

In general, at the lower rack settings CO_2 emissions decreased while O_2 levels increased with greater alcohol substitution. These trends also point to combustion deterioration - supporting the proposed explanation for increased CO formation.

Incomplete combustion, the mechanism responsible for increased CO production, generally has a similar effect on unburned hydrocarbon emission. Cylinder wall quenching of the homogeneous charge in SI engines is known to cause increased exhaust emission of unburned HC. At least one investigator has shown that mechanisms responsible for increased HC emissions in SI engines are similar to those in fumigated Diesel engines (49).

Other factors may have been responsible for the rapid increase in HC emissions as the misfire limit was approached. Here, combustion began late in the cycle due to extended ignition delays (Figs. 5.3 and 5.4). Bulk quenching of the fuel oil and alcohol charges may have resulted as pressures dropped rapidly during the expansion stroke.

Distinct differences in HC and CO emissions were not observed for ethanol and methanol; water content also did not appear to affect the emission levels of these pollutants. It was expected that combustion would be deteriorated due to the presence of water vapor in the combustion chamber and create higher unburned HC emission levels.

Operating conditions which provide high local temperatures in oxygen-rich areas generally promote large production of oxides of nitrogen. Decreases in NO_x levels at the 1/3 rack setting were probably a result of peak temperature reduction due to the cooling effect of the vaporizing alcohol and the ensuing combustion deterioration.

Increases in NO_x production at the 2/3 and full rack settings are less easily explained. Rapid burning of the homogeneous alcohol charge at the beginning of combustion resulted in increased peak pressures for most of these load conditions. Locally high peak cylinder temperatures accompanied these rapid combustion events. Nitric oxide (NO) formation is a function of the local temperature, local oxygen concentration, and the time available for reaction. These dependencies suggest that increased NO production may have occurred in lean alcohol fuel areas during the initial stages of combustion.

Levels of NO_x production also showed a dependence on alcohol type and quality. The relative difference in latent heats of vaporization of methanol and ethanol, and their effect on the degree of charge cooling probably caused these differences. Similarly, increased water content of the alcohol should have depressed peak temperatures, explaining the relatively lower NO_x emission levels for the low proof alcohols.

5.4.2 Solid Phase Emissions

Decreases in particulate loading rates occurred with increased ethanol substitution. This was a direct result of the sootless flame, characteristic of alcohols. The formation of a homogeneous ethanol charge may also have been responsible for leaner combustion of the injected fuel oil.

The limited biological analysis performed in this work indicated that the activity of the raw particulate and its soluble organic extract appear to have been enhanced by ethanol substitution (Table 4.2). Determination of the exact mechanisms responsible for this increased biological activity is outside the scope of this study.

5.5 Summary and Conclusions

The increasing unavailability of petroleum-based fuels necessitates the development of alternative energy sources. Alcohols represent one of the most attractive near-term alternative fuels. Diesel engine combustion of this fuel is an important consideration in light of the increasing application of this power plant in the light-duty automotive sector.

In the past, most alcohol-in-Diesel engine feasibility studies have examined the combustion of anhydrous fuels. Physical and economical implications may make small-scale distillation of these high-quality alcohols impractical.

This study was undertaken to determine the effects of aqueous alcohol fumigation on Diesel engine combustion and emissions. The following conclusions, in accordance with the specific objectives of this study, have been drawn from the experimental results:

1) Slight thermal efficiency improvements, resulting primarily from constant volume combustion of the homogeneous alcohol charge, are possible with limited alcohol fumigation (up to 30% by energy) at the 2/3 and full rack operating conditions.

2) Alcohol type and quality (as low as 140 proof) have an insignificant effect on thermal efficiency up to the point of engine misfire.

3) Rapid combustion of the alcohol charge, resulting in unacceptably high rates of pressure rise (especially at the higher load conditions), may limit the percent of alcohol substitution to avoid engine damage over prolonged use.

4) Fumigation of alcohol produces increased ignition delays; higher water content of the alcohol lengthens this delay period.

5) Maximum possible alcohol substitution is set by engine misfire, a consequence of the alcohol and water vaporization charge cooling effect.

6) Increased carbon monoxide and unburned hydrocarbon formation resulted during alcohol fumigation, but showed a strong rack dependence. Water content had no significant effect on these emissions.

7) Relative levels of NO_x emissions decreased with higher alcohol water content for all load conditions.

8) Particulate emissions are reduced by ethanol fumigation. Limited biological analysis of this particulate indicates that

ethanol fumigation may increase the biological activity, as measured by the Ames test, of the raw particulate and its soluble organic extract.

5.6 Recommendations for Future Work

This study is part of a continuing program to evaluate the utilization of alternative fuels in Diesel engines. To date, results indicate that limited quantities of aqueous alcohol can be used efficiently while avoiding Diesel knock. However, enhanced biological activity of the resultant particulate emissions may discourage alcohol usage. Further work is justified in examining the effects of optimized injection timing on improving combustion efficiency and reducing engine knock. A method of determining and controlling the degree of vaporization of the alcohol charge preceding ignition, may also contribute toward controlling combustion severity; exhaust waste heat reclamation may be applicable in this regard. Direct engine-to-microprocessor interfacing would allow a more thorough investigation of peak pressure and rate of pressure rise characteristics. The significance of the biological findings cannot be understated; further investigation of emissions related work is paramount. In addition to the pollutants already considered, gas chromatography may prove useful in measuring aldehydes, not considered in this work.

APPENDIX A

The Salmonella/Ames Test (33)

"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 tester 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, and 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

Knock Quantification System

Knock is generally undesirable in engine combustion due to the damaging mechanical stresses resulting from associated high peak pressures and rates of pressure change (dp/dt). In this study, knock quantification was used as a means of comparing combustion intensity as different types and quantities of alcohol were fumigated.

Hoffman (68) and Duke (69) concluded that analysis of knock in SI engines is best conducted by examination of the filtered dp/dt signal. The reader is referred to their work for a more complete explanation. Knock in Diesel combustion is less easily detected and, consequently, is more difficult to quantify. As a result, qualitative description of Diesel knock based on aural detection has generally been used.

In an attempt to quantify knock in this study, filtered pressure as well as filtered dp/dt signals were examined. Initial testing revealed that knock intensity measurement based upon rates of pressure change correlated best with audible manifestations of harsh combustion in the cylinder. Therefore, it was decided to use a filtered dp/dt signal for quantifying knock.

A quartz pressure transducer (Kistler Model 6031) communicated with the combustion chamber through a small connecting passageway in the cylinder head. By shunting the signal (current) from this transducer to ground through a high impedance resistor, a small voltage proportional to the rate of pressure change in the cylinder could be measured.

The dp/dt signal was amplified and passed through a KROHN-HITE Model 3700 band-pass filter which was set to attenuate frequencies outside a 2000-8000 Hz bandwidth. Filtering effectively removed pressure changes associated with the normal compression and expansion strokes of the engine; electrical noise resulting from mechanical vibrations was also eliminated. The filtered dp/dt signal represented pressure fluctuations which occurred within the engine as a result of autoigniting fuel-air pockets. This filtered signal provided the basis for knock quantification.

The filtered dp/dt signal was input to a comparator that detected voltages (in this case proportional to rate of pressure rise) which exceeded a selected threshold level. The comparator output stepped to a high state when the threshold level was exceeded and triggered a monostabilizer or "one-shot". The monostabilizer delivered a square wave pulse of selected duration (15 msec) to a digital frequency counter (Beckman Model 6380 counting unit, Model 678 function unit, and a Model 683 input module). The purpose of the monostabilizer was to deliver only one pulse to the frequency counter per severe combustion cycle. By connecting the needle lift signal (see Section 3.2.2) to the time base of the frequency counter, the fraction of combustion cycles in which the filtered dp/dt signal exceeded the selected threshold voltage was determined. This value was displayed on the digital output of the frequency counter. An electrical schematic and a characteristic signal display pattern of the knock quantification system are presented in Figs. B.1 and B.2, respectively.

The threshold level of the comparator was set such that distinct, audible knock which occurred during certain operating conditions was

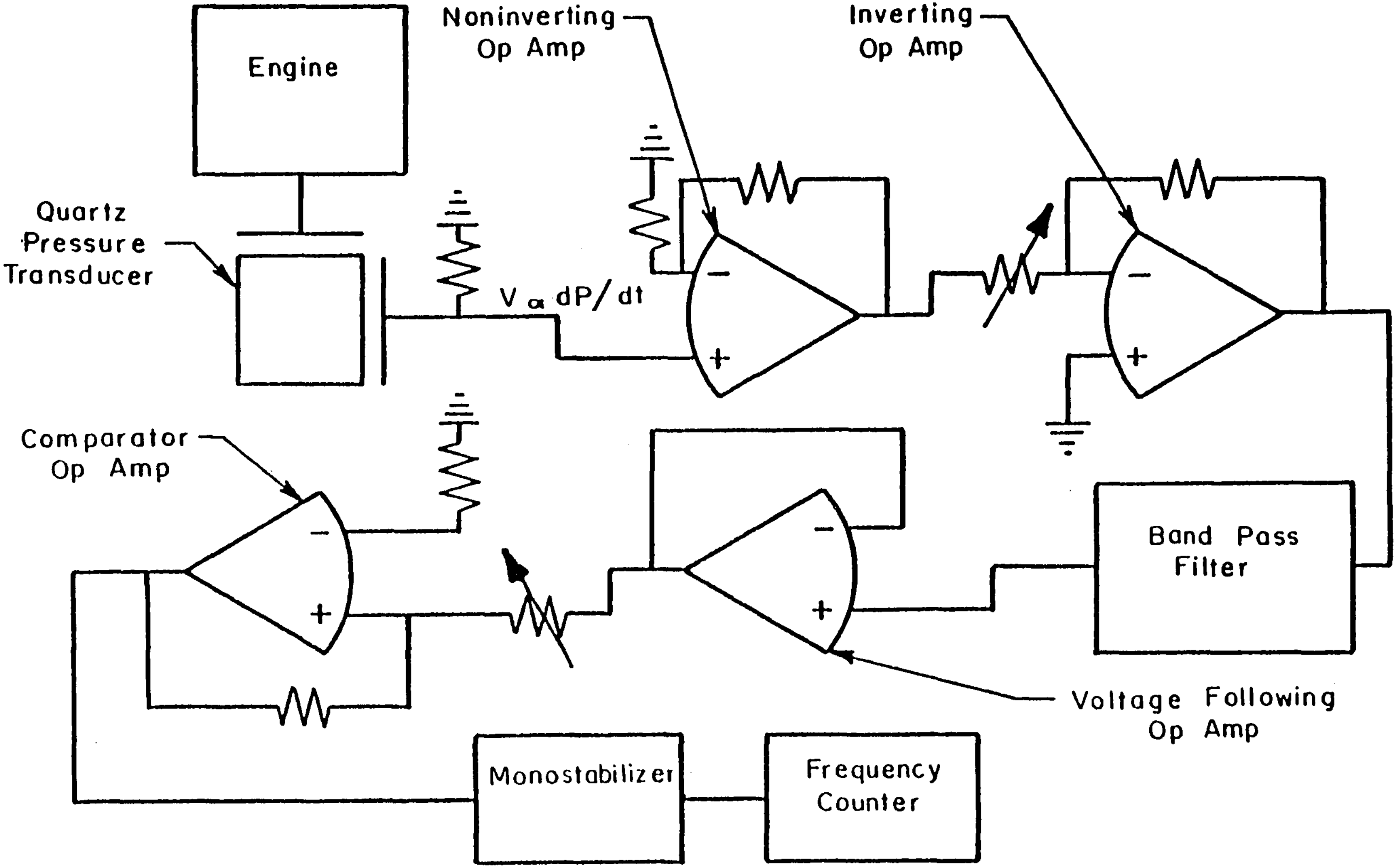

Figure B.1 - Knock Quantification System Electrical Schematic

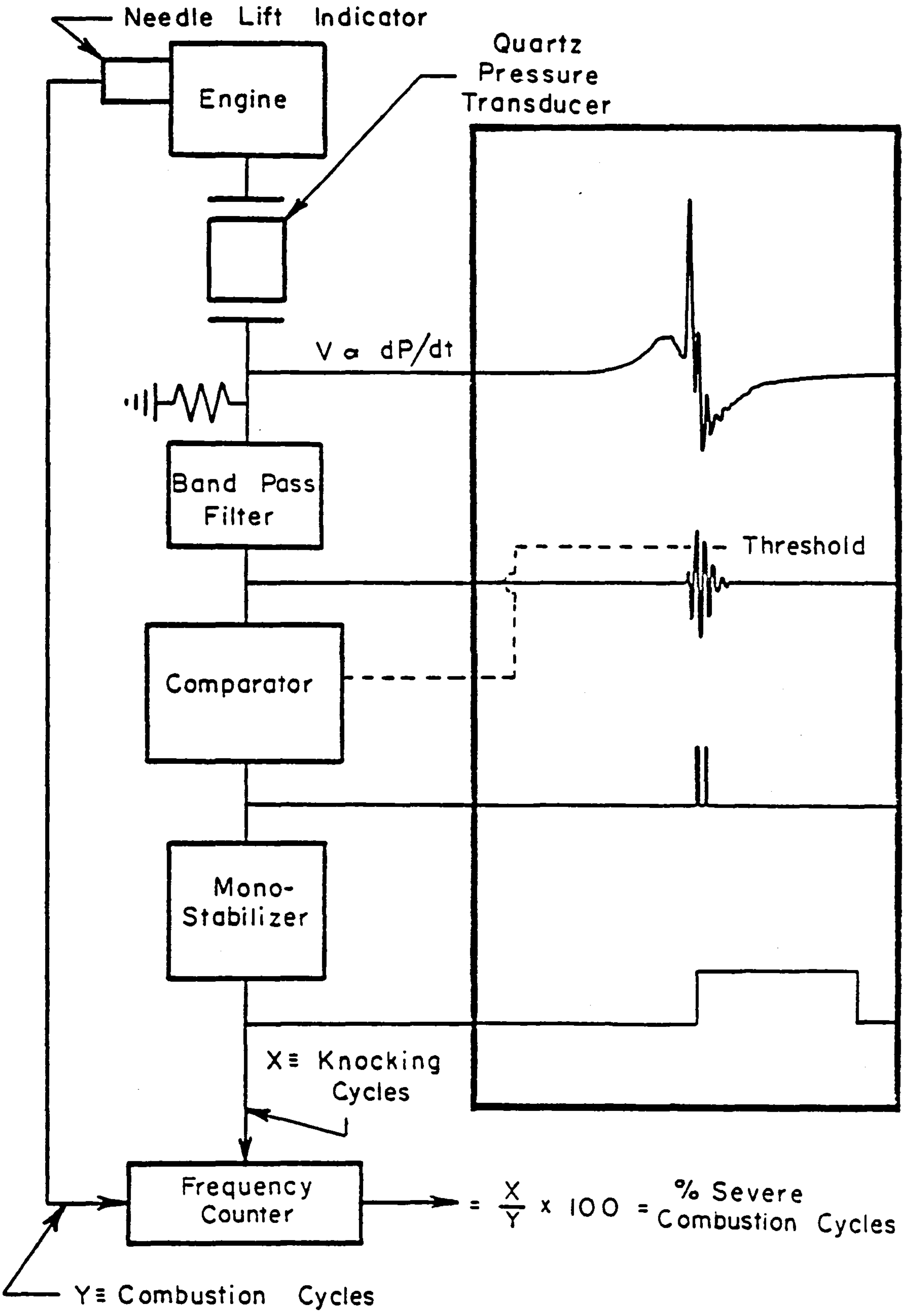

Figure B.2 - Knock Quantification System - Characteristic Signal Display detected. By displaying a pressure trace (from a second transducer) on a digital oscilliscope and measuring the maximum slope, it was determined that strong, audible knock generally resulted when dp/dt values exceeded 100 psi/degree of crank angle. Capacitive and inductive effects of the circuitry as well as the fact that a filtered rather than absolute dp/dt signal was being evaluated deterred more definitive determination of exact dp/dt threshold levels. Nevertheless, changes in combustion intensity resulting during various operating conditions were detected.

The knock quantification system was used in a portion of the alcohol fumigation work. The experimental results are shown in Fig. B.3. Trends in the knock intensity data correlate with trends observed in the dp/dt and peak pressure data presented in Section 4.2.3. Knock generally increased as alcohol was fumigated; however, as the misfire limit was approached the percentage of knocking combustion cycles decreased.

Although certain trends are evident, the data is widely scattered for different proofs of alcohol fuels; this degree of scatter was not evident in other pressure phenomena data. Additional work must be completed to explain these results. Further analysis may determine if instrument malfunction or engine cycle-by-cycle variations skewed the experimental results.

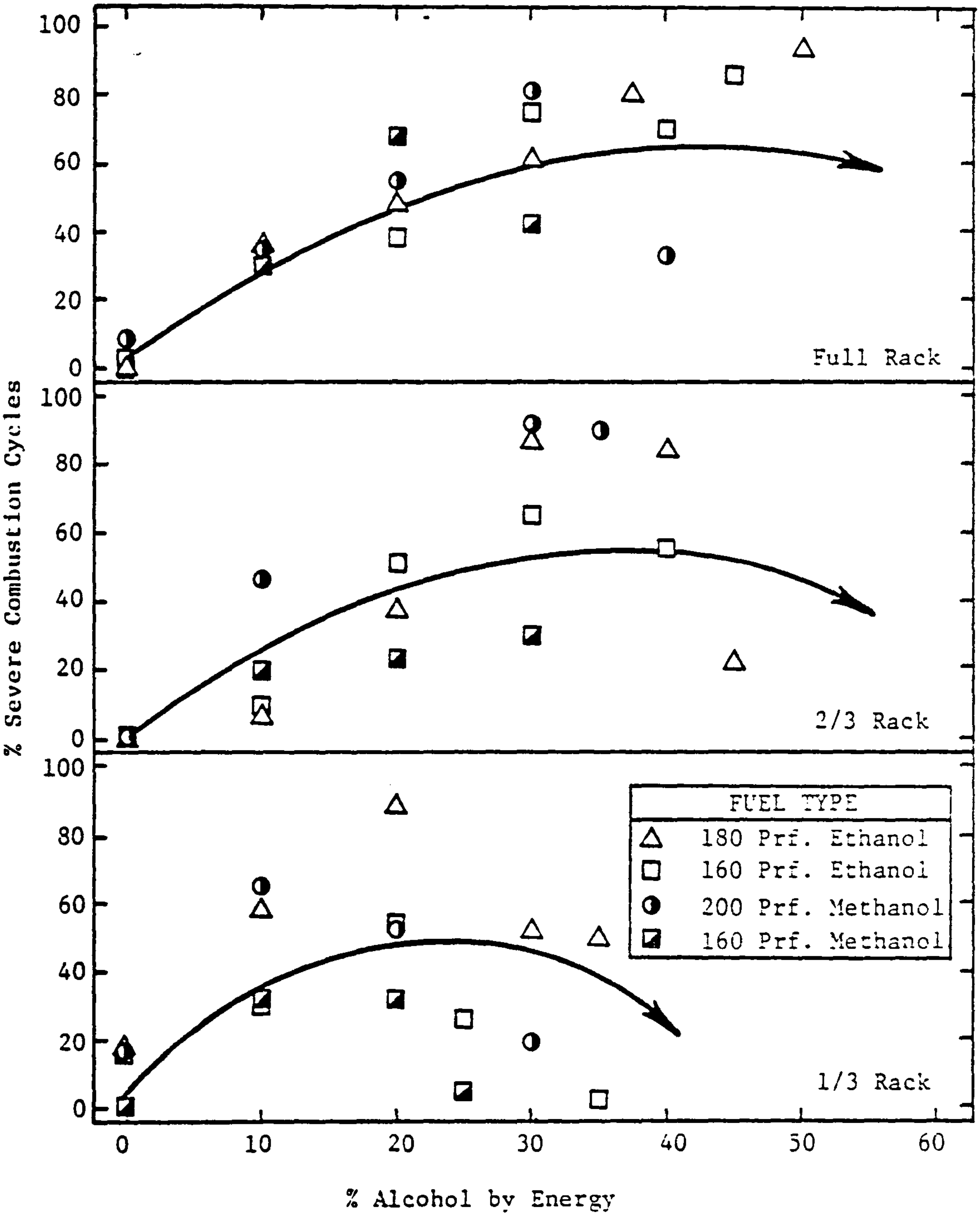

Figure B.3 - Severe Combustion Cycles Measured by the Knock Quantification System as a Function of Fumigated Ethanol and Methanol

APPENDIX C

Reduced Experimental Data

Key to Data

%ALCH	percent of total fuel energy supplied by fumigated alcohol
THEFF	engine thermal efficiency (%)
BHP_c	brake horsepower corrected to standard test conditions, T=545°R, P=29.38 in. Hg
BMEP	brake mean effective pressure (psi)
BSFC	brake specific fuel consumption (lbm fuel/BHP_c-hr)
BSEC	brake specific energy consumption (Btu/BHP_c-hr)
PMAX	peak pressure (psi)
PRATMAX	maximum rate of pressure rise (psi/degree crank angle)
IGNDLY	ignition delay (degrees crank angle)
AF	air-fuel ratio
PHI	equivalence ratio
TEX	exhaust temperature (°F)
VOLEFF	volumetric efficiency (percent)
CO	power specific emission of carbon monoxide (gm/ihp-hr)
HC	power specific emission of unburned hydrocarbons (gm/ihp-hr)
NOX	power specific emission of oxides of nitrogen (gm/ihp-hr)

Table C.1

Reduced Experimental Data

Operating Condition: 1/3 Rack, 2400 RPM

Fuel: 200 Proof Ethanol

%ALCH	0	10	20	30	40
THEFF	15.85	14.42	13.08	12.14	8.17
BHP$_c$	1.36	1.24	1.13	1.05	.71
BMEP	20.98	19.06	17.31	16.08	10.84
BSFC	.83	.975	1.14	1.30	2.06
BSEC	16067.	17632.	19453.	20943.	31419.
PMAX	931.	957.	985.	846.	715.
PRATMAX	91.	105.	160.	123.	55.
IGNDLY	-	26.12	27.94	30.81	33.34
AF	49.97	46.36	43.67	41.63	39.68
PHI	.299	.302	.302	.300	.298
TEX	392.	373.	362.	352.	341.
VOLEFF	87.7	86.2	85.6	86.2	87.1
CO	5.21	7.39	10.25	14.5	23.32
HC	.90	2.32	2.97	4.68	8.41
NO	2.30	1.73	1.60	1.06	.40
NOX	2.99	2.99	3.20	2.68	1.48

Table C.2

Reduced Experimental Data

Operating Condition: 2/3 Rack, 2400 RPM

Fuel: 200 Proof Ethanol

%ALCH	0	10	20	30	40	50
THEFF	22.68	23.16	23.48	22.75	21.91	21.45
BHP$_c$	2.89	2.98	3.02	2.91	2.80	2.74
BMEP	44.23	45.63	46.15	44.58	42.84	41.96
BSFC	.58	.61	.63	.69	.76	.82
BSEC	11214.	10959.	10834.	11183.	11608.	11858.
PMAX	927.	942.	975.	984.	937.	861.
PRATMAX	83.	107.	124.	134.	93.	86.
IGNDLY	25.49	25.85	27.93	29.02	31.75	35.28
AF	33.10	30.65	29.08	27.71	26.47	25.27
PHI	.452	.457	.454	.450	.448	.445
TEX	546.	534.	528.	519.	493.	514.
VOLEFF	86.9	85.5	85.3	85.3	85.1	84.0
CO	6.47	8.04	8.67	10.27	11.92	13.98
HC	1.31	2.50	2.79	4.01	4.28	5.41
NO	2.48	2.49	2.47	2.28	2.02	1.57
NOX	2.73	2.86	3.15	3.19	3.17	2.50

Table C.3

Reduced Experimental Data

Operating Condition: Full Rack, 2400 RPM

Fuel: 200 Proof Ethanol

%ALCH	0	10	20	30	40	50	55
THEFF	23.98	25.49	25.91	26.49	26.80	26.38	26.00
BHP_c	4.08	4.33	4.40	4.52	4.55	4.48	4.41
BMEP	62.42	66.26	67.31	69.06	69.58	68.53	67.48
BSFC	.55	.55	.57	.59	.62	.67	.69
BSEC	10609.	9981.	9816.	9597.	9492.	9643.	9780.
PMAX	993.	1029.	1042.	1052.	1076.	996.	953.
PRATMAX	134.	139.	161.	168.	165.	170.	140.
IGNDLY	23.11	25.92	26.33	27.88	28.94	32.90	33.84
AF	24.43	22.68	21.67	20.58	19.57	18.73	18.31
PHI	.612	.618	.610	.606	.601	.599	.598
TEX	716.	707.	686.	690.	666.	642.	645.
VOLEFF	85.8	83.9	84.0	84.1	82.6	80.8	80.2
CO	11.65	12.29	13.65	14.25	13.74	14.91	12.73
HC	2.16	2.41	2.41	2.35	2.47	3.24	4.23
NO	2.31	2.45	2.62	2.81	2.91	2.66	2.73
NOX	2.44	2.64	2.88	3.17	3.31	3.24	3.46

Table C.4

Reduced Experimental Data

Operating Condition: 1/3 Rack, 2400 RPM

Fuel: 180 Proof Ethanol

%ALCH	0	10	20	30	35
THEFF	15.45	14.32	13.33	11.95	10.32
BHP_c	1.33	1.23	1.15	1.04	.90
BMEP	20.46	18.88	17.66	15.91	13.81
BSFC	.86	.98	1.12	1.32	1.58
BSEC	16462.	17770.	19082.	21282.	24633.
PMAX	873.	892.	898.	862.	744.
PRATMAX	92.	119.	135.	121.	70.
IGNDLY	27.29	28.29	29.81	31.10	34.27
AF	50.09	46.43	43.80	41.61	40.28
PHI	.298	.302	.301	.300	.299
TEX	400.	381.	369.	371.	356.
VOLEFF	89.3	87.0	87.1	87.3	87.7
CO	4.93	7.24	10.39	14.93	19.52
HC	.809	1.75	3.17	4.44	6.04
NO	2.36	1.71	1.54	.97	.52
NOX	3.04	2.97	3.13	2.60	1.90

Table C.5

Reduced Experimental Data

Operating Condition: 2/3 Rack, 2400 RPM

Fuel: 180 Proof Ethanol

%ALCH	0	10	20	30	40	45
THEFF	22.28	22.97	23.28	22.59	21.80	21.18
BHP$_c$	2.86	2.95	2.99	2.89	2.82	2.72
BMEP	43.70	45.10	45.80	44.23	43.18	41.61
BSFC	.59	.61	.64	.70	.77	.81
BSEC	11417.	11076.	10927.	11262.	11672.	12009.
PMAX	929.	971.	972.	995.	937.	815.
PRATMAX	91.	109.	114.	196.	156.	75.
IGNDLY	25.05	26.35	28.22	30.23	33.84	36.14
AF	32.92	30.92	29.28	28.0	26.43	26.29
PHI	.454	.453	.450	.448	.444	.437
TEX	550.	536.	525.	511.	491.	504.
VOLEFF	87.30	86.90	87.20	87.10	87.50	88.20
CO	7.24	8.51	9.18	10.56	11.95	13.87
HC	1.60	2.38	2.77	3.06	3.82	5.15
NO	2.53	2.37	2.24	2.31	1.80	1.14
NOX	2.74	2.79	3.00	3.04	2.98	2.06

Table C.6

Reduced Experimental Data

Operating Condition: Full Rack, 2400 RPM

Fuel: 180 Proof Ethanol

%ALCH	0	10	20	30	40	50
THEFF	23.85	25.23	26.23	26.61	26.85	26.52
BHP_c	4.07	4.30	4.46	4.54	4.57	4.52
BMEP	62.24	65.73	68.18	69.41	69.93	69.06
BSFC	.55	.56	.57	.59	.62	.66
BSEC	10664.	10085.	9699.	9558.	9474.	9593.
PMAX	994.	1015.	1069.	1103.	1085.	1010.
PRATMAX	96.	98.	118.	161.	182.	154.
IGNDLY	23.46	24.69	27.23	28.89	31.03	34.28
AF	24.49	22.85	21.72	20.70	19.79	18.80
PHI	.612	.613	.607	.603	.598	.597
TEX	726.	713.	694.	676.	664.	652.
VOLEFF	86.5	85.0	84.1	83.5	81.8	81.4
CO	10.46	12.64	12.62	12.85	11.83	11.10
HC	2.03	2.30	2.36	2.57	2.73	3.32
NO	2.20	2.34	2.56	2.63	2.73	2.66
NOX	2.33	2.50	2.80	2.99	3.17	3.27

Table C.7

Reduced Experimental Data

Operating Condition: 1/3 Rack, 2400 RPM

Fuel: 160 Proof Ethanol

%ALCH	0	10	20	30	35
THEFF	15.11	14.54	13.0	10.98	9.74
BHP_c	1.31	1.25	1.13	.96	.84
BMEP	20.11	19.0	17.28	14.66	12.91
BSFC	.88	.97	1.15	1.44	1.67
BSEC	16839.	17506.	19570.	23188.	26122.
PMAX	867.	901.	887.	840.	730.
PRATMAX	102.	104.	134.	103.	51.
IGNDLY	26.93	28.73	29.81	31.11	33.63
AF	49.87	46.18	43.76	41.51	40.85
PHI	.301	.303	.301	.302	.297
TEX	400.	383.	364.	355.	358.
VOLEFF	89.4	86.9	86.3	85.1	85.8
CO	4.42	6.86	10.67	17.32	21.27
HC	.697	1.83	3.20	4.86	7.23
NO	2.43	1.98	1.64	.73	.37
NOX	3.04	3.28	3.22	2.29	1.67

Table C.8

Reduced Experimental Data

Operating Condition: 2/3 Rack, 2400 RPM

Fuel: 160 Proof Ethanol

%ALCH	0	10	20	30	40
THEFF	22.24	23.08	23.84	23.56	22.34
BHP$_c$	2.85	2.97	3.06	3.02	2.85
BMEP	43.63	45.38	46.77	46.25	43.63
BSFC	.59	.61	.63	.67	.75
BSEC	11435.	11023.	10671.	10797.	11387.
PMAX	922.	954.	983.	959.	895.
PRATMAX	84.	96.	128.	128.	132.
IGNDLY	25.35	26.78	28.72	31.25	34.70
AF	32.99	30.80	29.36	27.86	26.74
PHI	.455	.455	.450	.448	.444
TEX	553.	540.	516.	503.	498.
VOLEFF	87.6	85.5	84.2	83.0	81.9
CO	7.02	8.37	8.79	10.51	12.45
HC	1.62	2.47	2.63	3.41	4.26
NO	2.56	2.43	2.28	2.01	1.63
NOX	2.80	2.82	2.94	2.86	2.63

Table C.9

Reduced Experimental Data

Operating Condition: Full Rack, 2400 RPM

Fuel: 160 Proof Ethanol

%ALCH	0	10	20	30	40	45
THEFF	23.81	25.34	26.70	27.40	27.71	27.07
BHP$_c$	4.06	4.31	4.61	4.68	4.74	4.59
BMEP	62.13	65.97	70.51	71.56	72.43	70.16
BSFC	.55	.55	.56	.58	.60	.63
BSEC	10657.	10038.	9528.	9284.	9179.	9394.
PMAX	964.	998.	1060.	1083.	1007.	890.
PRATMAX	109.	101.	129.	162.	168.	114.
IGNDLY	23.52	26.43	28.29	29.95	33.05	36.58
AF	24.46	22.66	21.23	20.48	19.33	19.11
PHI	.612	.619	.608	.609	.611	.602
TEX	722.	708.	697.	689.	700.	695.
VOLEFF	85.8	82.6	81.7	79.9	78.9	78.6
CO	9.89	14.11	12.30	11.99	10.71	10.67
HC	2.14	2.26	2.35	2.41	2.73	3.28
NO	2.17	2.22	2.49	2.68	2.43	2.18
NOX	2.30	2.38	2.73	3.03	2.75	2.50

Table C.10

Reduced Experimental Data

Operating Condition: 1/3 Rack, 2400 RPM

Fuel: 140 Proof Ethanol

%ALCH	0	10	20	30
THEFF	15.76	14.21	12.75	10.94
BHP_c	1.36	1.22	1.10	.94
BMEP	20.78	18.68	16.94	14.49
BSFC	.84	.99	1.18	1.45
BSEC	16143.	17908.	19954.	23256.
PMAX	885.	894.	914.	848.
PRATMAX	95.	98.	163.	89.
IGNDLY	26.98	29.09	30.24	32.10
AF	50.34	47.12	43.90	42.33
PHI	.298	.296	.299	.294
TEX	396.	380.	359.	345.
VOLEFF	88.4	86.9	85.8	85.4
CO	5.05	7.89	11.18	18.10
HC	.781	1.60	2.56	3.69
NO	2.29	1.57	1.28	.61
NOX	2.88	2.80	2.74	1.99

Table C.11

Reduced Experimental Data

Operating Condition: 2/3 Rack, 2400 RPM

Fuel: 140 Proof Ethanol

%ALCH	0	10	20	30	35
THEFF	22.83	23.16	23.72	22.70	22.22
BHP_c	2.95	2.97	3.05	2.91	2.86
BMEP	45.22	45.39	46.62	44.52	43.82
BSFC	.58	.61	.63	.70	.73
BSEC	11141.	10997.	10723.	11204.	11447.
PMAX	947.	963.	993.	945.	924.
PRATMAX	87.	89.	139.	138.	104.
IGNDLY	25.63	26.68	29.09	31.04	33.19
AF	33.03	30.96	29.43	27.95	27.06
PHI	.453	.453	.449	.448	.446
TEX	550.	530.	516.	497.	493.
VOLEFF	86.7	85.3	83.8	82.4	82.4
CO	7.18	8.13	8.42	10.40	10.83
HC	1.54	2.03	2.31	2.61	3.00
NO	2.45	2.21	2.07	1.65	1.49
NOX	2.71	2.68	2.82	2.57	2.46

Table C.12

Reduced Experimental Data

Operating Condition: Full Rack, 2400 RPM

Fuel: 140 Proof Ethanol

%ALCH	0	10	20	30	40
THEFF	24.34	25.55	26.55	28.03	28.19
BHP_c	4.12	4.34	4.52	4.77	4.81
BMEP	63.03	66.34	69.14	72.98	73.51
BSFC	.54	.55	.56	.56	.59
BSEC	10452.	9959.	9582.	9076.	9025.
PMAX	991.	1024.	1036.	1092.	1058.
PRATMAX	113.	.150	161.	207.	230.
IGNDLY	23.62	25.92	28.73	30.02	34.49
AF	24.67	23.08	21.9	20.54	19.4
PHI	.606	.609	.602	.605	.608
TEX	716.	691.	696.	689.	685.
VOLEFF	85.6	82.5	80.6	79.4	79.2
CO	10.20	11.68	12.46	10.68	10.43
HC	1.99	2.79	2.45	2.41	2.53
NO	2.20	2.23	2.13	2.19	2.03
NOX	2.33	2.40	2.37	2.44	2.34

Table C.13

Reduced Experimental Data

Operating Condition: 1/3 Rack, 2400 RPM

Fuel: 200 Proof Methanol

%ALCH	0	10	20	30
THEFF	15.05	13.10	11.49	9.49
BHP_c	1.30	1.13	.99	.82
BMEP	19.93	17.31	15.21	12.58
BSFC	.88	1.14	1.43	1.91
BSEC	16894.	19439.	22135.	26875.
PMAX	854.	902.	880.	793.
PRATMAX	76.	133.	121.	80.
IGNDLY	26.64	28.37	30.24	31.97
AF	49.63	43.42	39.23	36.29
PHI	.301	.305	.304	.298
TEX	400.	386.	377.	356.
VOLEFF	88.5	86.2	86.1	85.2
CO	4.96	7.21	11.09	19.12
HC	.81	1.48	2.69	3.73
NO	2.32	1.53	1.09	.37
NOX	2.97	3.47	3.48	2.40

Table C.14

Reduced Experimental Data

Operating Condition: 2/3 Rack, 2400 RPM

Fuel: 200 Proof Methanol

%ALCH	0	10	20	30	35
THEFF	22.01	22.50	22.52	22.0	21.21
BHP_c	2.82	2.89	2.89	2.83	2.74
BMEP	43.18	44.23	44.23	43.18	41.96
BSFC	.60	.66	.73	.82	.89
BSEC	11555	11304	11293	11563	11995
PMAX	910	949.	973.	953.	910.
PRATMAX	90.	119.	133.	142.	120.
IGNDLY	25.27	26.85	28.52	31.97	34.49
AF	32.93	28.76	26.61	24.36	23.34
PHI	.454	.461	.448	.444	.444
TEX	547.	532.	519.	498.	480.
VOLEFF	86.8	83.9	84.3	83.8	83.4
CO	7.38	7.85	9.22	10.44	10.86
HC	1.56	1.92	2.01	2.12	3.20
NO	2.51	2.16	1.83	1.51	1.37
NOX	2.82	3.01	3.28	3.25	3.39

Table C.15

Reduced Experimental Data

Operating Condition: Full Rack, 2400 RPM

Fuel: 200 Proof Methanol

%ALCH	0	10	20	30	40
THEFF	23.71	25.03	25.95	26.56	26.13
BHP_c	4.03	4.24	4.40	4.52	4.46
BMEP	61.72	64.86	67.31	69.06	68.18
BSFC	.56	.59	.63	.68	.75
BSEC	10729.	10161.	9803.	9579.	9736.
PMAX	945.	1004.	1038.	1062.	851.
PRATMAX	91.	112.	120.	142.	73.
IGNDLY	23.54	26.50	27.43	29.74	34.85
AF	-	21.64	19.87	18.03	16.85
PHI	-	.613	.601	.600	.585
TEX	710.	704.	696.	679.	689.
VOLEFF	-	83.0	83.2	81.7	80.3
CO	11.06	13.09	12.56	10.53	9.12
HC	2.69	2.15	2.47	1.95	2.25
NO	2.32	2.32	2.52	2.71	2.18
NOX	2.47	2.53	2.90	3.30	2.91

Table C.16

Reduced Experimental Data

Operating Condition: 1/3 Rack, 2400 RPM

Fuel: 160 Proof Methanol

%ALCH	0	10	20	25
THEFF	14.85	13.50	12.10	10.52
BHP_c	1.28	1.17	1.05	.92
BMEP	19.74	17.99	16.06	14.15
BSFC	.89	1.10	1.36	1.63
BSEC	17127.	18839.	21015.	24180.
PMAX	868.	885.	784.	646.
PRATMAX	88.	120.	91.	40.
IGNDLY	26.64	29.01	33.55	35.94
AF	49.0	43.4	39.8	37.81
PHI	.305	.305	.298	.301
TEX	396.	369.	370.	359.
VOLEFF	87.6	85.8	84.9	84.3
CO	4.47	7.36	15.29	21.92
HC	.669	1.14	2.16	3.32
NO	2.42	1.34	.44	.17
NOX	2.94	3.10	2.17	1.30

Table C.17

Reduced Experimental Data

Operating Condition: 2/3 Rack, 2400 RPM

Fuel: 160 Proof Methanol

%ALCH	0	10	20	30
THEFF	21.77	22.15	22.42	21.76
BHP$_c$	2.81	2.83	2.87	2.79
BMEP	42.96	43.31	44.01	42.61
BSFC	.60	.67	.73	.83
BSEC	11682.	11485.	11335.	11694.
PMAX	897.	923.	925.	758.
PRATMAX	82.	112.	119.	76.
IGNDLY	25.63	28.00	30.68	37.38
AF	32.92	29.29	26.67	24.38
PHI	.382	.456	.449	.443
TEX	538.	507.	504.	504.
VOLEFF	86.9	83.6	82.2	80.9
CO	7.57	7.41	8.60	10.56
HC	1.59	1.73	1.99	2.38
NO	2.45	2.05	1.64	1.24
NOX	2.65	2.78	2.66	2.39

Table C.18

Reduced Experimental Data

Operating Condition: Full Rack, 2400 RPM

Fuel: 160 Proof Methanol

%ALCH	0	10	20	30
THEFF	22.99	24.62	25.61	26.1
BHP_c	3.89	4.17	4.34	4.44
BMEP	59.56	62.75	66.37	67.94
BSFC	.57	.60	.64	.69
BSEC	11064.	10333.	9929.	9749.
PMAX	929.	993.	1018.	966.
PRATMAX	85.	122.	122.	97.
IGNDLY	23.68	27.00	29.40	34.63
AF	24.28	21.48	19.61	17.80
PHI	.616	618.	.609	.607
TEX	716.	705.	685.	703.
VOLEFF	85.2	82.7	79.4	78.8
CO	11.39	11.08	11.65	9.84
HC	2.09	1.92	1.67	1.72
NO	1.98	2.27	2.09	2.12
NOX	2.10	2.49	2.45	2.51

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16. Abstract

A single-cylinder DI Diesel engine was fumigated with ethanol and methanol in amounts up to 55% of the total fuel energy. The primary objectives of this work were to determine the effects of aqueous alcohol fumigation on engine thermal efficiency, combustion intensity and gaseous exhaust emissions. Assessment of changes in the biological activity of raw particulate and its soluble organic fraction were also made using the Ames Salmonella typhimurium test. Alcohol fumigation improved thermal efficiency slightly at moderate and heavy loads, but increased ignition delay at all operating conditions. Carbon monoxide and unburned hydrocarbon emission generally increased with alcohol fumigation and showed no dependence on alcohol type or quality. Oxide of nitrogen emission showed a strong dependence on alcohol quality; relative emission levels decreased with increasing water content of the fumigant. Particulate mass loading rates were lower for ethanol-fueled conditions. However, the biological activity of both the raw particulate and its soluble organic fraction was enhanced by ethanol fumigation at most operating conditions.

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