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Alcohol as a Fuel for Farm and Construction Equipment

MASTER

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Prepared for:
U.S. Department of Energy
Assistant Secretary for Conservation and Renewable Energy
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Alternative Fuels Utilization Program

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ABSTRACT

This final report summarizes work in three areas dealing with the utilization of ethanol as fuel for farm and construction diesels. The first part is a review of what is known about the retrofitting of diesels for use of ethanol and the combustion problems involved. The second part is a discussion of the work that has been done under the contract on the performance of a single-cylinder, open-chamber diesel using solutions and emulsions of diesel fuel with ethanol. Data taken include performance, emissions and cylinder pressure-time for diesel fuel with zero to forty percent ethanol by volume. Analysis of the data includes calculation of heat release rates using a single zone model. The third part is a discussion of work done retrofitting a multicylinder turbocharged farm tractor diesel to use ethanol by fumigation. Three methods of ethanol introduction are discussed; spraying ethanol upstream and downstream of the compressor and prevaporization of the ethanol. Data on performance and emissions are given for the last two methods. A three zone heat release model is described and results from the model are given. A correlation of the ignition delay using prevaporized ethanol fumigation data is also given. Comparisons are made between fumigation in DI and IDI engines.

1. INTRODUCTION

An adequate supply of diesel fuel for both farm and highway uses is vital to the nation's economy. A serious shortage of diesel fuel at planting time, for example, would be disastrous to farmers and to our nation's food supply. For this reason it is important to consider ways of utilizing alternate fuels in diesels. Gasoline is obviously one such fuel, but might also be in short supply. The use of ethanol produced from biomass offers one method of providing an emergency alternative fuel supply. Large scale production of methanol provides yet another possible alternative fuel. However, neither ethanol nor methanol is a satisfactory diesel fuel because of the inherent difficulty in making them autoignite in a diesel. It is thus necessary to retrofit existing diesel engines to allow the use of alcohol.

At the time of the initiation of this contract there was considerable interest among farmers in the local production of ethanol and its utilization as an emergency fuel for farm diesels. Many questions concerning methods of retrofitting and the use of ethanol as a diesel fuel were being raised both by the diesel users and diesel powered equipment producers. The contract was thus set up by DOE to answer some of these needs. The specific purposes of the contract were: one, to keep abreast of current technology in the utilization of alcohol in farm and construction equipment and to act as a source for those seeking such information; two, to investigate methods of retrofitting and to test such methods experimentally as required; three, to conduct research to improve the knowledge of the effects of ethanol, as an alternative fuel, on diesel combustion.

The remainder of the text of this final report is divided into three main sections and a summary. The next section gives an overview of the status of alcohol utilization as a fuel in diesel engines. It is followed by a section giving the major results of the single cylinder diesel study of diesel fuel — ethanol emulsions and solutions. The final discussion section gives the major results of the study on the use of fumigation of ethanol in a four cylinder turbocharged diesel tractor engine. Each of these sections represents an abbreviated report; those wishing more details should see the list of publications given in Appendix A. Those wishing to read only the conclusions will find these conveniently compiled in the summary at the end of the report.

2. TECHNOLOGY ASSESSMENT

It is well known that because of its high octane number and poor lubricity alcohol is a poor diesel fuel. Yet there still is much confusion among both technical and lay persons as to appropriate techniques and precautions for using alcohol in diesel engines. We believe that the major obstacle to be overcome in using alcohol in a diesel is flame initiation. The lubricity problem can be overcome with small quantities of additives, for example caster oil (Holmer, 1979) and wear problems seem solvable. Consequently the use of alcohol in diesel engines will be discussed below according to the ignition technique used. The section ends with a brief discussion of the problem of accelerated wear in engines using alcohol.

Cetane Improvers

One approach to using alcohol would be to improve its cetane number and lubricity by chemical additives. The cetane improvement necessary is so great that large amounts of cetane improver must be added to the alcohol, 20% or more of Cetanox (Holmer, 1979) and 15% or more of hexylnitrate. Preheating the induction air in a heat exchanger using engine coolant decreases the demand for additive, however cold starting problems may result. The primary drawback to this method is the expense and availability of the cetane improvers. If better cetane improvers can be found, this could become a viable method for alcohol use in diesels. It has the advantage of essentially total substitution of alcohol for diesel fuel. If a viable cetane improver were found and used with the normal injection system, it would result in a decrease in engine power due to the alcohol's lower energy content. Therefore either a new injection system with a larger volume would be necessary or lower engine power (approximately two thirds lower power with ethanol) would have to be accepted. There is also concern over the injector life when using neat alcohol because of lack of lubricity and possible cavitation damage. In the present study alcohol emulsions resulted in injector damage, see Table 3-4.

Spark Ignition

Another alternative is to convert the diesel engine into a spark ignited engine. It has been shown (Finsterwalder, 1972; Urlaub et al., 1974; Adelman, 1982) that this is feasible in a diesel engine. Converting to spark ignition has the advantage that one can run on pure alcohol but has the disadvantage of not being well suited for retrofitting, particularly for turbocharged engines.

Because of knocking problems the compression ratio would have to be lowered to a value which is compatible with the octane number of the fuel. The method used to get the alcohol into the cylinder can cause problems. If the injector is replaced by a spark plug and a carburetor is used, the cylinder-to-cylinder distribution of the alcohol will probably be poor because the inlet manifold is not designed for two-phase flow. Furthermore if throttling is used to control the load there will be a corresponding loss in engine efficiency. If the injector is used, one must find a suitable location for the spark plug and the problem of injector wear still remains.

Similar to spark ignition is the idea of using surface or glow plug ignition (Nagalingam et al., 1980; Miyamoto et al., 1982). The alcohol would be injected so that it impinged on a hot surface which would result in ignition. To date no one has been successful in building a multi-cylinder working engine of this type. Thus, the concept of creating an ignition source in the cylinder of a diesel engine remains a promising but as yet undeveloped means of providing a multi-fuel engine.

Twin Injectors

Another method of ignition of low cetane fuels is to provide for a pilot injection of diesel fuel. This would require an engine with a special head with two injectors, and is thus expensive. Dual injection engines have been built and tested, the most notable is the work done by Pischinger et al. (1979) and Holmer et al. (1979). Pischinger examined the amount of pilot diesel fuel, injection timing and injection rate of diesel fuel and methanol, and the influence of compression ratio on engine operation. It was concluded that:

- A large portion of diesel fuel can be substituted by alcohol while maintaining a reliable ignition and knock free combustion. The amount of diesel fuel used as the pilot was roughly equivalent to that needed at idle.

- Efficiency is equal to or better than the efficiency of the standard diesel engine.

- Lower peak pressures and maximum rates of pressure rise are achieved.

- Very low smoke values are obtained and a soot limit is nonexistent.

- Nitric oxide emissions are reduced by more than half in comparison to standard diesel operation.

- The HC and CO emissions are equal to or lower than the corresponding emissions of the standard diesel operation.

- Thermal and mechanical stresses are reduced.

- The compression ratio can be varied over a wide range (14.5:1 - 19.3:1) without adversely affecting the normal operating behavior.

The above results of Pischinger et al. and the results of Holmer et al. are encouraging. Engine performance is good and large substitution rates are possible. The principal drawback of this method is its implementation expense. It is not well suited for retrofitting existing diesel engines.

Fumigation

The concept of using a pilot injection to ignite low cetane number fuel (alcohol) is expensive because of the extra injector system. A similar method can be used, however, if the low cetane number fuel is introduced into the cylinder through the intake valve as a lean fuel-air mixture. The alcohol would be ignited by diesel fuel injected directly into the cylinder. This approach has also been studied by several researchers (Alperstein et al., 1958; Panchapakesan et al., 1978; Cruz et al., 1979). It involves installing a carburetor on the engine so it will have disadvantages similar to those of the spark ignition conversion. All of the researchers found similar results. At small pilot levels it was relatively easy to operate at light loads with small quantities of alcohol; progressive increases in alcohol resulted in abrupt failure of ignition due to quenching. This placed a limit on the amount of alcohol that could be burned at any one speed, the effect being more severe as the speed is increased due to lengthening of the ignition delay relative to crankangle. Using the maximum replacement rate to achieve power equal to normal diesel operation, torque curves of engine operation with fumigated alcohol and diesel injection indicate that the fuel consumption on a unit energy basis is equal to or better than that of pure diesel.

Heating the intake charge to overcome the quench limitation has also been attempted. With mixture heating it is possible to achieve a higher percentage of alcohol substitution. Under these conditions the maximum load limit was no longer limited by quench but instead by knock of the end gas. This leads to a situation in which the limiting alcohol utilization at high loads and low speeds is determined by knock and at high speeds the limit is determined by quenching. Thus the operating conditions are going to be markedly different between low and high speed if similar alcohol utilization is to be achieved.

In 1979 a commercial company proposed a prototype system for fumigating ethanol into a turbocharged diesel engine. High pressure air from the exit side of the compressor was used to pressurize an alcohol tank which then introduced a stream of ethanol into the intake manifold on the low pressure side of the compressor. We tested this so-called "Aquahol System" in our laboratory and the results are given in Section 4 of this report.

Recently another manufacturing company has started testing a fumigation system of different design (Corn Grower, 1981). The alcohol flow rate is generated by an auxiliary pump and is injected into the high pressure side of the compressor only after the engine reaches two thirds of its load capacity. We are not familiar with details of the operation of this system.

Fumigation is reasonably well suited for retrofit. Anticipated problems include the possibility of overloading the engine and possibly an increased rate of engine wear. These, however, can be considered control and development problems and not technical ones. However one should bear in mind that fumigation would be used as a diesel fuel extender and not a substitute. The average substitution possible would probably be less than 50 percent of the energy.

Combining Alcohol and Diesel Fuel

The use of gasohol in automobile spark-ignition engines is well known, but a similar approach to diesels is not viable because diesel fuel and alcohol do not mix well. Methanol and diesel fuel are insoluble while ethanol and diesel are compatible in practice only if both fuels are completely dry. Typically there will be a small amount of water in the diesel fuel which is usually enough to cause mixture separation. Under ideal conditions a solution of 30% ethanol and 70% diesel has been used in a farm tractor (Strait et al., 1979). This solution was found to be very unstable. Tests using this solution resulted in engine efficiencies on a per unit energy basis equivalent to the same engine run on straight diesel. One notable observation is the virtual elimination of smoke and particulates with only small additions of alcohol. This clearly warrants further investigation and is added motivation to use alcohol in some proportion in diesel engines. However recent experiments by Heisey and Lestz (1981) suggest that there may be additional health concerns with particulates from alcohol supplemented diesels.

The intolerance of alcohol diesel blends to water contamination may be overcome with the use of surfactants. Currently no such surfactants are available commercially at a price which would make them feasible; though the work of Reeves et al. (1982) suggest that they may be available shortly.

Emulsions

Another means of mixing the two fuels is to run them through an emulsifier. This device breaks the alcohol into very tiny droplets so that the fuel at the outlet is a mixture of diesel fuel and alcohol droplets. The emulsions are usually unstable but this is not a problem if the emulsifier is installed in line with the injector. The system is complicated by the fact that a large portion of the fuel must be recirculated thus a separate pump and fuel cooler must be provided along with the emulsifier unit.

The addition of emulsified alcohol lowers the cetane number and thus limits the amount of alcohol used to a small percentage of the total fuel used, 20 to 30 percent. The injector must be properly sized to accomodate the lower energy emulsion fuels, and some flexibility is lost. The use of emulsified fuels is currently best suited to steady state, full load conditions; the design of systems suitable for field conditions requires further development.

Engine Wear

Alcohol's influences on internal combustion engine lubrication differ from those of petroleum fuels because of their differing physical and chemical properties. Because of their high latent heat of vaporization liquid alcohols are more likely to reach the cylinder wall than petroleum based fuel. Since alcohol is immiscible with oil, but is miscible with water, condensates of unburned alcohol and water in the engine form an emulsion with oil.

Blowby gas of an alcohol-fueled engine contains higher concentrations of components expected to contribute to corrosion and wear than petroleum fuel engine blowby; acetaldehyde, formaldehyde and formic acid would be expected in the blowby, as they are present in the exhaust.

It is believed that wear in an alcohol engine involves several mechanisms. An emulsion, if formed and distributed in an engine, may restrict the supply of oil for boundary lubrication. Alcohol and water droplets in the emulsion may flash to vapor on contact with hot surfaces, leaving insufficient oil on the area to be lubricated. Alcohols and their corrosive combustion products attack metals such as aluminum, copper and lead alloys, create pits and provide sites for fatigue crack initiation at surfaces. Abrasive particles are generated and distributed between sliding surfaces, intensifying engine component wear. Alcohol may decrease the effectiveness of oil additives by changing their chemical environment.

Tests done by Chui and Millard (1981) found the problem of wear to be especially severe during engine warm up, which accelerated wear rates in the cylinder bore, cam follower and rod bearings. It was also found that the wear problem was more severe with methanol than ethanol. If alcohol is to be used in diesel engines on a retrofit basis, the oil change interval should be drastically reduced or a special oil used.

Conclusions

To date there are no practical, inexpensive methods for using alcohol as a diesel fuel. Cetane improver additives and refinery produced emulsions are currently not viable techniques. Methods such as spark or hot surface ignition and pilot injection are expensive and probably limited to new engine designs. Thus, only fumigation and on-board emulsification are serious candidates for current use. Either method, if perfected, could be used to retrofit existing engines and could provide a choice of fuel. However, both methods require two fuel systems and do not allow use of 100% alcohol. Both methods must be used with caution since they could cause engine damage.

3. U.W. WORK ON DIRECT INJECTION OF EMULSIONS AND SOLUTIONS

Anhydrous ethanol will go into solution with diesel fuel, but the presence of as little as 0.5% water will cause separation. Thus for practical applications the combining of ethanol and diesel fuel requires either a surfactant or an emulsifier.

Microemulsions with a chemical stabilizer can produce emulsions which will last for a month or more (Baker, 1981) so that one can imagine the stabilized emulsion being made at the time it is purchased and used shortly thereafter. Another approach is to put a mechanical emulsifier on the engine and to produce the emulsion as it is needed. Such emulsifiers can use a shearing mixer (Lawson et al., 1981; Murayama et al., 1980), or a venturi nozzle (Agosta, 1977) or even ultrasonics. Interestingly, the quality of the emulsion does not seem to have a strong influence on the engine performance (Lawson, 1981). What happens to the emulsion after it has gone through the injector tip and spray breakup is not known and thus the effect of emulsion quality on the combustion events can only be judged from the engine performance data.

In the research reported here, both anhydrous ethanol-diesel solutions and unstabilized emulsions were used in a direct-injection, single-cylinder, four-stroke diesel engine. The engine intake air was heated and pressurized to simulate turbocharged engine conditions. Engine performance, regulated gas phase emissions, smoke and cylinder pressure were measured. Heat release rates were calculated using the measured cylinder pressure as an input. Exhaust particulate mass and aldehydes were not measured.

Table 3-1

Engine Specificatons

TACOM/Labeco CLR

Single Cylinder, 4 Cycle, Direct Injection

Bore x stroke	4.5 x 4.5 in. (11.43 x 11.43 cm)	
Displacement	71.57 cu. in. (1172.8 cu. cm)	
Compression Ratio	24.0:1 (measured)	
Connecting Rod Length	9.0 in (22.86 cm)	
Injection Pump	American Bosch	
	Type - APE 1B-80P-4843A	
	8mm Plunger Diameter	
	Timed for Port C1	
Fuel Injector	Roosa Master 20939	
	4 - Hole	
	155 Degree Included Angle	
	0.012 in. Orifice (0.0305 cm)	
	Opening Pressure 2800 psi (193 bar)	
Valve Timing (nominal)	Intake Valve Opens	20°CA BTDC
	Closes	40°CA ABDC
	Exhaust Valve Opens	50°CA BBDC
	Closes	10°CA ATDC

In the following pages a brief description of the experimental apparatus is given followed by a summary of the results and conclusions. A much more detailed report is given by Iwamoto (1982) in his M.S. thesis.

Apparatus

Table 3-1 lists the specifications of the modified TACOM-Labeco engine used in this study. The engine head is a one-of-a-kind machined head (Jessel, 1979) with a Roosa Master injector. The piston is a Mexican hat style giving a 24:1 compression ratio. The intake valve is shrouded so as to allow experiments with different swirl ratios. A cross section and plan view of the engine cylinder are shown in Fig. 3-1. The valve shroud position shown corresponds to the position found to give best BSFC at optimum injection timing, 1000 RPM and 0.5 equivalence ratio. The swirl ratio, estimated from steady flow bench tests, was 1.4 at this shroud setting. The engine as modified is not an optimized combustion system. As will be seen later, the engine gave higher than desirable smoke levels but otherwise acceptable performance.

Fuel System

The fuel system for emulsions of ethanol and diesel fuel used an Ontario Research Hydroshear unit. Because of the small fuel consumption of the single cylinder engine, the Hydroshear design was modified by Ontario Research. The modification resulted in a single discharge orifice

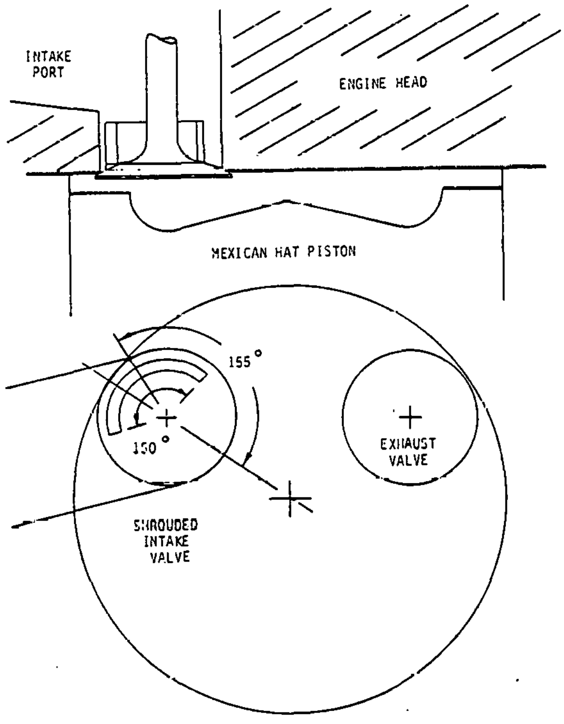


Figure 3-1.
Position of Shrouded Valve of the TACOM-Labeco Engine for Minimum BSFC at an Engine Speed of 1000 RPM

instead of the standard dual orifice. The vortex shearing principle used in the unit is based on an adaptation of the Ranque tube. To obtain good emulsions the fluids should pass through the units at least three times with a very short elapsed time (seconds) between passes. Because of the multipass needs and the fact that the flow through the emulsifier is much larger (15 times) than the flow through the injector, the emulsion flows in a recirculation loop with only a small amount diverted to the injector. The energy added to the fuel is removed by a heat exchanger. Figure 3-2 shows a schematic of the system. Although the diaphragm pump provides excellent metering of its 1380 kPa (200 psi) output, it also causes pressure pulsations. The accumulators shown were added to reduce the pulsations, but the high pressure accumulator was only moderately successful in removing the pressure fluctuations. A gear pump would have been better in this regard, but an appropriate gear pump without materials compatability problems was not located.

Cylinder Pressure

The cylinder pressure was measured with a water cooled AVL 12QP300CVK transducer mounted flush with the cylinder head. The transducer was prepared and calibrated using the procedures of Lancaster et al. (1975). The output was D.C. offset and amplified to bring the signal to a range of -4.5 to +4.5 volts. No RTV coating was used on the diaphragm.

Crankangle pulses were generated by a 720 pulse per revolution encoder driven off the crankshaft. The encoder also gives a marker pulse once per revolution which was electronically "ANDed" with a signal from a half speed shaft to give a marker only at BDC of the compression stroke (both signals are used to trigger the marker pulse). The marker pulse was adjusted to agree with a scribe line marker on the fly wheel.

The pressure data were recorded by use of the lab data acquisition system which consists of an analog to digital converter, a programmable clock

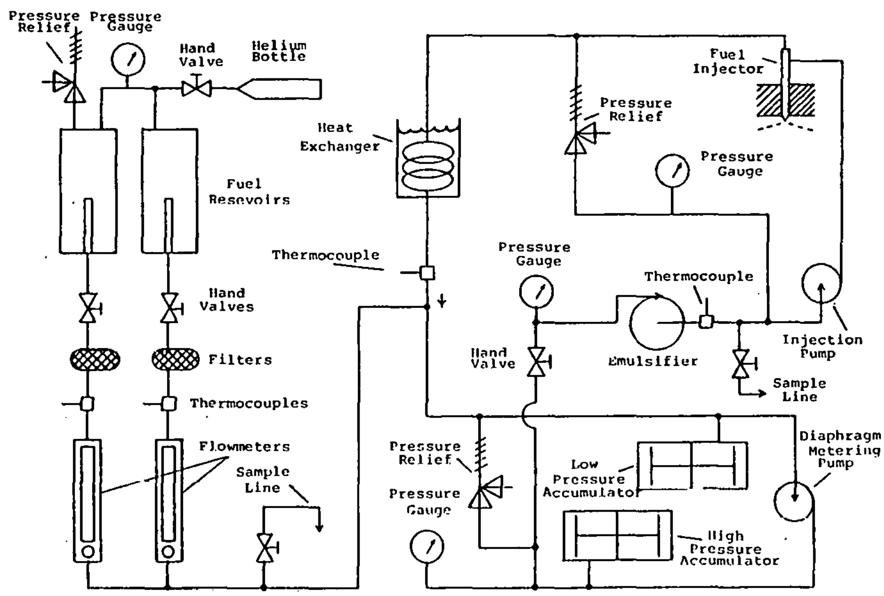


Figure 3-2.
Schematic of the Fuel Delivery System for Direct Injection of Emulsions and Solutions

The same system was used when running ethanol diesel fuel solutions but with the emulsifier by-passed.

Emissions Measurements

Exhaust gas emissions were measured by extracting a flow from the pipe leading out of the exhaust surge tank. A heated FID was used for unburned hydrocarbons, a chemiluminescent analyzer for NO_x, NDIR instruments were used for CO and CO_2 and a paramagnetic instrument for O_2. Smoke readings were taken using a Bosch sampling pump. Unfortunately, because of space limitations, the exhaust was at 145 kPa back pressure at the point where the sample was extracted. Thus the readings were corrected in an approximate manner by multiplying by the atmospheric to back pressure ratio.

generator, an RT-11 microprocessor with memory module, an interactive terminal with graphics capability, a dual 8 inch magnetic disk drive unit and a printer. Pressure data were taken over the 720 crankangles at 0.5° intervals and cycle averaged for 44 cycles. The digitizer has 10 bit accuracy.

For analysis of the data it was transferred from the disk storage to magnetic computer tape. The tape could be read directly into the Harris computer.

Heat Release Analysis

The pressure-crankangle data were spline fit to obtain pressure derivative data. These data were then used in a single zone, first law analysis to provide an apparent rate of burning

Krieger and Borman, 1966). Properties of the products were calculated using the program of Olikara and Borman (1975) and heat transfer was calculated using the correlation of Annand (1962). The leading constant of the Annand correlation was adjusted to make the total heat transfer from the start to end of combustion agree with a first law balance over this interval. In such a balance the work term is evaluated from the pressure data. The internal energy is calculated using the ideal gas temperatures, obtained from the known pressure and trapped mass, assuming that the cylinder gas is homogeneous air plus residual before combustion and homogeneous equilibrium products after the end of combustion.

The first law rate equation was solved by use of a fourth order Runge-Kutta routine with automatic step sizing and error control.

Test Results

The test data were taken with diesel fuel, diesel fuel-ethanol solutions and diesel fuel-ethanol emulsions. The engine parameters varied for each fuel were equivalence ratio, injection timing and intake air temperature. The values of the fixed parameters and the ranges of the varied parameters are given in Table 3-2.

TABLE 3-2

Engine Operating Conditions

Fixed Parameters	
Engine Speed	1000±2 RPM
Boost Pressure	144.8±0.7 kPa
Exhaust Backpressure	144.8±0.7 kPa
Coolant Temperature	80.6±2.0°C
Oil Temperature	68.3±2.5°C
Swirl Ratio	1.4
Variable Parameters	
Fuel Type	
Solutions of Ethanol in Diesel Fuel	0-40% by Volume
Emulsions of Ethanol in Diesel Fuel	0-40% by Volume
Equivalence Ratio	ϕ=0.3 & ϕ=0.5
Fuel Injection Timing	30° - 15° BTDC
Intake Air Temperature	51.7°C & 93.3°C

Test Fuels

The diesel fuel used in the tests was mixed in-house by combining number 1 and number 2 diesel fuel to obtain an API gravity of 35.1. The properties of this fuel, termed here as "research diesel fuel", are listed in Table 3-3. Also listed in Table 3-3 are some of the property values for AMOCO D2-6, a high cetane fuel.

TABLE 3-3

Chemical Analysis Results of the Fuels

Sample of	Research Diesel Fuel	Amoco D2-6 Diesel Fuel
Carbon, %	86.90	86.40
Hydrogen, %	12.71	13.50
Carbon/Hydrogen Ratio	6.84	6.40
Molecular Weight	175	215
Hydrocarbon Types (ASTM D875), % (v/v)		
Saturates	68.4	83.8
Olefins	1.4	1.5
Aromatics	30.2	14.7
Heat of Combustion, gross kJ/kg	45,095	-
Heat of Combustion, net kJ/kg	42,397	
Sulfur, %	0.20	-
API Gravity @ 15.6°C	35.6	-

* Fuels tested at Phoenix Chemical Laboratory, Inc.

Ignition Delay

Cetane tests were run with the two diesel fuels and solutions of these fuels with anhydrous ethanol. The data were taken at the Waukesha Engine Cetane Testing Lab following the ASTM D613 method. The lower aromatic content of the D2-6 fuel (14.7%) limited the amount of ethanol that would go into solution. Addition of 9% castor oil allowed solutions of 30% ethanol to be formed. The cetane test results are given in Fig. 3-3. The slopes of the lines in Fig. 3-3 are constant.

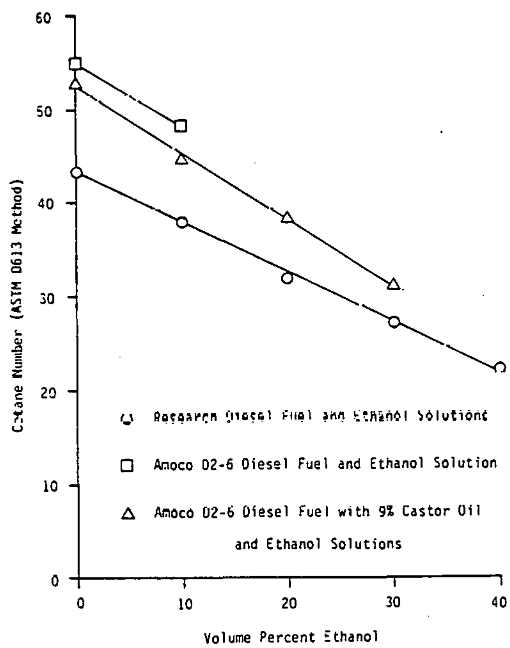


Figure 3-3.
Cetane Number versus Volume Percent Ethanol in Diesel Fuel

The data of Moses et al. (1980) run with a 47.4 cetane diesel fuel show a similar trend.

Engine data were taken with the research diesel fuel and solutions and emulsions of this fuel and ethanol. The effects of injection timing for the neat diesel fuel, 20% solutions and a 20% emulsion are shown in Fig. 3-4. Note that the emulsion contained 1% water and that constant equivalence ratio is essentially constant fuel energy per unit mass. For the neat diesel fuel, the best BSFC was found to be at 22° CA injection for both the 0.3 and 0.5 equivalence ratio. The solution and emulsion gave about the same result with the differences between them and the diesel fuel being essentially constant with injection timing. Figure 3-5 shows the effect of alcohol substitution on ignition delay at a fixed injection timing and two different intake temperatures at very light load. The higher intake temperature reduces the ignition delay by about one crank degree. Again the solutions and emulsion show only small differences. Figure 3-6 shows the effects of both air temperature and injection timing. At this load the air temperature had no effect, but advancing the injection to 30° caused a 2° increase in ignition delay. It should be noted that as the alcohol percentage is increased the cetane number decreases but the volume of fuel must increase for a fixed equivalence ratio (energy). Such an increase in volume flow may have an influence on the spray characteristics, especially at very light loads.

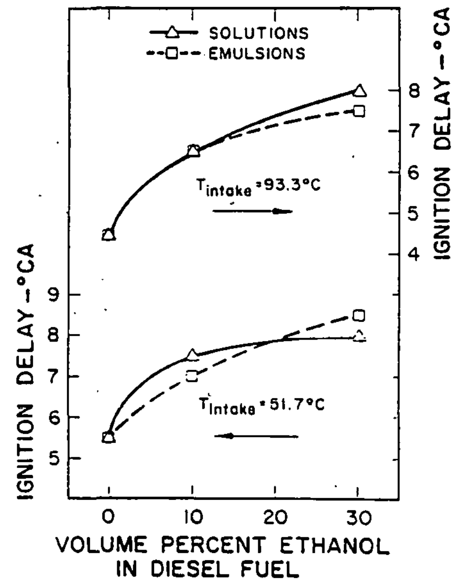


Figure 3-5.
Ignition Delay versus Volume Percent Ethanol in Diesel Fuel for 1000 RPM at an Equivalence Ratio of 0.3 and Injection Timing of 22°BTDC

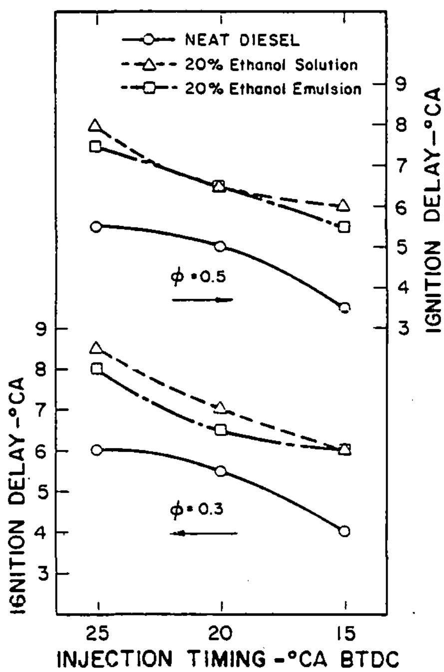


Figure 3-4.
Ignition Delay versus Injection Timing for 1000 RPM, Overall Equivalence Ratios of 0.3 and 0.5 and Various Fuels

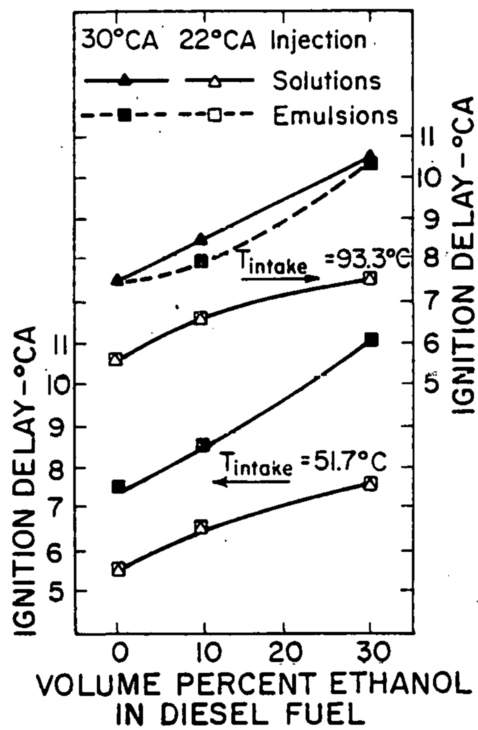


Figure 3-6.
Ignition Delay versus Volume Percent Ethanol in Diesel Fuel for 1000 RPM at Equivalence Ratios of 0.5 for Two Injection Timings and Two Air Temperatures

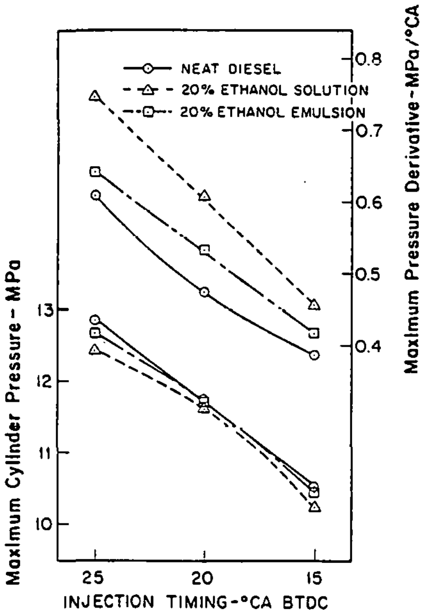


Figure 3-7.
Maximum Cylinder Pressure and Maximum Pressure Derivative versus Injection Timing for 1000 RPM, Overall Equivalence Ratio of 0.3 and Various Fuels

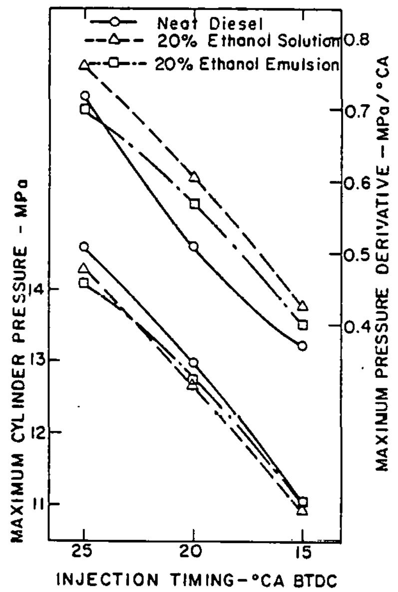


Figure 3-8.
Maximum Cylinder Pressure and Maximum Pressure Derivative versus Injection Timing for 1000 RPM, Overall Equivalence Ratio of 0.5 and Various Fuels

The effects of increased ignition delay are normally to increase the rapid, "premixed", portion of the combustion and thus cause higher rates of pressure rise. Such rapid combustion is typically not detrimental to efficiency but if excessive may cause damage to the engine. Figures 3-7 and 3-8 show the effects of injection timing on peak pressure and rate-of-change-of-pressure for neat diesel fuel and diesel fuel with 20% alcohol for two different loads. The maximum cylinder pressure is slightly lower with alcohol, but the rate-of-pressure-rise is higher. The solutions gave higher rates than the emulsions indicating more vaporized fuel and air were ready to burn for the case of solutions. Heat release analysis (Figs. 3-9 and 3-10) shows that the first

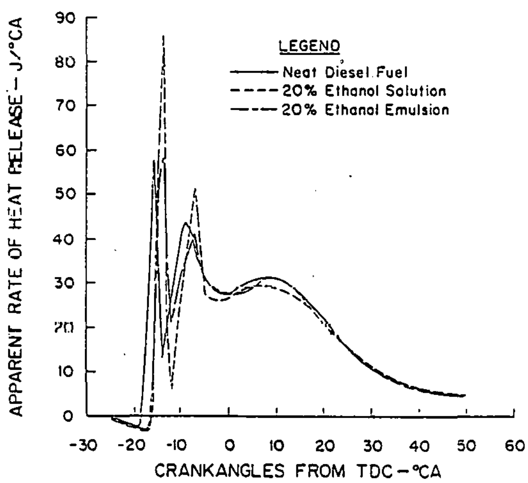


Figure 3-9.
Apparent Rate of Heat Release versus Crankangle for Various Fuels at 1000 RPM, Equivalence Ratio of 0.3, Injection Timing of 25°BTDC and Intake Air Temperature of 51.7°C

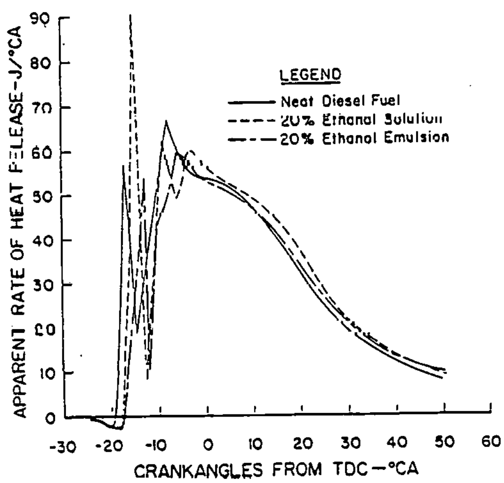


Figure 3-10.
Apparent Rate of Heat Release versus Crankangle for Various Fuels at 1000 RPM, Equivalence Ratio of 0.5, Injection Timing of 25°BTDC and Intake Air Temperature of 51.7°C

spike in the burning rate curve is about the same for the 20% emulsion and the neat diesel fuels but for the emulsions the spike is slightly retarded. The solution spike is also retarded but is 80% higher than the emulsion spike. The remaining portion of the burning curves are not much different for the three fuels. In considering the physical processes that play a role in the initial burning spike it appears that droplet vaporization for the solutions distilled off the more volatile alcohol while the emulsions with the alcohol encapsulated by the diesel fuel vaporized at a rate similar to that of neat diesel fuel. If microexplosions (Lasheras, et al., 1981) were to take place, it would seem that emulsions would give microexplosions more readily than solutions. One might also expect the microexplosions would cause a more radical change in the burning rate than was observed. Figure 3-11 shows maximum pressure and pressure derivative data for the 0.5 equivalence ratio and 22° timing with various amounts of alcohol. The combustion for the 30% and 40% alcohol solutions was quite harsh and gave audible knock. The data shown in Figs. 3-8 and 3-11 were taken on different days so that some differences in the data are evident.

Figure 3-11 also shows indicated thermal efficiency. Again the differences between emulsions and solutions are small and within measuring accuracy. A slight rise in thermal efficiency with alcohol addition was noticeable for both solutions and emulsions.

Emissions

Figure 3-12 shows that a 20% alcohol emulsion reduced nitrogen oxides more than a 20% solution. The reductions and differences were greatest at advanced injection timing and higher load. Part of the difference between the emulsions and solutions may be the presence of the 1% water in the emulsions which would lower the NO_x. Recall that the solutions gave a higher premixed spike which could cause higher NO_x.

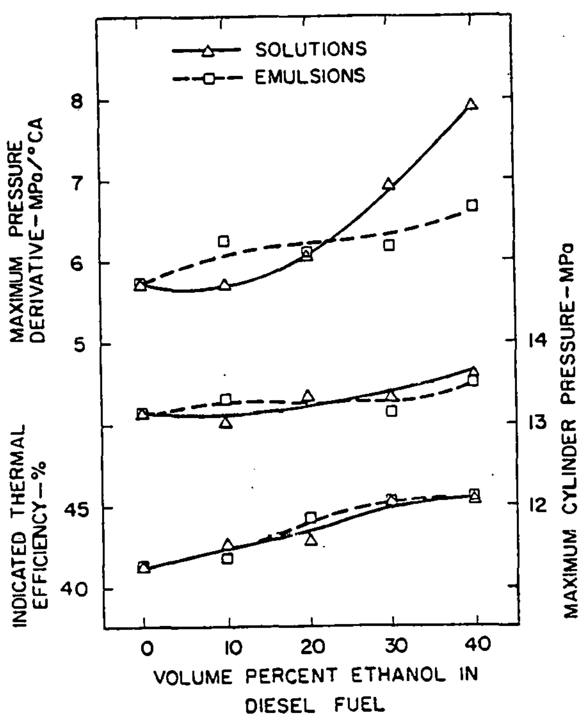


Figure 3-11.
Indicated Thermal Efficiency, Maximum Cylinder Pressure and Maximum Pressure Derivative versus Volume Percent Ethanol in Diesel Fuel for 1000 RPM at an Equivalence Ratio of 0.5 and Injection Timing of 22°BTDC

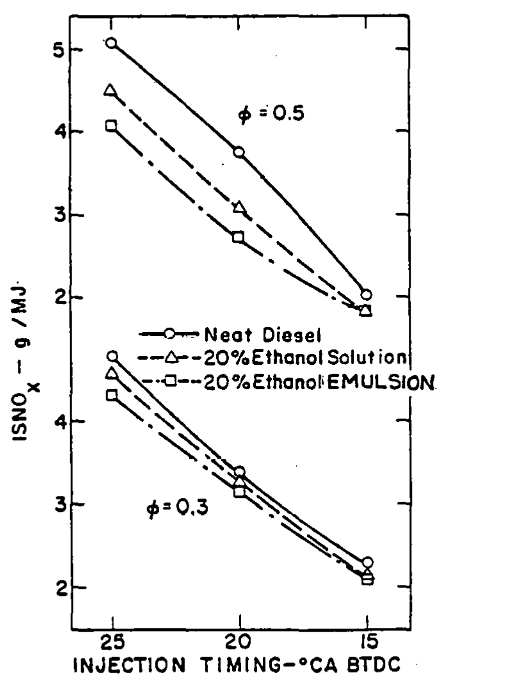


Figure 3-12.
Indicated Specific NO_x Emissions versus Injection Timing for 1000 RPM, Overall Equivalence Ratios of 0.3 and 0.5 and Various Fuels

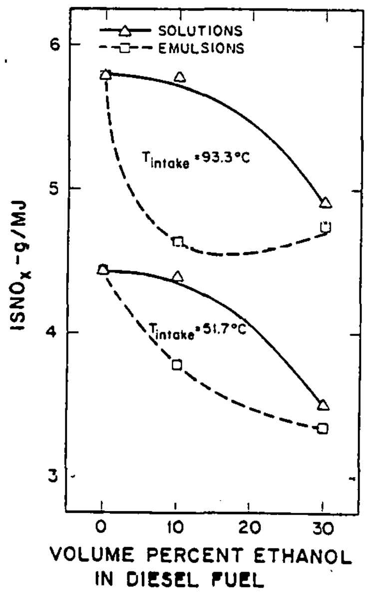


Figure 3-13.
Indicated Specific NO_X Emissions versus Volume Percent Ethanol in Diesel Fuel for 1000 RPM at an Equivalence Ratio of 0.3 and Injection Timing of 22°BTDC

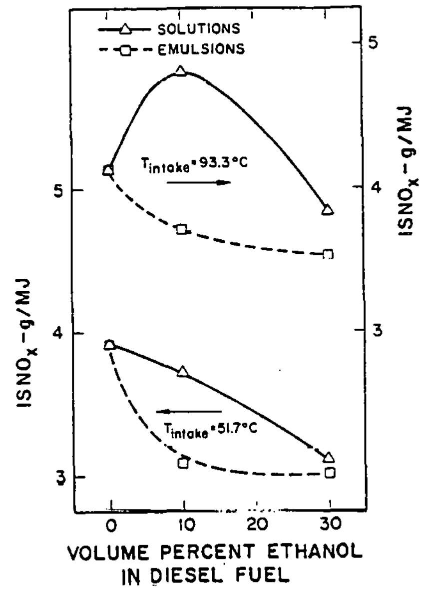


Figure 3-14.
Indicated Specific NO_X Emissions versus Volume Percent Ethanol in Diesel Fuel for 1000 RPM at an Equivalence Ratio of 0.5 and Injection Timing of 22°BTDC

The differences between the effects of solutions and emulsions on NO_X is even more apparent in Figs. 3-13 and 3-14 which show that the emulsions lowered the NO_X while the solutions had much less effect and actually increased the $ISNO_X$ at the higher load and higher intake temperature. Recall again that the emulsions increased the ignition delay but did not increase the premixed burning very much while the solutions gave a much larger amount of premixed burning.

Bosch smoke numbers were high for this engine and were only slightly reduced by the alcohol. Figure 3-15 shows the trends. The differences between the solutions and emulsions is probably not significant. At an advanced timing of 30° BTDC, solutions had a larger influence in decreasing smoke than did emulsions.

Unburned hydrocarbons increased (Fig. 3-16) dramatically with addition of alcohol but the increase was less pronounced at the higher percentages. This may indicate that alcohol-air

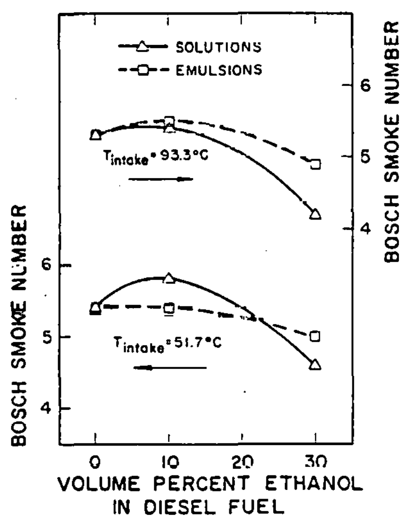


Figure 3-15.
Bosch Smoke Number versus Volume Percent Ethanol in Diesel Fuel and 1000 RPM at an Equivalence Ratio of 0.5 and Injection Timing of 22°BTDC

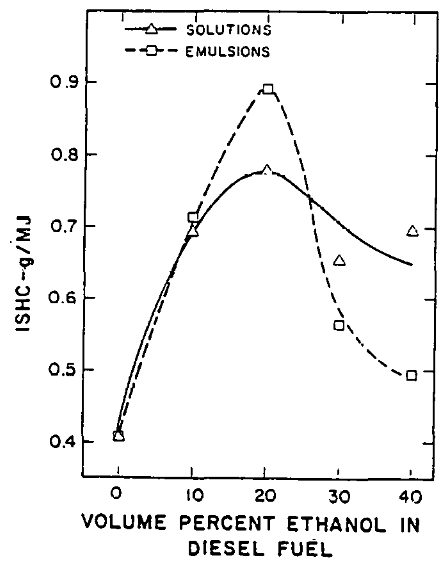


Figure 3-16.
Indicated Specific Unburned Hydrocarbon Emissions versus Volume Percent Ethanol in Diesel Fuel for 1000 RPM at an Equivalence Ratio of 0.5 and Injection Timing of 22°BTDC

mixtures formed which were too lean to burn, but that at the higher substitution values, the fuel-air ratio of some of these mixtures was higher. Another factor is the insensitivity of the FID to aldehydes which may mean that the actual hydrocarbons were higher at the larger volume percentages of ethanol, but were not measured.

Figure 3-17 shows the effects of alcohol on ISCO. The large increase in CO found for the solutions may again indicate some partial oxidation of very lean mixtures.

Injector Wear

Very severe injector failure was found when using the higher percentage emulsions. When a failed injector was tested on the pop-off tester, the nozzle reached about one-half to two-thirds of its opening pressure and simply squirted fuel with no atomization. The cause of the failure appeared to be grooves which appeared on the needle, especially in the region where the needle seats. Table 3-4 shows the data on injector life and as can be seen, the solutions caused no problem. It should be noted that other investigators (Lawson et al., 1981) have run injectors with alcohol added to the injection pump gallery and have not experienced such failures. Thus poor emulsion quality does not seem to be the likely cause of failure.

4. FUMIGATION OF A TURBOCHARGED DIRECT INJECTION DIESEL

As discussed in the technology assessment, fumigation appears to be a viable means of retrofitting diesel engines to use alcohol. It was anticipated that the alcohol substitution percentage would be knock limited at high loads and quench limited at light loads. The objectives of this work were to determine the exact limits of alcohol substitution, their dependence on alcohol proof and to examine the details of combustion (ignition delay, rate of pressure rise, peak pressure, energy efficiency and emissions) while fumigating ethanol in the engine.

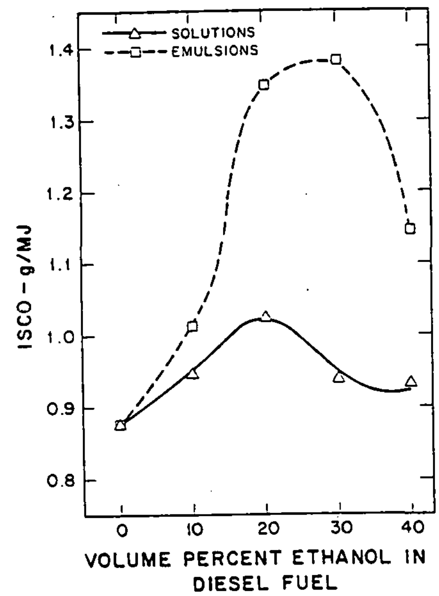


Figure 3-17.
Indicated Specific Carbon Monoxide Emissions versus Volume Percent Ethanol in Diesel Fuel for 1000 RPM at an Equivalence Ratio of 0.5 and Injection Timing of 22°BTDC

Table 3-4.

TESTS RUN ON TACOM SINGLE CYLINDER DIESEL USING ROOSA MASTER 4 HOLE, "PENCIL NOZZLE", MODEL 20939

	NEAT DIESEL		SOLUTIONS				EMULSIONS				
	Without Hydroshear & Pulsafeeder Pump	With Hydroshear & Pulsafeeder Pump	% Ethanol by Volume				% Ethanol by Volume (1% Water by Volume)				% Ethanol by Vol (5% Water by Vol.)
			10	20	30	40	10	20	30	40	20
Normally Aspirated	S	S	S	S	S	S	ND* (S)	ND* (S)	ND* (S)	ND* (S)	ND
Supercharged	S	ND	S	S	S	S	U	UU	UUU	UUU	UU

SYMBOLS AND ABBREVIATIONS

ND = Not Determined
S = Satisfactory, nozzle life is greater than 20 hours
U = Unsatisfactory, nozzle life is less than 20 hours
UU = Unsatisfactory, nozzle life is less than 10 hours
UUU = Unsatisfactory, nozzle life is less than 5 hours
ND* = Not enough endurance test data; accumulated hours for all percentages was about 10 hours

Experimental Apparatus

The engine used in this investigation was a 4-cylinder, 4-stroke, turbocharged diesel engine manufactured by J.I. Case Company. Details about the engine are given in Table 4-1. The engine is coupled to an electric dynamometer having a springless weighing scale.

TABLE 4-1.

J.I. Case 336BD Diesel Engine Specifications

General	
Maker	J.I. Case
Type	4 Cylinder, 4 Stroke Cycle, Valve-in-Head Turbo-Charged
Bore	4-5/8 Inches (117.5 mm)
Stroke	5 Inches (127 mm)
Piston Displacement	336 Cubic Inches (5506 cm^3)
Compression Ratio	15.8 to 1
No Load Governed Speed	2330 to 2370 RPM
Rated Engine Speed	2200 RPM
Engine Idling Speed	700 to 750 RPM
Fuel System	
Fuel Injection Pump	Robert Bosch, Type PES Multiple Plunger
Pump Timing	30 Degrees Before Top Dead Center (Port Closing)
Fuel Injectors	Pencil Type (Opening Pressure) 20 MPa (2800 PSI)
Governor	Variable Speed, Fly-Weight Centrifugal Type, Integral Part of Injection Pump

The air flow into the engine was measured by an inclined manometer which was connected to a laminar flow element. Diesel fuel flow rate was measured by an automatic beam balance with digital time indicator and the alcohol flow rate by a calibrated rotameter with constant temperature water circulation. A magnetic pickup and electronic counter system were used to measure engine speed.

An AVL 8QP500-ca quartz pressure transducer was installed in the first cylinder to monitor the cylinder pressure. Care was taken to follow the correct calibrating and operating procedure of the transducer (Lancaster et al., 1975). A pump transducer (AVL 7QP2500) was mounted on the first cylinder fuel line 37.5 mm from the injection pump. Needle lift of the number 1 injector was measured by a Kaman KD-2400 proximity indicator mounted on top of the injection nozzle. Crank-angle marks were generated by a shaft encoder which generated 3600 pulses per revolution. All of the data signals were digitized and recorded by either the lab data acquisition system, as described in the solution and emulsion section, or a Nicolet Explorer III digital oscilloscope.

Smoke data were obtained by means of a EFAW 65A smoke sampling hand pump and EFAW 68A smoke meter. Other emissions, HC, CO, CO_2, NO_x and O_2, were measured as described previously.

Experimental Procedure

Because the engine is governor controlled, it was not possible to fix the rack position, therefore the experimental matrix was determined by speed and load. At the desired speed and load the engine was run on pure diesel until steady state was attained, as indicated by stable oil and coolant temperatures. After taking the data for pure diesel the alcohol supply was turned on in gradually increasing amounts. This caused the engine output to increase so adjustment of the diesel fuel flow rate was necessary. The substitution limits, as determined by knock at high loads and misfire at low loads, were determined audibly.

All of the data on substitution limits were taken at the manufacturer's injection timing, 30° BTDC for the pump plunger movement with needle lift occurring at approximately 18° BTDC. During the investigation into ignition delay some data were taken at a retarded injection timing, pump plunger movement at 24° BTDC, however no limiting substitution data were taken at this retarded timing.

Fumigation Techniques and Data

Three fumigation techniques were investigated. The first technique was a retrofit system that was being considered for marketing by a commercial company. The second was direct injection of the alcohol through an atomizing nozzle into the inlet manifold downstream of the compressor and the third technique introduced prevaporized alcohol into the inlet air upstream of the compressor. As these three techniques were tested sequentially they will be presented sequentially in the text below.

Prototype Aquahol Fumigation System

The prototype commercial system is shown schematically in Fig. 4-1. The concept of the

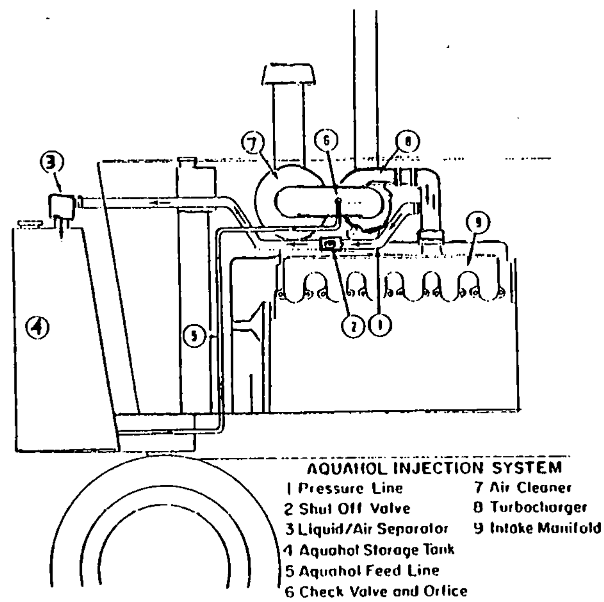


Figure 4-1.
Prototype Commercial Fumigation System Schematic

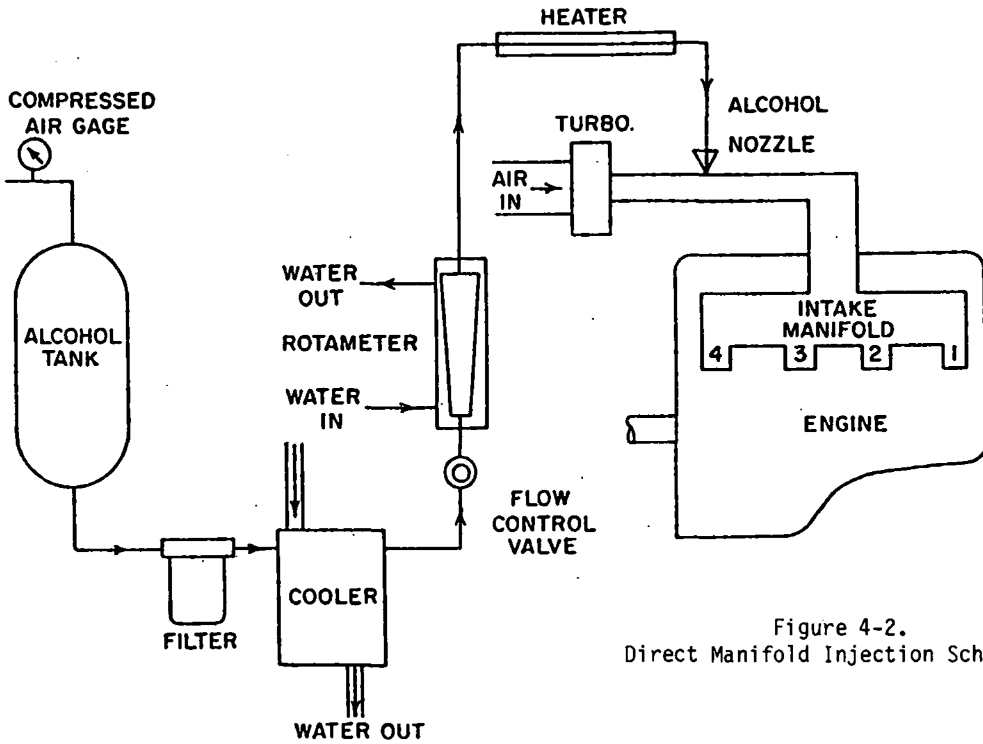


Figure 4-2.
Direct Manifold Injection Schematic

device was to use the pressure generated by the compressor as the driving force for introducing alcohol into the inlet manifold. The objective of the tests run at the U.W.-Madison, to obtain reliable performance data, was never realized. When the system was operating, alcohol was observed pooling in the compressor. This was deemed unsatisfactory operation and the engine was shut down. Attempts were made to improve the alcohol atomization and vaporization to avoid compressor pooling but all failed. No further tests of the commercial system were carried out.

Direct Manifold Injection

To avoid the problem of liquid in the compressor a different technique of introducing the alcohol was attempted. Figure 4-2 shows the schematic of the experimental set-up. The engine is the J.I. Case engine described in Table 4-1 however now the ethanol is injected downstream of the compressor using a Delavan 1/4 J pressure atomizer. Using this technique the data matrix shown in Table 4-2 was completed. The results obtained were similar to those which had been previously put forth in the literature (Pischinger et al. 1980; Cruz et al., 1979). The efficiency at high loads increased slightly with the addition of ethanol, while at low loads it decreased. The ignition delay increased with ethanol addition and at high loads the rate of pressure rise and the maximum pressure increased with ethanol addition. The emission data were also similar to those reported in the literature (Broukhiyan and Lestz, 1981). The Bosch smoke number and NO_x emissions

TABLE 4-2

Matrix of Engine Operating Conditions

Alcohol Proof	Engine Speed RPM	Beam Load lb (N)	Horsepower
160	2100	69.2 (307.8)	96.9
	2100	60.6 (269.5)	84.8
	2100	40.4 (179.7)	56.6
	2100	19.7 (87.6)	27.6
	1500	79.8 (355.0)	79.8
	1500	60.6 (269.5)	60.6
	1500	39.6 (176.1)	39.6
	1500	20.6 (91.6)	20.6
	1100	60.0 (266.9)	44.0
	1100	39.7 (176.6)	29.1
	1100	20.0 (89.0)	14.7
200	2100	69.2 (307.8)	96.8
	2100	60.5 (269.1)	84.7
	2100	39.8 (177.0)	55.7
	2100	20.0 (89.0)	28.0

Note: Beam loads of 70 lb_f (311.4N) and 80 lb_f (355.8N) correspond to full load at 2100 rpm and 1500 rpm, respectively, for this engine.

decreased with the addition of alcohol while the unburned hydrocarbons and CO increased. It was also observed that for this engine and fumigation technique, the proof of the ethanol did not affect the engine efficiency, ignition delay, rate of pressure rise or peak pressure. Figure 4-3 shows the ethanol substitution obtainable at 2100 RPM over the load range of the engine. In this figure the mass percent alcohol is defined as the mass of 200 proof ethanol divided by the mass of diesel plus 200 proof ethanol multiplied by 100. It can be observed that the proof of the ethanol does not affect the substitution percentage at a particular load. The complete set of substitution data is given in Table 4-3.

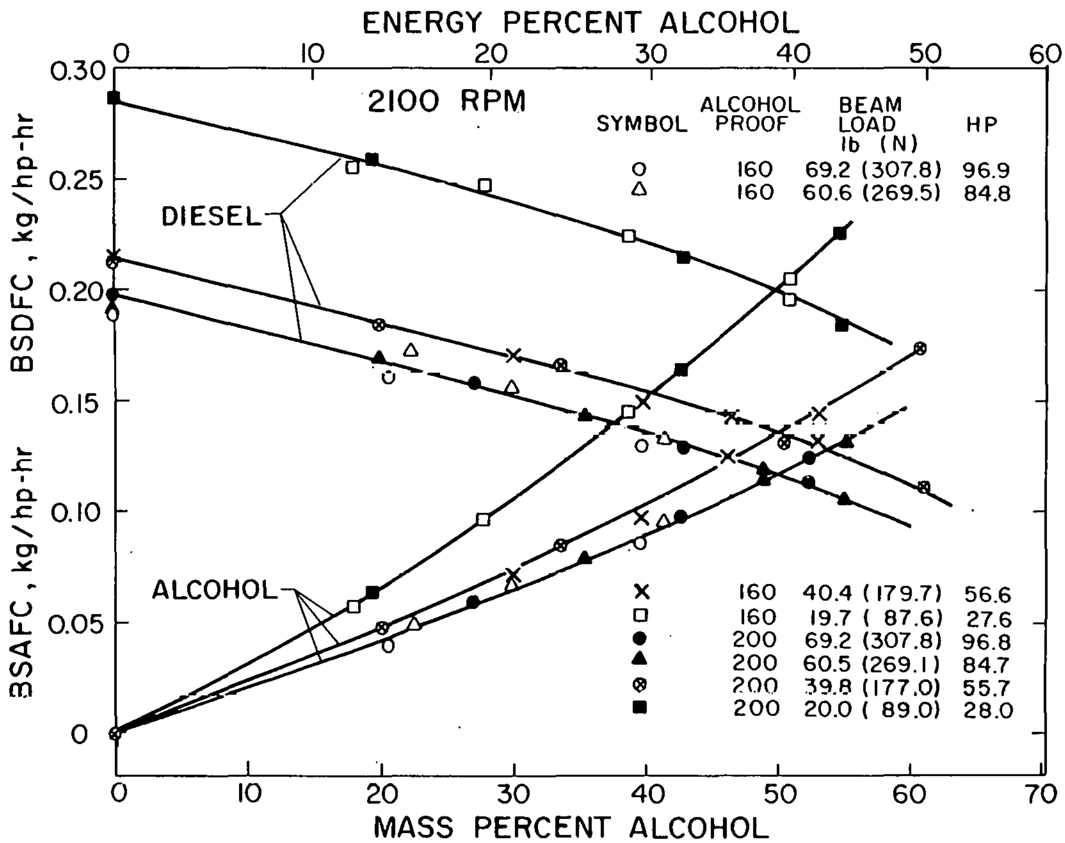


Figure 4-3.
Brake Specific Fuel and Alcohol Consumption versus Percent Alcohol Substition, 160 and 200 Proof Ethanol RPM

TABLE 4-3

Engine Performance with Alcohol Fumigation

Alcohol Proof	Engine RPM	H.P.	Maximum Mass % Alcohol	Maximum Energy % Alcohol	Diesel Fuel Saving, %	Change in Energy Efficiency, %	Relative[1] Peak Pressure	Relative[2] Maximum Pressure Rise Rate
200	2100	96.84	52.2	41.0	43.5	+2.8	1.11	2.98
		84.67	55.4	44.0	46.2	+3.7	1.16	2.49
		55.69	60.9	50.0	48.1	-1.7	--	--
		28.00	55.2	43.6	38.0	-10.2	--	--
160	2100	96.90	39.6	29.3	31.7	+2.2	1.13	1.04
		84.83	41.5	31.0	34.0	+2.5	1.08	1.80
		56.60	53.5	42.0	41.9	+1.7	--	--
		27.60	51.1	39.6	33.9	-5.1	--	--
	1500	79.84	39.7	29.3	26.6	+3.0	--	--
		60.60	42.6	31.9	36.6	+3.0	--	--
		39.60	60.3	48.9	45.4	-2.5	--	--
		20.60	53.7	42.3	33.2	-10.3	--	--
	1100	44.0	42.1	31.6	33.7	+5.6	--	--
		29.1	37.3	26.8	28.8	+0.3	--	--
		14.7	49.9	39.0	35.1	-4.1	--	--

[1] Maximum cylinder pressure with alcohol/maximum cylinder pressure with no alcohol.

[2] Maximum pressure rise rate with alcohol/maximum pressure rise rate with no alcohol.

In an attempt to determine the uniformity of the ethanol distribution from cylinder to cylinder the exhaust temperatures were measured at each of the exhaust ports. Figure 4-4 shows the exhaust temperatures at four different loads as a function of alcohol substitution percentage. Because one of the thermocouples broke, data were obtained for only three cylinders. It is apparent from Fig. 4-4 that the cylinder to cylinder distribution of ethanol was poor.

The final method of fumigation attempted was designed to overcome the problem of cylinder-to-cylinder distribution of the ethanol. This method will be discussed below. Further discussion of the results of the liquid injection tests is given by Chen et al. (1981).

Vaporized Ethanol Introduction

In an attempt to overcome the alcohol distribution problem a vaporizer was built and installed on the engine. A schematic of the vaporizer is shown in Fig. 4-5. The superheated alcohol vapor was introduced into the intake manifold through a tee immediately before the inlet of the turbocharger. With the vaporizer installed, all of the exhaust temperatures were within 5°C of one another and decreased uniformly with increasing ethanol substitution percentage. From these data it was interpreted that the cylinder-to-cylinder distribution problem had been solved.

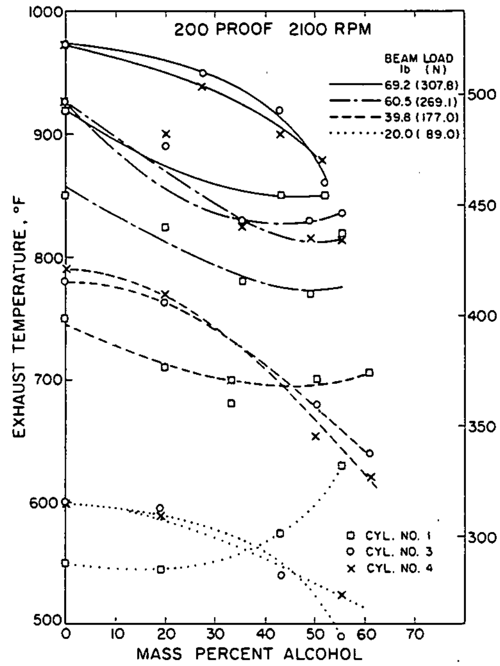


Figure 4-4.
Exhaust Temperatures of the Different Cylinders versus Percent Alcohol, 200 Proof Ethanol, 2100 RPM

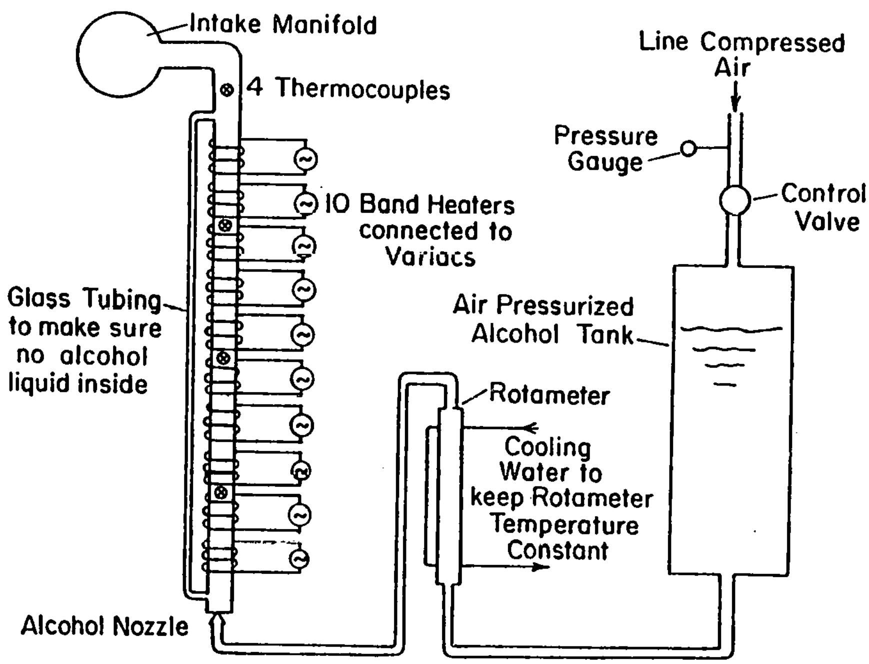


Figure 4-5.
Schematic of the Alcohol Vaporizer

Figure 4-6 shows ignition delay versus alcohol percentage over the entire load range at 1500 RPM. Data are shown for both the vaporized alcohol introduction and post compressor liquid injection into the manifold. The upper curves show the ignition time as depicted by the start of the rapid pressure rise and the lower curve shows the start of injection as depicted by 10 percent of the maximum needle lift. Therefore the true ignition delay would be the difference between the top curves and bottom curves. From Fig. 4-6 two observations are immediately apparent. First, alcohol increases the ignition delay, which is already well documented; however the increase is not as large as it initially appears from inspection of the pressure traces. Second, a significant portion of the ignition delay appears to be associated with vaporizing the alcohol in the cylinder, as shown by the large ignition delays for the liquid alcohol injection.

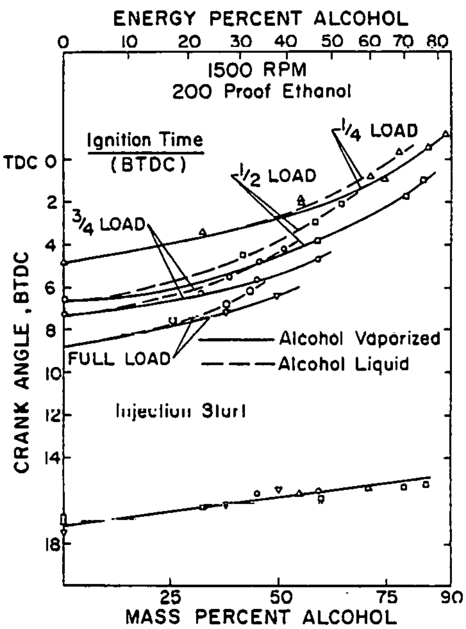


Figure 4-6.
Crankangle at Start of Injection and Ignition versus Alcohol Substitution Percentage, 200 Proof Ethanol, 1500 RPM, Vaporized and Liquid Injection

The peak pressure and rate-of-pressure-rise for engine operation with and without the vaporizer are shown in Figs. 4-7 and 4-8. It is observed that vaporizing the alcohol decreases the peak pressure and rate-of-pressure-rise relative to operation with liquid manifold injection. Also shown on the figures are the two commonly observed limiting criteria, instability at low loads and knocking at high loads. In all cases, vaporizing the alcohol extended the maximum possible alcohol substitution percentage. It is also interesting to note that at one-half load the peak pressure in cylinder number one increased with liquid manifold injection and decreased with vaporized operation, Fig. 4-7. The unstable operation at one quarter load is believed to be due to the low

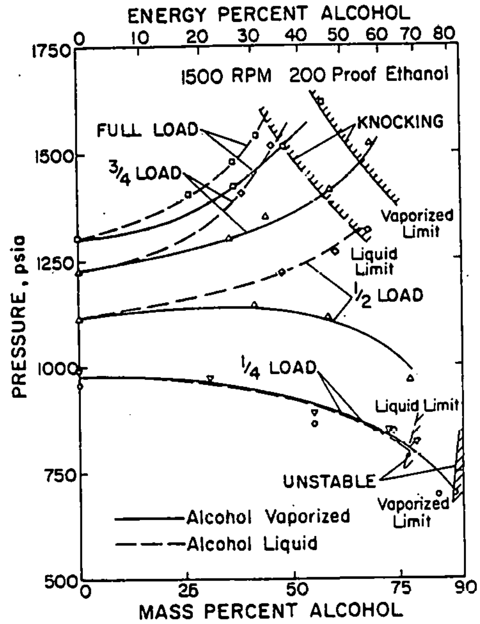


Figure 4-7.
Peak Cylinder Pressure versus Ethanol Percentage, 200 Proof Ethanol, 1500 RPM, Vaporized and Liquid Injection

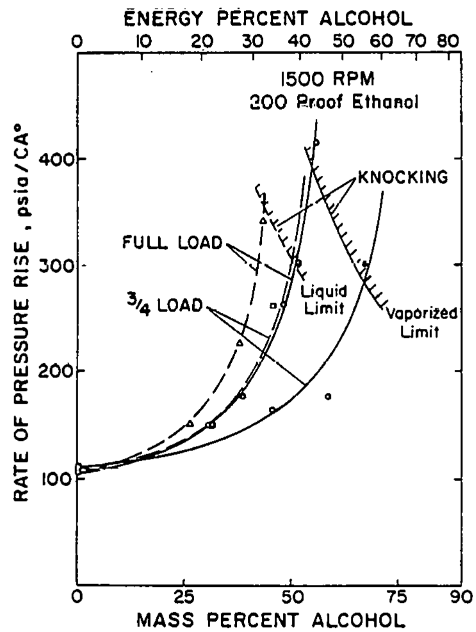


Figure 4-8.
Maximum Rate of Pressure Rise versus Ethanol Percentage, 200 Proof Ethanol, 1500 RPM, Vaporized and Liquid Injection

quality of the injection of the small quantity of diesel fuel used to ignite the ethanol air mixture.

Emission data for the vaporized ethanol showed the same trends as for the liquid direct manifold injection. Operation at light loads with over 80 percent energy substitution would probably be precluded in this engine by the excessive hydrocarbon emissions (>7000 ppm). However it is felt that obtaining smooth engine operation at such a high percentage of ethanol substitution is significant. Murayama et al. (1982) were also able to obtain up to 80 percent energy substitution in their I.D.I. diesel engine by reducing the pre-chamber to clearance volume ratio.

If the design specification for peak cylinder pressure is used to determine the maximum alcohol substitution, the maximum substitution remains approximately the same. For the engine used, peak pressures in excess of 1600 psi are not recommended. Under full load conditions this limit is reached at the same time that knocking begins; as long as knocking is avoided the design specifications of the engine are not exceeded.

Figures 4-9 and 4-10 show the thermal efficiency of the engine with the vaporizer unit working. The efficiency is calculated considering the internal energy required to vaporize the alcohol, not the total energy input into the vaporizer. That is, the effectiveness of the vaporizer is not considered. The trends observed are similar to those of previous data; there is a slight increase in thermal efficiency as the alcohol percentage is increased at high loads and a decrease with alcohol substitution at low loads.

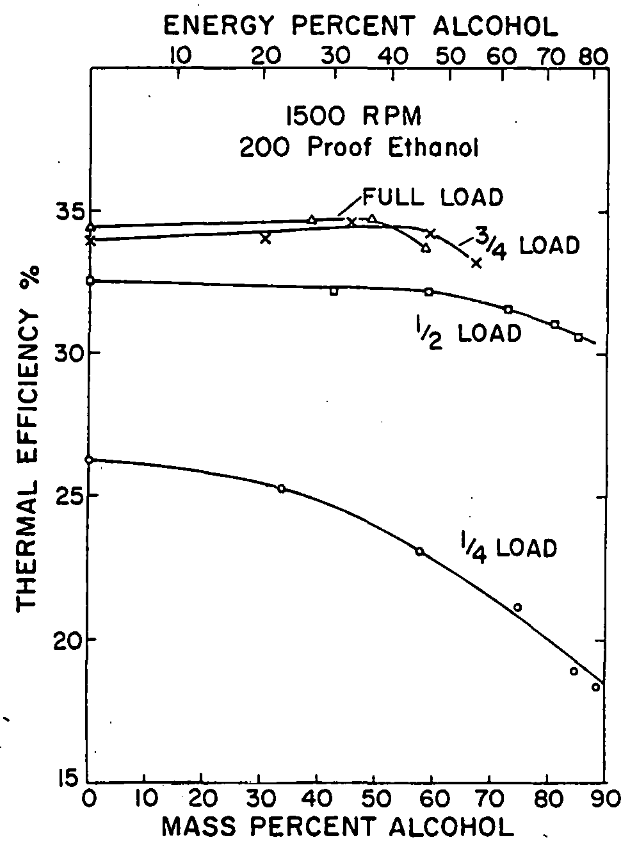


Figure 4-9.
Thermal Efficiency versus Alcohol Substitution Percentage, 1500 RPM, 200 Proof Ethanol

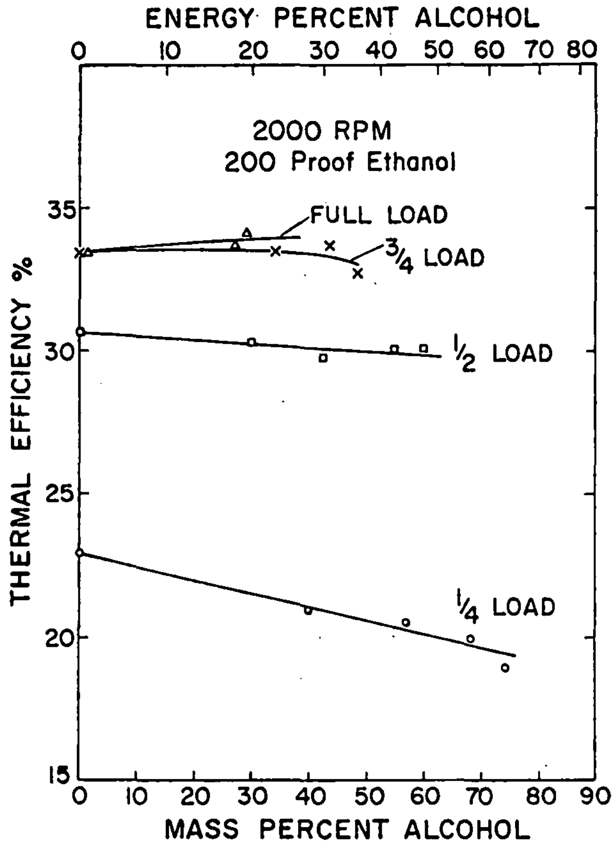


Figure 4-10.
Thermal Efficiency versus Alcohol Substitution Percentage, 2000 RPM, 200 Proof Ethanol

No data were taken on engine wear with alcohol use. This is currently a question of great concern. From the information available in the literature, it appears that wear is most severe under transient operating conditions and is more severe if liquid alcohol is present in the combustion chamber. Since alcohol was introduced only under steady operating conditions and for the most part only in the vapor phase, one would expect the engine wear to be minimal. We did not have a single failure which could be related to the use of alcohol in the engine. However it should again be emphasized that we were operating the engine in the best possible wear-avoiding conditions.

Prevaporized Lower Proof Ethanol

To further examine the effects of aqueous ethanol on engine performance a water vaporizer, similar to the alcohol vaporizer shown in Fig. 4-5, was constructed and installed in parallel with the ethanol vaporizer. An additional pressure transducer was installed in cylinder number 3 to check the cylinder-to-cylinder distribution. In addition, implicit information regarding the distribution was obtained from exhaust temperature

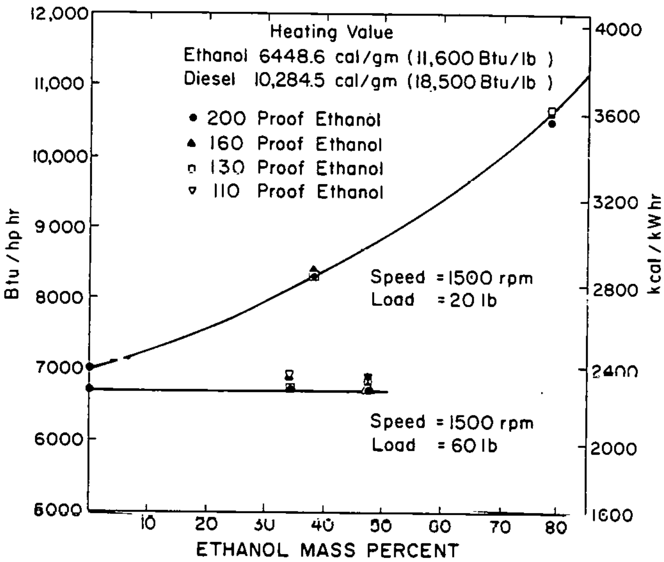


Figure 4-11.
Specific Energy Consumption versus Ethanol Mass Percent, 1500 RPM, 20 lb and 60 lb Beam Load 200, 160, 130 and 110 Proof Ethanol

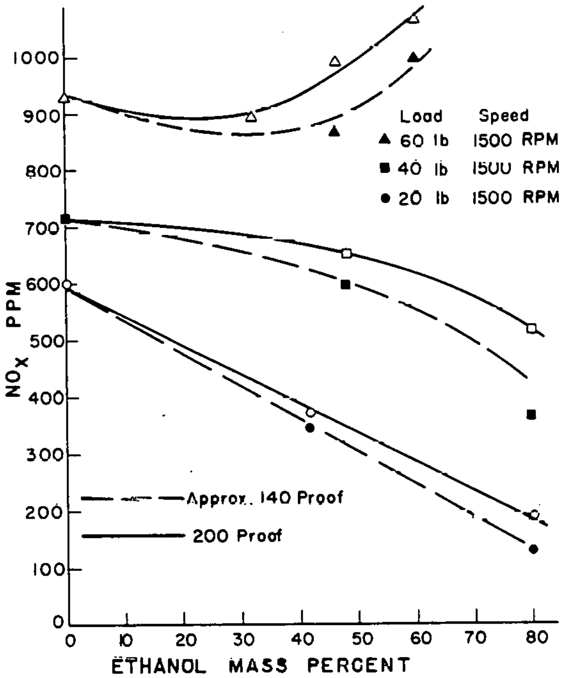


Figure 4-12.
NO_x Concentration versus Ethanol Mass Percent, 1500 RPM, 20 lb, 40 lb and 60 lb Beam Load, 200 and aprox. 140 Proof Ethanol

measurements. Because of the limited time only a small experimental matrix was investigated. The engine speed was set at 1500 RPM, and the beam load was adjusted to give power outputs of 14.9, 29.8 and 44.7 kW (20, 40 and 60 HP) respectively. At each load and speed aqueous ethanol was progressively added. The proofs tested were 160, 130 and a few data points at 110 proof and at each operating point NO_x, unburned hydrocarbons (FID) and cylinder pressure data were taken. As had been found in our previous work with liquid manifold injection, Chen et al. 1981, the proof of the alcohol had little affect on the engine operation in terms of unburned hydrocarbons, efficiency, peak pressure and rates of pressure rise. Figure 4-11 shows the engine's energy efficiency versus mass fraction of neat and aqeuous ethanol at 1500 RPM and two different loads. Within the repeatability of the data, the ethanol proof has no effect on the engine efficiency. Graphs of unburned hydrocarbons, peak pressure, exhaust temperatures and heat release also showed an absence of effects due to different proofs of ethanol. The only significant change that was observed in our testing was a decrease in NO_x emissions with decreasing proof. This is shown in Fig. 4-12.

Comparison of the pressure traces of number one and number three cylinder indicated that the combustion in the two cylinders was very similar. Under the worst conditions, maximum ethanol substitution at light loads, the peak cylinder pressures differed by only four percent.

Discussion of the Fumigation Data

To better understand how fumigation changes the combustion, an analysis of the ignition delay phenomena and a heat release analysis were carried out.

Ignition Delay

Three techniques were used to define the ignition delay period. The proximity transducer on the injector indicated the start of injection. The start of combustion was determined either by a rise in the pressure, above the normal motored trace; the change in the ratio of specific heats, obtained from a log-log plot of pressure and cylinder volume; or a sharp change in the heat release rate, obtained from the heat release model. A comparison of these three techniques is shown in Fig. 4-13. From the figure it is observed that the sudden change in the heat release rate and the change in the polytropic exponent occur at the same crankangle. The rapid change in slope of the pressure trace is retarded from the change in heat release rate and polytropic exponent by less than one crankangle degree. It was concluded that all methods gave comparable results.

As stated previously the basic trend of the data is an increase in ignition delay with increased alcohol mass fraction, but there were two important exceptions. At high load and high speed, which results in high turbocharging temperature and pressure, the ignition delay decreased with increased alcohol mass fraction and at retarded injection timing the ignition delay also decreased with increased alcohol substitution.

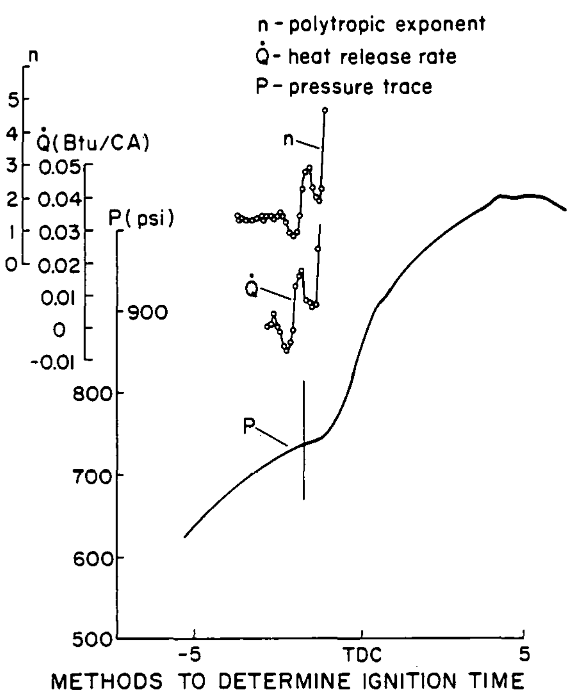


Figure 4-13.
Comparison Between Different Indicators of Start of Combustion; n, the Polytropic Exponent; Q̇, the Heat Release Rate; P, the Pressure Trace Method

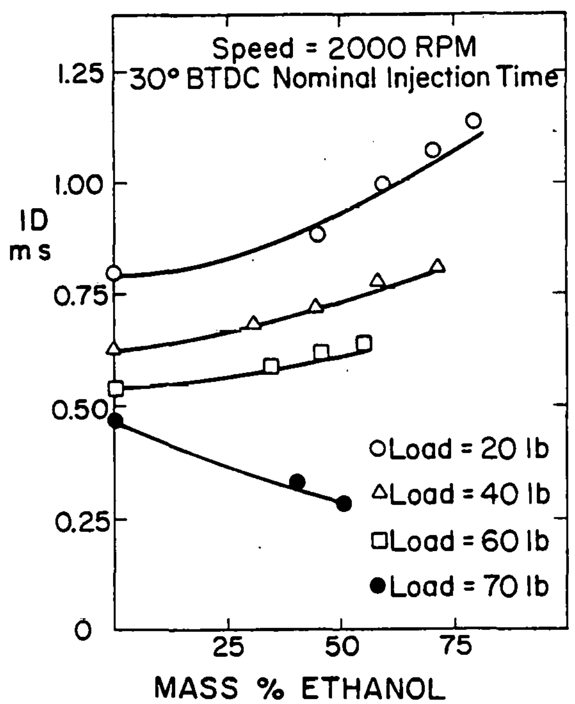


Figure 4-14.
Ignition Delay versus Mass Percent Ethanol, 200 Proof Ethanol, 2000 RPM, Nominal Injection Timing (30°BTDC)

The ignition delay results at normal injection timing and 2000 RPM are shown in Fig. 4-14.

There are three reasons why ethanol fumigation would change the ignition delay.

1) Changes in inlet conditions due to evaporative cooling.
2) Decreases in the compression pressures and temperatures due to decreases in the mixture's specific heat ratio with increased ethanol substitution.
3) Chemical effects of the alcohol in the cylinder.

As the ethanol substitution percentage is increased phenomena 1 and 2 will result in lower temperatures and pressures at the start of injection, which would tend to increase the ignition delay. Figure 4-15 shows the decrease in temperature and pressure at the point of injection for different mass percentages of ethanol addition.

At high speed and heavy load the turbocharger is operating at a maximum boost condition. The turbocharger will increase the inlet pressure and temperature over the naturally aspirated condition, with the increase being a maximum at high speed and load. This is in opposition to the effects of alcohol fumigation, which are shown in Fig. 4-15. The fumigated alcohol may be viewed as undergoing an induction period during the compression stroke, which given sufficient time would lead to autoignition. To understand the ignition delay we must then know at what stage of its preignition induction process the alcohol is in when the diesel injection occurs. At the heavy load, high speed operating point the compression temperatures are higher so the alcohol will be further along in the preignition reaction sequence then it would be at lighter loads and slower speeds. It seems reasonable that if the alcohol were on the verge of autoignition when the diesel fuel were injected, then the ignition delay would be very short.

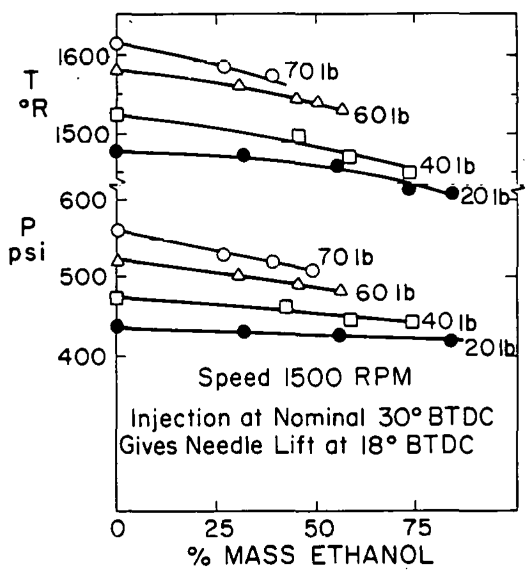


Figure 4-15.
Pressure and Temperature at the Start of Ignition versus Ethanol Percentage, 200 Proof Ethanol, 1500 RPM, Nominal Injection Timing (30°BTDC), Timing and 2000 RPM is shown in Fig. 4-14

To test this hypothesis the engine was run at full load, 2000 RPM with approximately 50% (mass) alcohol substitution and then the diesel fuel was suddenly shut off for one cycle. It was observed that the alcohol in the cylinder did autoignite under the above conditions and that the point of ignition was slightly retarded from the previous cycle where diesel fuel had been injected. The results of this test are shown in Fig. 4-16.

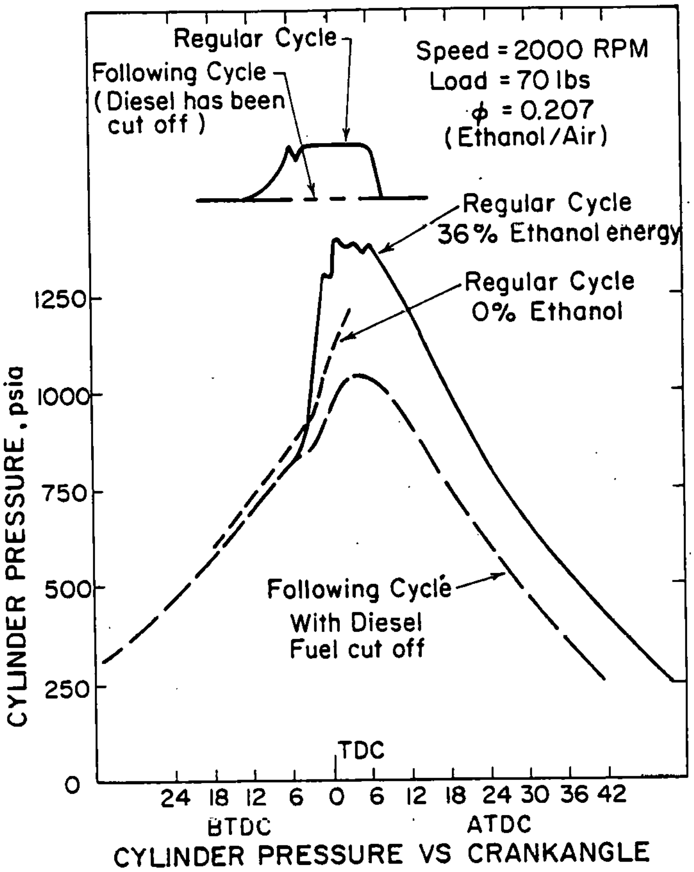


Figure 4-16.
Pressure Traces for Fumigated Cycle and Diesel Cut Off Cycle, 200 Proof Ethanol, 2000 RPM, 70 lb_f Beam Load, Ethanol-Air Equivalence Ratio of 0.207

From these data it was concluded that alcohol is undergoing preignition reactions during compression and, under the proper operating conditions, ignition of the diesel fuel can be enhanced. Retarding the injection timing gives more time for the preignition reactions to progress and therefore can enhance the diesel fuel ignition also. Similar experiments were run under conditions where the alcohol addition increased the ignition delay and it was observed that no autoignition occurred. The general conclusion is that unreacted alcohol vapor will retard ignition while the partially reacted alcohol may help it. The three important factors are length of induction period, temperature and density.

Attempts were made to obtain a suitable ignition delay formula for the alcohol fumigated diesel operation. The basic form of the expression used is

$$\text{I.D.} = A\exp(E/RT)/p^n \quad [\text{msec}]$$

where

A = Constant related to engine geometry
E = Global activation energy, [cal/gmole]
n = Exponent; global reaction order
T_2 = Temperature, time averaged over the delay period, [K]
p = Pressure, time averaged over the delay period, [atm].

Ignition delay formulas of this form have been presented by many researchers for many different conditions. We chose to modify the expression for dual fuel operating into the following form:

$$\text{I.D.} = AP^{-(M_1 n_1 + M_2 n_2)}\exp(M_1E_1/T + M_2E_2/T)$$

or

$$\text{I.D.} = AP^{-(n_1 + M_2\Delta n)}\exp(E_1/T + (M_2\Delta E/T))$$

with

M_1 = Mole Fraction of Fuel 1
M_2 = Mole Fraction of Fuel 2

$$\Delta E = E_2 - E_1 \qquad \Delta n = n_2 - n_1$$

This form of the equation needed yet another modification. The activation energy of the alcohol at the time of the injection of the diesel fuel is not a property of the fuel; it depends on the temperature and pressure history of the compression process prior to injection, i.e., the engine operating condition. The final modified form of the expression is:

$$\text{I.D.} = AP^{-(M_d n_d + M_{al} n_{al})}\exp[(M_d E_d + M_{al} E_{equ})/T]$$

where

E_{equ} = Equivalent activation energy and is basically a function of temperature and pressure
M_d = Mole fraction of diesel
M_{al} = Mole fraction of alcohol

The final form of the equation is then

$$\text{I.D.} = 1.308\ P^{-(M_d \cdot 0.700 + 0.414\ M_{al})}$$

$$\exp(1965\ M_d/T + M_{al}\ E_{equ}/T)$$

with

$$E_{equ} = 872 - (14.976T - 0.0285T^2 + 1.423(T/100)^4)M^2_{al}$$

This equation gives an accurate correlation with our data of the ignition delay versus alcohol substitution percentage at different engine operating conditions, as shown in Fig. 4-17.

Our efforts to model the ignition delay stopped at this point. It is felt that additional analysis is needed on a more fundamental level to fully understand the interactions, both chemical and physical, during the ignition process between the two different fuels.

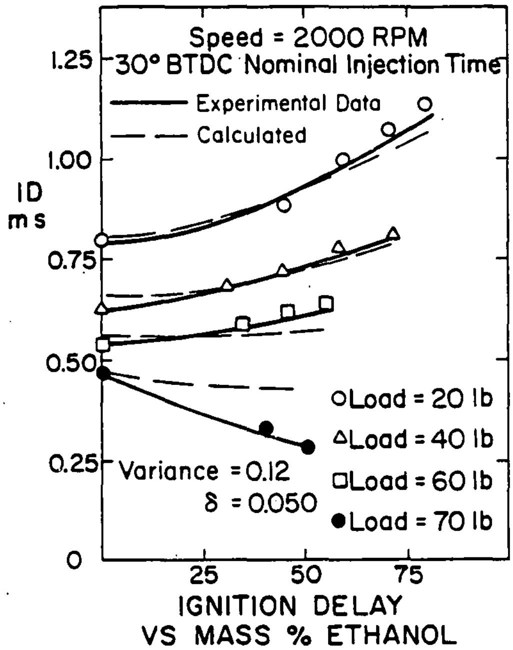


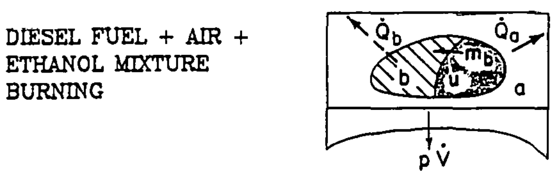
Figure 4-17.
Comparison of Experimental Ignition Delay and Ignition Delay Formula. Experimental Conditions 200 Proof Ethanol, 2000 RPM, Normal Injection Timing (30°BTDC)

Heat Release Analysis

A heat release analysis of the fumigation data was carried out using a three zone first law model. The experimental pressure data which is an input to the model was averaged over 40 cycles and then smoothed with a spline fit so the calculated time derivatives change smoothly and continuously. As in other heat release analyses the pressure is considered uniform throughout the cylinder and the gases are assumed to behave ideally. The first law rate equations and mass conservation equations were solved using a fourth order Runge-Kutta routine.

The cylinder gases are modelled as three

BURNING IN THREE ZONE MODE



Q_a = convection
Q_b = radiation
Zone "a" = air + ethanol
Zone "b" = products
Zone "u" = diesel fuel + ethanol + air

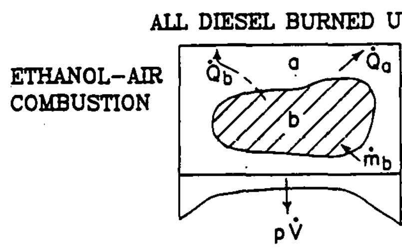
Figure 4-18.
Schematic of the Three Zone Heat Release Model

zones, Fig. 4-18:
1) A diesel fuel plus air plus ethanol zone
2) An air plus ethanol zone
3) A burned zone.

The diesel-alcohol-air mixture, with an assumed equivalence ratio of 1.0, is assumed to react first. The instantaneous equilibrium concentration of chemical species was obtained from an equilibrium subroutine (Olikara and Borman, 1975). Heat is transferred from the gas to the walls by both convection and radiation. The alcohol air zone is assumed transparent to radiation and the only heat exchange with the wall is by convection while the burned zone transmits energy only by radiation. The unburned zone is assumed adiabatic. Heat and mass transfers between zones are neglected with the exception of mass transfer between the unburned and the burned zone.

After the diesel-alcohol-air zone has completely reacted the model becomes a two zone model of air-alcohol mixture and burned equilibrium products (Fig. 4-19).

BURNING IN TWO ZONE MODE



$$\dot{Q}_a = \tilde{h}A\ (T_a - T_w)$$
$$\dot{Q}_b = \epsilon\sigma A\ (T_b^4 - T_w^4)$$

Figure 19.
Schematic of the Two Zone Heat Release Submodel

The equivalence ratio in the diesel-alcohol-air zone was arbitrarily chosen to be one. A sensitivity analysis of the model to this assumption was carried out with equivlanece ratios of 0.8, 1.0 and 1.2. The results are shown in Fig. 4-20. As can be seen from the figure, the heat release rate (HRR) is not very sensitive to the choice of equivalence ratio.

Figure 4-21 shows the effect of ethanol fumigation on the temperature history of the burned and air-alcohol zones. The temperature of the burned zone, T_b, is lowered with the addition of ethanol because less diesel fuel burns. The temperature of the alcohol-air zone, T_a, is lowered because the ratio of specific heats is lower for the air-alcohol mixture then it is for just air.

Figure 4-22 shows the effect of ethanol fumigation on the heat transfer during combustion. As seen from the figure, in the model, fumigation reduces both convection and radiation heat transfer. The decreases are caused by the lower temperatures of the zones a and b, and changes in the emissivity with changes in the change density (Annand, 1963).

The effects of ethanol mass fraction on the heat release rate are shown in Figs. 4-23, 4-24,

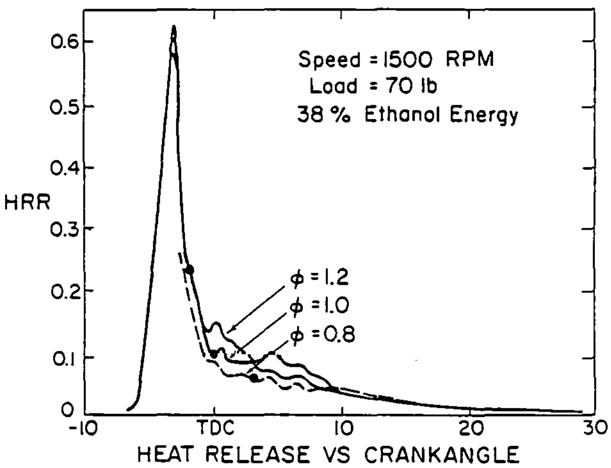


Figure 4-20.
Comparison of Predicted Heat Release Rates with Different Unburned Zone Equivalence Ratios

The Dots Denote the Crankangle at Which the Model Predicts the Unburned Zone, u, Has Zero Value; the Point Where the Model Makes The Transition From the Three Zone Model To The Two Zone Model.

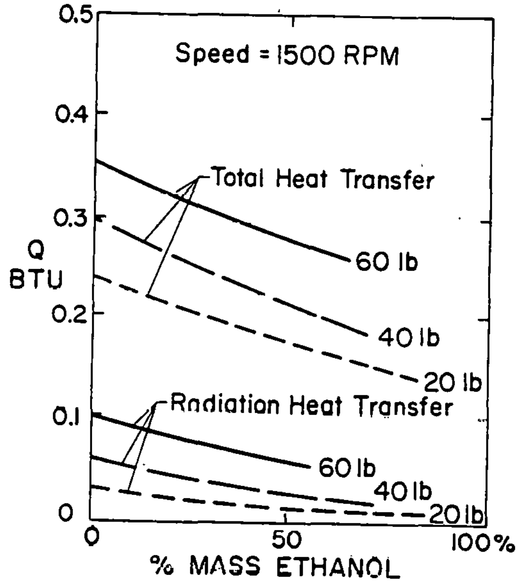


Figure 4-22.
Predicted Effect of Fumigation on Combustion Heat Transfer, 200 Proof Ethanol

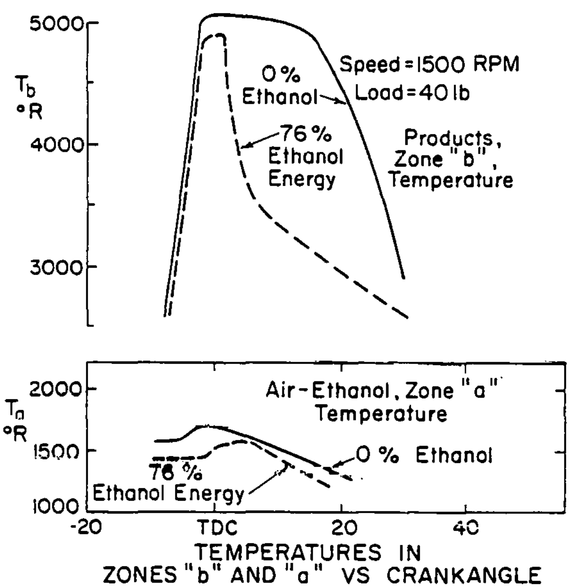


Figure 4-21.
Effect of Ethanol Fumigation on the Temperature History of the Burned Zone and the Air-Ethanol Zone; 200 Proof Ethanol, 1500 RPM, 40 lb_f Beam Load, Normal Injection Timing (30°BTDC)

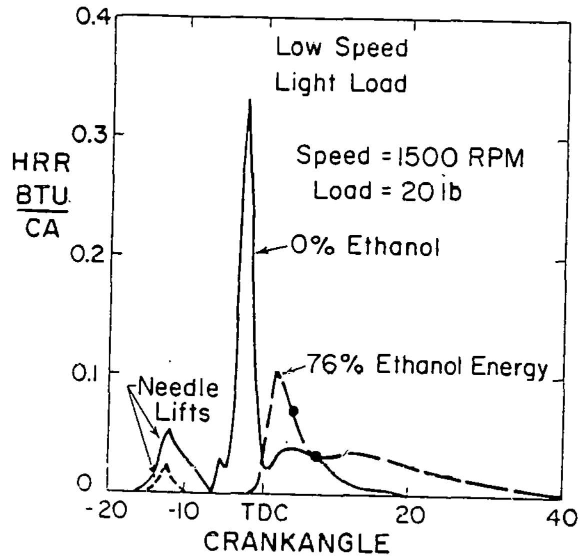


Figure 4-23.
Predicted Heat Rate for Diesel and Fumigated Ethanol, 200 Proof Ethanol, 1500 RPM, 20 lb_f Beam Load, Normal Injection Timing (30°BTDC)

The Dots Denote the Crankangle at Which the Model Predicts the Unburned Zone, u, Has Zero Value; The Point Where the Model Makes the Transition From the Three Zone Model to the Two Zone Model.

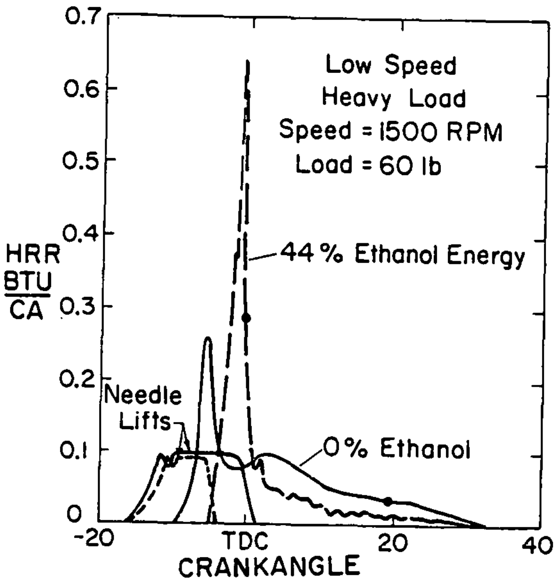


Figure 4-24.
Predicted Heat Release Rate for Diesel and Fumigated Ethanol, 200 Proof Ethanol, 1500 RPM, 60 lb_f Beam Load, Normal Injection Timing (30°BTDC)

The Dots Denote the Crankangle at Which the Model Predicts the Unburned Zone, u, Has Zero Value; the Point Where the Model Makes the Transition From the Three Zone Model to the Two Zone Model.

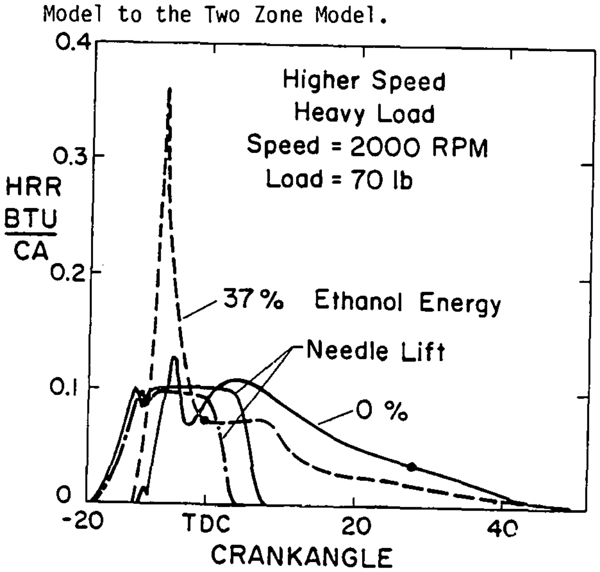


Figure 4-25.
Predicted Heat Release Rate for Diesel and Fumigated Ethanol, 200 Proof Ethanol, 2000 RPM, 70 lb_f Beam Load, Normal Injection Timing (30°BTDC)

The Dots Denote the Crankangle at Which the Model Predicts the Unburned Zone, u, Has Zero Value; the Point Where the Model Makes the Transition From the Three Zone Model to the Two Zone Model.

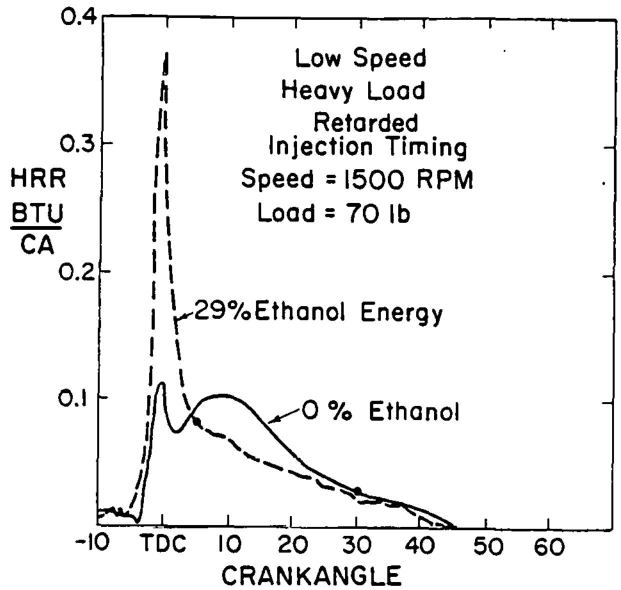


Figure 4-26.
Predicted Heat Release Rate for Diesel and Fumigated Ethanol, 200 Proof Ethanol, 1500 RPM, 70 lb_f Beam Load, Normal Injection Timing (30°BTDC)

The Dots Denote the Crankangle at Which the Model Predicts the unburned Zone, u, Has Zero Value; the Point Where the Model Makes the Transition From the Three Zone Model to the Two Zone Model.

4-25 and 4-26. At light load, Fig. 4-23, it was observed that the ignition delay is large. Most of the diesel fuel will have been injected into the cylinder before ignition. For the case of neat diesel most of the combustion will be premixed and there will be an associated high peak in the heat release rate. Under constant load conditions with fumigated ethanol, less diesel is injected. At the point of ignition the alcohol vapor is still not ready to burn, so the amount of premixed combustion, and subsequently the heat release rate, decreases with increasing ethanol fumigation and the combustion duration increases.

At heavy load and low speed, Fig. 4-24, where the alcohol has caused an increase in the ignition delay the premixed portion of the combustion to increase and the heat release rate also increases. As shown in the figure most of the combustion takes place in the premixed mode. Figure 4-25 shows the heat release for the heavy load, high speed case. This was the operating regime which resulted in shorter ignition delay with increased ethanol fumigation. Under these conditions the alcohol combustion contributes to the premixed heat release spike so the premixed portion of combustion is larger than for neat diesel under the same operating condition, but smaller than that of the heavy load and low speed case. Finally, as shown in Fig. 4-26, retarding the injection timing at low speed has the same effect as increasing the speed in respect to the heat release rate.

Comparison of DI and IDI Fumigation Results

In conjunction with the research work carried out by the University of Wisconsin — Madison, DOE also funded research at Pennsylvania State University which included fumigation of an automotive, IDI diesel engine. The experimental matrices were sufficiently similar that a comparison of results from the two programs was possible.

The engine used at Penn State was a naturally aspirated, indirect injection 1978 Oldsmobile 5.7ℓ V-8 with a 22.5:1 compression ratio. The Wisconsin engine, as described in detail in Table 4-1, was an open chamber turbocharged JI Case engine with a nominal compression ratio of 15:1. It was found that the IDI engine had lower emission levels than did the DI engine. However the emissions underwent larger percentage changes with the addition of alcohol for the IDI than for the DI engine. As the load and speed were increased in both engines their respective percentage changes in emissions with ethanol addition approached each other in value. The ignition delay data was found to behave in exactly the same manner. It was concluded that the primary reason for the larger sensitivity to ethanol addition of the IDI engine was that it had a higher compression ratio. Furthermore, it is believed that the turbocharging of the DI engine acted to effectively increase its compression ratio as the load and speed were increased. With the DI engine's effective increase in compression ratio the two engines will approach each other in their response to ethanol addition. When the effective compression ratios are approximately equal the differences in the absolute level of emissions are probably due to the difference in engine design. The reasons for these differences were not addressed. Further details of the work can be obtained by referring to the paper by Foster et al., 1982.

Fumigation Conclusions

The immediate conclusions that can be drawn from the three fumigation techniques are:

1) Ethanol fumigation, as a retrofit technique, appears to be a viable method of using alcohol in existing diesel engines.
2) At high loads, ethanol substitution results in a slight improvement in energy efficiency whereas at light loads a small degradation in thermal performance is observed.
3) Vaporizing the alcohol before it is introduced into the inlet manifold appears to overcome the problem of cylinder-to-cylinder distribution of the alcohol-air mixture.
4) The maximum rate of ethanol substitution at high loads was determined by the onset of knocking combustion or excessive peak cylinder pressures. At light loads the maximum possible ethanol substitution rate was determined by the onset of unstable combustion.
5) Emissions of oxides of nitrogen and smoke decreased with ethanol fumigation whereas unburned hydrocarbons increased as compared to diesel fuel operation.

The heat release model shows that fumigation causes a large increase in the initial burning spike even for the cases where the ignition delay is decreased by fumigation. If the assumption that the diesel fuel burns first is correct, then all of the diesel fuel burns in the initial burning spike. These predictions of the model can be explained in terms of our ignition delay analysis. That is, the extent of the ethanol's preignition reaction has a strong influence on the rate of heat release.

The model predicts that both the convection and radiation heat transfer are reduced during the combustion period due to the fumigation process. These results follow from the lower temperatures predicted by the model due to the lower ratio of specific heats of the air-alcohol mixture.

5. SUMMARY

The conclusion from the review of current methods for utilizing alcohol as a diesel fuel is that to date there is no practical, inexpensive method for retrofitting diesels to burn alcohol. Cetane improver additives and refinery produced emulsions are currently not viable techniques. Methods such as spark or hot surface ignition and pilot injection are expensive and probably limited to new engine designs. Thus, only fumigation and on-board emulsification are serious candidates for current use. Either method, if perfected, could be used to retrofit existing engines and could provide a choice of fuel. However, both methods require two fuel systems and do not allow use of 100% alcohol. Both methods must be used with caution since they could cause engine damage.

The solution and emulsion study was conducted on a TACOM-Labeco open chamber diesel fitted with a piston which gave a 24:1 compression ratio using a pencil nozzle. Emulsions were produced by an Ontario Research Hydroshear unit.

The results of the experiments conducted with neat diesel fuel agree with expected trends from the diesel combustion literature. The performance curves generated show that for this particular research engine, at an engine speed of 1000 RPM and a swirl ratio of 1.4, the minimum BSFC occurs at an injection timing of 22° BTDC. In general, retarding the injection timing from 22 to 15° BTDC, decreased oxides of nitrogen (NO_x) emissions but increased Bosch smoke number and the exhaust temperatures. Retarding the injection timing also decreased peak cylinder pressure and maximum rate of pressure rise. This effect was seen on the apparent rate of heat release where ignition delay was reduced and the amount of early (premixed) burning decreased.

When the fuel/air equivalence ratio was increased from 0.3 to 0.5 with neat diesel fuel, NO_x emissions increased as well as Bosch smoke number and exhaust temperature. Maximum cylinder pressure and rate of pressure rise increased with equivalence ratio. Injection delays were slightly longer for an equivalence ratio of 0.3. The initial rate of heat release was essentially independent of load except at an equivalence ratio of 0.5 and injection timing of 25° BTDC when there was little early (premixed) burning. The magnitude of heat release and the duration of the combustion process increased with increased equivalence ratio.

There were considerable differences between the results with neat diesel fuel and the ethanol-in-diesel solutions and emulsions. At the experimental conditions tested in this research work, the combustion characteristics and emissions of the ethanol solutions and emulsions were quite similar. These similar results were:

1) Longer ignition delays with increasing ethanol concentration
2) Higher rates of pressure rise and increased combustion noise with increasing ethanol concentration
3) An increase in NO_x emissions at 10% ethanol concentration decreasing at 20% and 30%
4) Decreasing Bosch smoke number with increasing ethanol concentration
5) Emissions of unburned hydrocarbons increased substantially with increasing ethanol concentration, nearly doubling at the 20% ethanol concentration over neat diesel fuel
6) CO emissions increased substantially with ethanol emulsions and only moderately with ethanol solutions
7) Indicated thermal efficiency increased with increasing ethanol concentration
8) With the ethanol fuel blends, the apparent rate of heat release is affected by injection timing and equivalence ratio in the same manners as neat diesel fuel
9) The heat release analysis shows that 20% ethanol-in-diesel fuel emulsions and especially solutions have increased ignition delays and higher initial rates of heat release over neat diesel fuel; once the combustion process has become diffusion limited, the shape and duration of the ethanol heat release curves are essentially the same as neat diesel fuel
10) Ethanol has a reduced effect on increasing ignition delay and the entire shape of the rate of heat release diagram at the higher equivalence ratio of 0.5 and retarded injection timing of 15° BTDC
11) Ethanol emulsions caused very rapid deterioration of the pencil injector nozzles used in this research.

From the combustion point of view, the use of ethanol and diesel fuel blends seems feasible when ethanol is added at moderate (10% to 20% by volume) concentrations. The ethanol blends would be beneficial in reducing soot emissions, but concern must be expressed for the possible increase in biological mutagenicity of the soluble fraction of particulates produced by the addition of alcohol to the diesel fuel. Unburned hydrocarbon emissions, which are usually low for a diesel, increase substantially with ethanol-in-diesel fuels. The lower cetane number, longer ignition delay, and higher rate of pressure rise of ethanol blends are problems which could increase starting difficulties, combustion noise and mechanical stress on engine components. The serious problem of injector damage is one that has to be dealt with, but seems to be peculiar to the engine-injection system used in this research.

Introduction of ethanol into the intake air stream is conceptually the most simple way to retrofit an existing engine for alcohol use. Several methods of introducing the ethanol were investigated. Introduction of ethanol as a spray ahead of the compressor appears to give potentially serious problems because of damage to the compressor by impinging liquid alcohol droplets. Injection after the compressor gave cylinder-to-cylinder distribution problems because the engine manifold was not designed for two phase fluids. Vaporization of the ethanol could be accomplished by a combination of a spray or carburetor and a heat exchanger using exhaust heat. For the laboratory study, the ethanol was vaporized using an electrically powered boiler. Such complete vaporization eliminated the problems of compressor wear and cylinder-to-cylinder distribution.

At high loads, ethanol substitution results in a slight improvement in energy efficiency whereas at light loads a small degradation in thermal performance is observed.

The maximum rate of ethanol substitution at high loads was determined by the onset of knocking combustion or excessive peak cylinder pressures. At light loads the maximum possible ethanol substitution rate was determined by the onset of unstable combustion.

Emissions of oxides of nitrogen and smoke decreased with ethanol fumigation whereas unburned hydrocarbons increased as compared to diesel fuel operation.

An abbreviated matrix was run to compare prevaporized lower proof ethanol fumigation with the prevaporized 200 proof ethanol results. It was determined that the cylinder-to-cylinder distribution was excellent (less than 4% difference in peak pressures between cylinders) and that the unburned hydrocarbon emissions, ignition delay, rates of pressure rise and the peak pressures were insensitive to the proof of the ethanol. Of the measured parameters only NO_x emissions changed as the prevaporized ethanol proof changed; the NO_x emissions decreased as the proof of the ethanol decreased.

Ethanol fumigation can either increase or decrease the ignition delay. Reduction in trapped mass temperature and pressure due to vaporization cooling and lowering of the specific heat ratio leads to an increased ignition delay. However, under conditions of higher temperatures (at higher loads and speeds) the alcohol may chemically react to cause a decrease in ignition delay which can partially or totally offset the effects on charge temperature. A correlation of the ignition delay (I.D.) was found in terms of the equation

$$I.D. = 1.308\ P^{-n}\ e^{C/T}\quad [msec]$$

where

P = time average cylinder pressure during delay period, [atm]

T = time average cylinder temperature during delay period, [°K]

$$n = 0.70\ M_d + 0.414\ M_e$$

$$C = 1965.\ M_d + 872.\ M_e - G\ M_e^3$$

$$G = 14.976\ T - 0.0285\ T^2 + 1.423\ (T/100.)^4$$

M_d = mole fraction diesel fuel in total fuel

M_e = mole fraction ethanol in total fuel = $1.0 - M_d$

A heat release analysis was carried out using a three zone thermodynamic model. Pressure and pressure derivative with time are inputs to the model. In this model the diesel fuel combines with ethanol-air mixture to burn at a fixed ratio. When all of the diesel fuel is burned, the remaining ethanol-air mixture burns.

The model shows that fumigation causes a large increase in the initial burning spike even for the cases where the ignition delay is decreased by the fumigation. If the assumption that the diesel fuel burns first is correct, then all of the diesel fuel burns in the initial burning spike.

The model's heat transfer is calculated as radiation from the products zone and convection from the air ethanol zone. The model predicts that both the total and radiation heat transfer are reduced during the combustion period due to the fumigation process.

Finally, a comparison was made between the fumigation data taken at the University of Wisconsin on the turbocharged DI diesel and that of Pennsylvania State University taken on a naturally aspirated IDI diesel engine. It was concluded that differences between the emission levels were caused by the different combustion chamber configurations. However, the engine's sensitivity to the addition of ethanol and how it varies with load and speed is primarily due to the different compression ratios of the two engines and that the effective compression ratio changes with speed and load of the turbocharged engine.

APPENDIX A

Publications and Presentations

"The Use of Alcohol in Farming Applications", G.L. Borman, D.E. Foster, O.A. Uyehara, P.W. McCallum and T.J. Timbario, National Technical Information Service publication DOE/CE/50025-1

"A Survey of Proposed Methods of Burning Alcohol in Diesel Engines", D.E. Foster, proceedings Canadian National Power Alcohol Conference, June 19-20, 1980, Winnepeg, Manitoba, Canada

"Ethanol Fumigation of a Turbocharged Diesel Engine", J. Chen, D. Gussert, X. Gao, C. Gupta and D.E. Foster, Earthmoving Industry Conference, April 6-8, 1981, SAE 810680

"Test Results of an Ethanol Fumigated Turbocharged Direct Injection Diesel Engine", D.E. Foster, proceedings Canadian National Power Alcohol Conference, October 13-15, 1981, Winnipeg, Manitoba, Canada

"Results from Ethanol Fumigation of an I.D.I. and D.I. Diesel Engine", D.E. Foster, P.S. Myers, S.S. Lestz, R.D. Fleming and E.E. Ecklund, proceedings 5th International Alcohol Fuel Technology Symposium, May 13-18, 1982, Auckland, New Zealand

"Combustion of Solutions and Emulsions of Ethanol and Diesel Fuels in a Direct Injection Engine", R. Iwamoto, M.S. Thesis, University of Wisconsin, Madison

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