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thesis entitled

ETHANOL FUEL FOR DIESEL TRACTORS

presented by

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has been accepted towards fulfillment
of the requirements for

Ph.D. degree in Ag Engr Tech

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Date May 21, 1981

O-7639

ETHANOL FUEL FOR DIESEL TRACTORS

By

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A DISSERTATION

Submitted to
Michigan State University
in partial fulfillment of the requirements
for the degree of

DOCTOR OF PHILOSOPHY

Department of Agricultural Engineering

1981

ABSTRACT

ETHANOL FUEL FOR DIESEL TRACTORS

By

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Ethanol appears to be a feasible motor fuel for farm engines because it can be derived from agricultural surpluses and residues as raw material in the ethanol production process. The use of this fuel in turbocharged diesel tractors is considered in this dissertation. This investigation was performed to evaluate the conversion of a diesel tractor for dual-fueling with ethanol by attaching a carburetor to the inlet air system or with the use of an alcohol spray-injection kit. The spray-injection approach consisted of a fuel tank, an air supply line, an orifice, a fuel filter, an instrument panel, and the necessary valves and fittings. In this system the mixture of water and alcohol is injected into the air stream by means of pressure from the turbocharger. The carburetor was attached to a by-pass apparatus which allowed the engine to start and shut off on diesel alone.

Different water-ethanol mixtures were used in the tests. For the spray-injection approach, the mixtures and also distilled water were used in combination with various nozzle sizes.

One energy unit of diesel fuel could be replaced with about one energy unit of spray injected ethanol, and one energy unit of carbureted ethanol replaced a little more than one energy unit of diesel fuel.

Approximately 46 percent of the energy for the turbocharged 65 kW diesel tractor could be supplied by carbureted ethanol, and about 30 percent by the spray-injection approach. Knock limited the extent of substitution of ethanol for diesel fuel. The degree of knock was associated with the amount of water in the ethanol. Spray-injected water had little effect on the engine performance.

The dual-fueling with ethanol caused a slight increase in brake thermal efficiency. Exhaust temperatures were much lower for equivalent high torque levels. Maximum power was increased by 36 percent with the spray-injection approach and about 59 percent with carburetion.

Approved C. Alan Rotz
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to

those who helped me reach the point where I am
at today through their continual support and understanding;
and especially to Professor William Chancellor from
the University of California at Davis;
my high school teacher Vicente Aladin;
and lastly senhor Geraldo Zebral a fine gentleman
from Lafaiete, Brasil.

ACKNOWLEDGEMENTS

A sincere thanks to the members of the thesis supervisory committee. I would especially like to thank Dr. Rotz for his promptness and guidance on the project, and for his overall support of my doctoral program in serving as my major professor.

The author wishes to express a very special thanks to Duane Watson for being extremely helpful with the building and testing of the apparatus.

The author wishes to acknowledge Ford Tractor Operations, Troy, Michigan for providing the tractor used in the work, and M & W Gear Company, Gibson City, Illinois for providing the spray-injection system. Financial support for the project was provided through a grant from the Michigan Department of Agriculture.

Gratitude is also extended to the following members of the Agricultural Engineering Department: Brad Borgman, Dick Ledebuhr, and Dick Wolthuis for their assistance with the construction and testing of the experimental apparatus. Special thanks to Maryanne Favreau for the excellent job in typing the thesis. Also, I wish to thank all the office staff of the Agricultural Engineering Department for their general helpfulness. The author would also like to express his feelings about being a member of the Agricultural Engineering Department at Michigan State University, where teachers and students live as friends and professionals.

The author wishes to thank Professor Comastri from Federal University of Vicosa, Brazil who provided him with the opportunity to continue his graduate studies in the United States. The scholarship granted to the author by the Universidade Federal de Vicosa, Brazil is very much appreciated.

Finally, I wish to express my appreciation to my friends Joan Peterson, Carlos Fontana, and Pauline Simons for their friendship and support during my time at Michigan State University.

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AF	air fuel ratio
BHP	brake horsepower
BHP_c	brake horsepower corrected
BTE	brake thermal efficiency
BT	brake torque
C	orifice meter constant (0.602)
CI	compression ignition engine
d	diameter of venturi
df	diameter fuel orifice
f	correction factor
g	gravitational acceleration
Hc	hydrocarbons
HHV	higher heating value.
h	pressure difference
ma	actual mass of air inducted for intake stroke.
mt	theoretical mass of air to fill the piston displacement volume under standard atmospheric conditions
n	mole
P1	barometric pressure
P2	standard barometric pressure.
p	weight (Newtons)
pto	power take off

Q	air flow.
R	arm length (dynomometer).
r	orifice meter radius
sg	specific gravity.
T	torque
t_1	intake air temperature during the test (absolute)
t_2	standard temperature (absolute)
V	volume.
γ	specific weight
$\dot{\eta}$	volumetric efficiency
ρ	density
Ψ	sizing jet coefficient

CHAPTER 1

INTRODUCTION

In light of the growing dependence of the world economy on fossil fuels, the demand for petroleum has been steadily increasing. This situation has been compounded with the recent political developments in the Middle East, moving energy to the front ranks of concern of the international community.

After World War II, the economies of all nations have become more and more interdependent. This economic interdependence was both the cause and the effect of the rapid recovery from the devastation of war and of a long period of desirable economic growth. But, on the other hand it created insecurity, since the nations did not control the destiny of their economies. Petroleum played a very important role, both in promoting economic growth, and in creating an economic insecurity (27).

Many studies have dealt with the energy problem from the economic view point. Emphasis has been placed on the impact caused by the recent increases in price of petroleum on the behavior of different national economies, on the creation of incentives for developing alternative forms of energy, and on the formation of international agreements.

Alcohol has been a strong candidate as an alternative source of motor fuel because it can be made from renewable resources and, thus,

be a net addition to the energy supply without overdrawing fossil fuel reserves. In actuality, the energy ratios for alcohol (energy in/ energy out) is not very satisfactory. However, this technology has been changing quite fast, and it is expected that the net gain of energy will increase in the future. The production of alcohol from an appropriate energy source can lead to the creation of a highly useful liquid fuel from a less useful form of energy (28).

Because of the convenience, natural gas is often used in the production of alcohol at the processing plant. Nevertheless, natural gas is also the prime source for nitrogen fertilizers and low polluting fuel for home and industrial heating, and should not be diverted to the production of alcohol. Other sources such as coal, biomass and solar energy, which are abundant or renewable, may be used. The decision of which source to use will depend on the cost of the energy source. Thus, there are alternatives which are independent of external markets and can readily take the place of natural gas at the alcohol processing plant (24).

Perhaps we should not analyze alcohol production in terms of energy in and energy out, because a renewable source of energy could be used, such as forest residues and other agricultural residues, and those are not suitable for fueling internal combustion engines. The alcohol net gain controversy varies from place to place and from country to country. In Brazil, alcohol production has proved to be an efficient way to replace fossil fuels (1).

Ethanol appears to be a feasible motor fuel for farm engines because it can be derived from agricultural surpluses and residues (in particular, corn in the United States) as raw material in the ethanol

production process. Furthermore, farmers can possibly incorporate the production of their own fuel as a by-product of the food production system. Even though ethanol production is practical through the use of sugar cane rather than corn in Brazil, it does not appear commercially viable in the United States at present (28).

Ethanol in the U.S. is mainly made from corn, which has created a controversy concerning the use of food for fuel. However, the part used to produce alcohol is starch, and by-products from the fermentation process can be used as a high protein feed supplement, which adds to the benefits of farm produced alcohol. The by-products can also be used in human consumption; however, it would require a change in eating habits. Such a change can probably be achieved with effective marketing and advertising.

Scientists are currently working on the development of commercial processes that will convert cellulose and plant fiber into alcohol, which, when successful, will open up a vast new source of feedstock for alcohol production. Cellulose feedstocks can be obtained from sources such as corn stalks, tree farms, saw mill wastes, crop residues, and even garbage (24).

The use of alcohols as fuel should benefit agriculture directly and indirectly. A very large proportion of the labor involved in the alcohol production would constitute employment opportunities in countries having labor surpluses in rural areas. In addition, it is possible that development of special alcohol crops may afford beneficial substitutes for crops that appear to be in economic over-production in some areas. For instance, the U.S. government pays farmers to not

produce in order to control prices. Instead of eliminating over-production, it should be possible to utilize surplus crops for alcohol production.

The impact of the use of biomass fuel has created an international controversy concerning food production. There is a food shortage in various parts of the world because of uneven distribution (27). With adequate planning and management, including the replacement of traditional crops to a substantial extent, a decrease in food production can be avoided (27). Furthermore, experts from the club of Rome (1) acknowledge that there is enough biomass for energy production to avoid an impact on food production. Perhaps a world policy should be formulated to provide even distribution of agricultural products for both food and fuel.

Petroleum will probably have to be replaced by other energy sources for all countries during the next fifty years. The substitution of petroleum for other sources of energy, in most cases, requires highly complex and large technical installations, and the magnitude of this task can only be recognized and assessed within a global context. Agriculture will certainly have to be viewed in this same context (1).

Ethanol might become the farmers' fuel. By modifying tractor engines to burn alcohol, farmers can use the fuel produced from their surplus crops to produce more crops (28).

Tractors, as well as other agricultural equipment, are powered by diesel engines. Diesel engines burn fuels by compression heating of the air-fuel mixture to the ignition temperature. Because of this, a wide variety of fuels can be used. Alcohol is pre-mixed with the

combustion air that is ignited by the flame produced when conventional diesel fuel is injected, provided that the pre-mixed alcohol does not auto-ignite in an uncontrollable way. These characteristics of the diesel engine permit the use of a number of biomass based fuel materials, in this case alcohol, which may be substituted for part of the petroleum fuels.

The dual-fueled engine, by virtue of its ability to run either on diesel fuel or on a mixture of diesel and alcohol fuel over a wide range of relative concentrations, in the author's judgement, is an attractive and suitable unit for operation where alcohol fuel supplies are readily available. Furthermore, the engine may be easily changed over from dual-fueling to diesel operation, which is of particular convenience when the alcohol supply may be fluctuating (4).

A kit, which is no longer on the market, developed by M & W Gear Company, allows diesel tractors to be converted to dual-fueling. This system includes a fuel tank, hose and nozzle to deliver the alcohol into the engine. The tank is pressurized with the outlet air from the turbocharger, which forces alcohol from the tank through a hose into the inlet to the turbocharger. This system provides little control of the alcohol fuel because the flow of alcohol into the engine increases with increase on turbocharger pressure. Turbocharger pressure varies as the load in the tractor varies (13).

A carburetor can also be used to introduce alcohol into the intake air. This method appears to be promising due to its low cost and the possibility that, with proper guidance and some knowledge of shop mechanics, the farmer could make the modification himself. The

dual-fueling of agricultural equipment having diesel engines can help farmers become less dependent on fossil fuels (4).

CHAPTER 2

LITERATURE REVIEW

Because of the recent interest in alcohol as an alternative fuel, this literature review covers wide aspects of alcohol and its use in internal combustion engines.

2.1 Comparison of Available Fuels

The combustible elements in fuels are predominantly carbon and hydrogen with small amounts of sulfur as the only other fuel element. Petroleum and alcohol fuels are both mixtures of complex hydrocarbon compounds, which are responsible for different properties of the fuels. Knowledge of these differences is very important to enable good understanding of the performance of fuels. Fuel comparisons are outlined in Table 1.

The definitions of the following characteristics are vital for understanding, evaluating and comparing fuels.

2.1.1 Caloric Value (heating value)

The heating value is one of the most important determinations necessary in fuel comparisons. It is defined as the amount of heat energy contained in the fuel, and is the basis for computations of thermal efficiency.

The most common method to determine the heat value is the bomb calorimeter. The heating value is determined by actually burning the

Table 1. Properties and characteristics of alcohol, diesel and gasoline fuels.

	Gasoline	Diesel	Ethanol	Methanol	Gasohol*	Diesohol**
Source	Petroleum	Petroleum	Agricultural products, petroleum	Forest products, natural gas, coal	--	--
Formula	C_4-C_{12} mix	C_6-C_{14} mix	C_2H_5OH	CH_3OH	--	--
Specific weight (grams/cc)	0.743	0.827	0.802	0.791	0.743	0.827
Energy content (kj/l)	33,727	38,746	23,580	18,006	32,723	37,210
Energy content (kj/kg)	45,820	54,659	29,772	22,678	44,192	52,170
Pump octane	87-98	20-30	98-102	99-102	90-91	--
Intake air/fuel ratio	14.9	15.0	9.0	6.5	14.0	14.6
Heat of vaporization (kj/kg)	409	156	921	1,176	419	--
Boiling point (°C)	-0.5-2	198-338	78	65	-0.5-216	78-338
Vapor pressure (Pa at 38° C)	345	--	120	220	392	--
Cetane	--	50	8	3	--	--
Reactions with materials	--	--	Some plastics, rubber, lacquer, paint, epoxy	Some plastics, rubber, copper, brass, terneplate***	--	--

*Defined as 10% anhydrous ethanol and 90% gasoline.
**Defined at 10% anhydrous ethanol and 90% diesel oil.
***A lead-tin coated steel used in automobile fuel tanks.
Sources: (References 3, 16).

fuel sample and measuring the temperature rise. The temperature rise is converted to heating value for fuel mass by algebraic comparison with the heating value of benzoic acid. Corrections are made for the heat contributed by the ignition wire, and the heat of formation of nitric acid and sulfuric acid, by-products of the combustion process (3).

Any fuel containing hydrogen yields water as one product of combustion. At atmospheric pressure, the partial pressure of the water in the resulting combustion gas mixture will usually be sufficiently high to cause water to condense if the temperature is allowed to fall below 48 to 60° C. This causes liberation of the heat of vaporization of any water condensed. The low heat value is determined assuming that no water vapor condensed in the process, whereas the high heat value is calculated assuming that all water vapor condensed (3).

Ethanol contains about two-thirds as much energy as gasoline and diesel fuels. Methanol has only 48 percent of the caloric value of gasoline. The low energy content of both causes the fuel consumption to normally be higher when alcohol is used. Consequently, a larger volume of alcohol has to be burned to provide the work delivered by petroleum fuels on a normal engine (20).

2.1.2 Octane Rating

The octane number of a fuel is determined by comparing the unknown fuel with various mixtures to find the knock tendency. A mixture of iso-octane and heptane is matched to the unknown fuel in knock tendency following the specified procedure in the American Society for Testing Materials -- Cooperative Fuel Research knock engine testing (16). A fuel

that causes less knock than iso-octane is rated by the amount of tetra-ethyllead in iso-octane required to match the knock of the unknown fuel (16).

The octane rating is a measure of the ability of a fuel to resist knocking during combustion. The knocking ratings of the fuels are in rough proportion to the auto-ignition temperatures.

The octane rating of alcohol is significantly higher than that of gasoline. The high octane rating allows the fuel to be burned in a more efficient high compression ratio engine. The benefits gained through higher compression offset some of the disadvantages of greater fuel consumption because of lower energy content (5).

2.1.3 Heat of Vaporization

The difference in energy content between saturated-vapor and saturated liquid is called latent heat of vaporization. It represents the quantity of energy required to vaporize a unit mass of a saturated liquid at a given temperature or pressure.

Because alcohol has a higher heat of vaporization and a lower vapor pressure than gasoline, starting alcohol-fueled engines in cold weather is difficult. The S.I. engine will not start on alcohol alone in temperatures below 10° C (10). For efficient operation in cold weather, the intake air should be around 90° C (10). The high heat of vaporization causes the alcohol to cool the intake air as it evaporates. This can increase the power output because a higher volumetric efficiency can be obtained with the cool compressed air. There is less work done on the compression stroke because vaporization holds down temperature and pressure which forms more dense air. More work is done

on the expansion stroke since the mole product per mole mixture is higher (16).

2.1.4 Vapor Pressure

For every liquid, the internal molecular activity is such that molecules escape from the surface until the pressure within the space next to the surface reaches a value that allows the net exchange of molecules between the liquid and vapor to come to equilibrium. This pressure is called the vapor pressure. Because molecular activity depends upon temperature, the vapor pressure in turn is a function of the temperature of the liquid. This induces problems with cold start, because the vapor pressure drops with a drop in temperature (16). Thus the fuel tends to resist evaporation as the temperature is decreased. This again causes problems in cold starting of engines on alcohol fuels.

2.1.5 Cetane Number

The cetane number represents to diesel engines what the octane rating represents to spark-ignition engines. However, the relationship between the cetane number of a diesel fuel and the performance of a diesel engine should not be confused with the relationship between the octane number of gasoline and the performance of a spark-ignition engine. With a spark-ignition engine, raising the octane number improves potential engine performance by allowing the compression ratio to be increased. In the diesel engine, the desirable level of cetane number is established by the requirements of good ignition quality at light loads and low temperatures (12). Cetane, also called hexadecane, has

a low self-ignition temperature and, therefore, is a good fuel to prevent knock in a compression ignition (CI) engine.

The reference scale for measuring CI knock is based upon hexadecane and heptamethylnonane as primary reference fuels with assigned values of 100 cetane and 15 cetane respectively (16).

The cetane number tends to increase as octane number decreases. Thus, fuels having very high octane numbers, such as alcohol are inappropriate fuels for direct fueling of diesel engines. In other words, alcohol will not ignite in the conventional diesel engine when compressed, because of its low cetane number. However, it can be used as a supplement to reduce diesel fuel consumption.

2.1.6 Stoichiometric Air/Fuel Ratio

The stoichiometric air/fuel ratio is the chemically correct mass of air used, to convert a given amount of fuel into completely oxidized products.

The speed with which fuel induction takes place in an engine (CI) makes it impossible to obtain perfect mixing of air and fuel. In order to consume all of one, there must be an excess of the other. If maximum power is desired, an excess of fuel must be supplied so that all the oxygen available will be consumed. If economy is desired, an excess of air must be present so that all the fuel will be burned (12).

A lower air/fuel ratio compared to gasoline and diesel fuels is required for alcohol, because of the presence of molecular oxygen in alcohols and the lower number of carbons oxidized in the combustion process (see Appendix C for derivations).

2.2 Alcohol Blends

When used as a fuel in spark-ignition engines, alcohols may be blended with gasoline or used alone. Alcohol blends are in some respects interchangeable with gasoline and therefore, seem to have the advantage of being more adaptable for current spark-ignition engines. Alcohol, when blended with gasoline (90% unleaded gasoline and 10% ethanol), is called "Gasohol." This mixture has been used successfully in automobiles.

When fueling gasoline/alcohol blends made with methanol, many automobiles perform unacceptably by present standards. Modifications are required to replace materials in the fuel system that are not compatible with alcohol blends (10).

The most noticeable effect when an alcohol blend is used in an unmodified engine is a leaner air-fuel mixture. An engine operating on an alcohol blend behaves as if the carburetor is adjusted to give less fuel (5). For both methanol and ethanol, the engine can operate satisfactorily with these lean mixtures according to Imgamells and Lindquist research, although the carburetor should be adjusted to compensate for added alcohol, by enlarging the jets (10).

Expected effects when unmodified engines are operated on a leaner mixture than the theoretical air-fuel ratio include the following:

1. Lower fuel economy is expected, because of the low heating value of alcohols. However, on the energy basis (km per GJ) the addition of alcohol to gasoline causes improvements in fuel economy, because thermal efficiency is increased.

2. More efficient burning because of its combustion characteristics, particularly when operating under lean conditions (10).
3. Deterioration in driveability, because of the sensitivity to phase separation with low amounts of water (10).
4. Higher power output--the heat of combustion of equal volumes of stoichiometric air/alcohol and air/gasoline mixtures are nearly identical. More power is obtained because of alcohol's higher latent heat of vaporization which cools the air entering the engine much more than gasoline, and this increases the air density and the mass flow (10). A 10% gain in power output with methanol is possible when very high mixtures are used (10).
5. Lower carbon monoxide and hydrocarbon emissions--As there is more air available to burn the fuel, combustion is more complete and emissions are lower. If the fuel mixture is very lean, however, hydrocarbon emissions will increase because of poor combustion characteristics (10).
6. Generally, higher emissions of nitrogen oxides--Emissions of nitrogen oxides peak on the lean side of the theoretical fuel air ratio. Tuning engines increasingly leaner from this peak, however, will again reduce oxide emissions (10).

7. Road octane increases, and better engine performance (10).
8. Cold starts generally do not present a problem. (10).

2.3 Straight Alcohol in S.I. Engines

The use of straight alcohol for spark-ignition engines introduces certain new problems. Intake systems of engines have to be redesigned to aid in vaporization of the fuel because of the much higher latent heat of vaporization. Some fuel system metals, plastics and elastometers have to be changed to avoid corrosion and incompatibility problems (10).

Pischinger (19) points out that a relatively high boiling point of alcohol along with heat of vaporization, which is 3.5 times higher than gasoline, make it very difficult to attain a lean mixture in the engine. The mixture formation along the way to the combustion chamber is hampered by these characteristics even in warm alcohol engines.
In the same way, heat is affected in gasoline engines with cold running conditions (leaner air-fuel ratio). This means that the alcohol engine never warms up properly in the carburetion area, so that the operation of a car so equipped always requires a rich mixture. As a result, both the fuel consumption and emissions of carbon monoxide become unacceptable. To get good performance, the intake air must be pre-heated along with the walls of the inlet manifold. The engine heat radiation contributes to this heating.

Difficulty in cold starting can be overcome by using a device to spray a second fuel in the intake air. It has also been proposed that

heaters be used to evaporate a small quantity of the alcohol instead of the use of a second fuel (19).

Other factors that have to be taken into account are the compression ratio and ignition timing. The compression ratio should be raised to take advantage of the high octane fuel. Ignition advance is recommended because alcohols have a lower flame temperature and faster burning velocity (19).

Power, fuel economy, emissions and maximum torque are determined by the amount of air that can be inducted into each cylinder and the energy that can be liberated by the combustion process utilizing the oxygen in the air. The cooling effect caused by a higher latent heat value decreases compression work or allows induction of a greater mass of air into each cylinder (4).

Carbon monoxide and hydrocarbon emissions are nearly the same for methanol and ethanol compared to gasoline alone, but a marked difference appeared in the emission of nitrogen oxides (7). For methanol, emissions of nitrogen oxides are slightly lower, and they peaked at a richer mixture of air/fuel.

Alcohol fuels have been used successfully in automobiles. In extremely severe road tests in Brasil covering more than 100,000 km, no damage whatsoever was observed in the engines that could be blamed on the use of alcohol (19). In January of 1978 the state of California demonstrated that a conventional automobile could be operated on 100 percent methanol. The fuel economy was excellent, ranging from 6 to 7 kilometer per liter (km/l), which corresponds to about 12 to 13 km/l on an energy equivalent to gasoline (because of the lower caloric value

of alcohol). The fuel economy of the gasoline version was lower, being about 8.9 km/l (15).

A Brasilian magazine "Quatro Rodas" (20) shows a comparison of two Brasilian Ford passenger cars. The cars are the same except that one was designed for alcohol and the other for gasoline. The average fuel consumption was 13.7 km/l for the gasoline version and 11.5 km/l for the alcohol (80 percent by volume) car. At constant speed (40 km/hr) and fifth gear, the alcohol car showed better fuel economy (17.0 km/l) than the gasoline version (16.8 km/l). The surprise was the brake thermal efficiency (BTE). The gasoline version had a 27 percent BTE, while the alcohol version had a surprising 37 percent BTE, which is even higher than the diesel cycle (around 35 percent).

2.4 Alcohol Use in Diesel Engines

Since the majority of tractors are powered by diesel engines a comprehensive review of the different methods of using alcohols in diesel engines is reported. The methods include the conversion of an engine to burn straight alcohol, and the conversion for dual-fueling or the use of alcohol to supplement diesel fuel.

2.4.1 Total Alcohol Fueling

Here, two approaches are possible for fueling a diesel engine with straight alcohol solutions. One approach is to place additives into the alcohol to improve its cetane rating; the other is to modify the engine to straight alcohol.

Because alcohols have a high octane ratio, they may be used in spark-ignition engines, which need a fuel that when mixed with air,

can be compressed in the cylinders without self-igniting. On the other hand, diesel engines need a fuel with exactly the opposite characteristics; in other words, a fuel that will self-ignite when injected into the cylinders. From the above explanation the question can be raised: How can alcohol (high octane ratio and very low cetane number) be used in diesel engines? Holmer (8) solved this problem by adding "cetane improvers" with high self-ignition capacity.

Engines are capable of burning alcohol through this method with few modifications. The main change would be the injector pump which must be modified to be self-lubricated, since alcohols do not have lubrication characteristics. Furthermore, the pump also must compensate for the fuel injected into the cylinders, because the fuels have different heat values, and burn at different air/fuel ratios.

Even though this method displaces 100 percent of normal diesel consumption it has been proved unfeasible due to technical problems and high price of cetane improvers (8). For these reasons, this method has low potential in saving diesel fuel unless a low price cetane improver is developed.

Through major mechanical modifications, an engine can be converted to burn straight alcohol. Modifications would include the addition of a spark-ignition system, bringing the compression ratio to around twelve-to-one, and a change in the fuel injection system (6).

A gasoline engine can be adapted and optimized to burn alcohol. This engine must have its compression ratio increased, and consequently its parts may be subjected to overstress because the original engine was not designed for high cylinder pressures. The ideal engine for

conversion to alcohol fueling is the diesel engine because it functions at a high compression ratio, and therefore it is reinforced to withstand stress under such a condition.

The alcohol consumption in a modified diesel engine is lower than that of a spark-ignition engine not optimized for alcohol. This difference in fuel consumption would increase even more when the engine is working with frequent acceleration and deceleration such as in urban traffic conditions (14).

The method of burning straight alcohol in a modified engine is called the Brandt system (26). The compression ratio is lowered to around twelve-to-one. The alcohol is injected at high pressure directly into the cylinders using a self-lubricated fuel pump and specially lubricated injectors. Alcohol is thus mechanically atomized into a fine mist which evaporates instantly in the cylinder as it absorbs the heat developed in the compression stroke of the engine. An ordinary spark plug is used to ignite the air/fuel mixture.

The required modifications include a decrease in compression ratio, addition of a spark-ignition system and replacement of the fuel injection system with an injection system compatible with alcohol. The engine can have its compression ratio altered by three approaches: 1) by changing the engine head, with a different clearance volume to achieve the desired compression ratios; 2) by replacing pistons with different shaped pistons to obtain the desired compression ratio; and 3) by replacing the connecting rods with shorter rods.

The spark plug can be installed by drilling a hole through the head at each cylinder and fitting a sleeve through the water jacket.

A distributor or electronic ignition system can be used to provide ignition timing.

Higher heat is required to vaporize the alcohol when greater amounts of water are present. Therefore, a heat exchanger is necessary to provide adequate vaporization, without interfering with air flow. Either the hot radiator water or the exhaust gases are available as a free heat source. In the case of turbocharged engines, the air is heated by compression, and the heat transferred from the exhaust through the turbocharger case, however, this heat is not enough to provide all the heat needed (9). Control of vaporization is important. Any heat in excess of that required for complete vaporization reduces air density and thus lowers the engine performance (9).

Although the Brandt system is quite sophisticated, it is not impossible for the farmer with a good knowledge of mechanics to make the modifications himself with proper directions. However, it should be done by a reliable mechanic, or shop. Even though recent work has not been done, this technique provides potential for replacing 100 percent of diesel fuel requirements. However, the fuel consumption is expected to be greater than diesel consumption, in spite of the fact that efficiency and power are expected to increase. The operational costs will depend on the production cost of alcohol (26).

2.4.2 Blending Diesel and Alcohol Using an Emulsifier Process

The preparation and injection of a homogeneous mixture of alcohol and diesel oil is technically feasible but requires a comparatively high engineering outlay. Its disadvantage is demulsification of the mixture when the engine is not running or during cold operation (11).

Goering (29) indicated that a diesel engine performed satisfactorily on a 10 percent anhydrous ethanol blend with diesel fuel, provided that the fuel system and injector pump were either chilled or pressurized to prevent vaporization of the alcohol and related vapor-lock problems.

Solly (23) developed a new fuel called "cocohol." This fuel was produced in a simple process by the chemical combination of ethanol and coconut oil. Cocohol has similar chemical and physical properties to cetane, the chemical pure standard for diesel fuel. The viscosity and solidification temperatures are very similar. Furthermore, the fuel was shown to operate a high speed diesel engine at a greater thermal efficiency than diesel oil. This fuel is still in the phase of research.

Pischinger et al. (18) tested a Volkswagen's Passat (Dasher in USA) which was available with gasoline, ethanol (in Brazil) or diesel engines. The test included alternative fuels such as: peanut oil and soybean oil, hydrated ethanol with "cetane number improver," and hydrated ethanol. They concluded that vegetable oils considered as alternatives to diesel oil have properties as adequate for diesel engines as those of the alcohols for S.I. engines. Partial substitution by blending was very attractive because of good miscibility, comparable energy densities, and the absence of any need to modify either engine or fuel systems with the consequent possibility of immediate introduction.

They also concluded that alcohol/diesel oil blends with reasonable proportions are possible if the poor miscibility is overcome by chemical or mechanical means. For 30 percent blends, the following rates of substitution can be expected:

1. 30 percent by volume vegetable oil substituting about 30 percent diesel oil.
2. 30 percent by volume ethanol substituting about 20 percent diesel oil.
3. 30 percent by volume methanol substituting about 16 percent diesel oil.

When compared the car powered by the swirl-chamber ethanol diesel engine with the one powered by the ethanol S.I. engine at comparable vehicle performances, they found the consumption of the S.I. car to be very close to that of the diesel.

2.4.3 Dual-fueling

Since the total conversion of the diesel engine to alcohol is expensive, dual-fueling seems more attractive. Different ways of dual-fueling include: 1) the mixture of alcohol and diesel fuel at the intake gallery of the injection pump; 2) the use of two injection pumps (one for alcohol and one for diesel fuel) to supply fuel to a single injector; 3) the dual-fueling of the diesel engine with carbureted alcohol; 4) the use of two injectors in each cylinder with a separate fuel system for each; and 5) spray-injection using turbocharger pressure.

In the first process the alcohol is sprayed by means of an injection pump, at the intake gallery of the diesel injection pump. This design calls for a complete and separate diesel fuel system with tank, feed, pump, injection pump and separate injectors (8). This system can replace up to 50 percent of the diesel, but it requires high technology. The fuel consumption in terms of volume, is expected to be greater than

the operation with diesel alone. The reduction of the costs of operation will depend on the production cost of the alcohol.

Basically, the use of two injection pumps (one for alcohol and one for diesel fuel) to supply fuel to a single injector work in the same way as in the first method, but instead of spraying the mixture of alcohol and diesel into the gallery, it goes to a third injection pump, which sprays the mixture into the cylinder (8). As in the first method, this system can save up to 50 percent diesel fuel, and the engine efficiency is improved.

In the case ofdual-fueling a diesel engine with carbureted alcohol, a carburetor is installed on the combustion air intake system of a diesel engine. With the manifold carburetion system, the alcohol is mixed with the manifold air. The ignition is then induced by the conventionally injected diesel fuel (4). This method has good potential for a partial substitution of diesel fuel, since it is less expensive to construct than the others, and the farmer can do it himself with proper information. This system can displace approximately 50 percent of the energy for diesel engines (5). Although the production cost of alcohol is prohibitive at present in the United States, the costs of operation can be reduced, since the efficiency and power increase.

Smith (22), mentioned that a sonic nozzle which atomizes the alcohol in a modified intake manifold was able to burn more than 50 percent alcohol in a diesel engine without the fuel separation, knocking, and lubrication problems normally associated with diesel/alcohol blends. The alcohol droplets enter the cylinders with the air stream, totally separate from the diesel fuel system. He also reported that compared to using straight diesel fuel, the test engine developed higher power.

For dual-fueling with dual injectors, two injectors are installed in each cylinder with a separate fuel system for each. The engine is fitted with a distributor type injection pump, and a simple hole injector in each cylinder head is installed. With this system, as much as 80 percent of the fuel energy can be supplied from methanol with proper timing of both injection systems (8).

The replacement of diesel fuel here is the highest among the dual-fueling methods, but it requires sophisticated engineering. Furthermore, the investment to install this system is very high. It is doubtful that the operational costs for this process can be lowered, even with low production cost of alcohol, because of the high investment cost in the system.

The M & W Gear Company kit, referred to in the Introduction, has been used to convert a turbocharged diesel tractor to dual-fueling. In this system, a mixture of water and alcohol (aquahol) is injected into the air stream by means of pressure from the turbocharger. A tank is pressurized with the outlet air from the turbocharger, which forces alcohol from the tank through a hose into the inlet. This system was promoted to increase power and efficiency, and replace about 30 percent of normal diesel consumption (13).

Based upon the literature review for this thesis, the author has prepared a table to summarize the different methods of using alcohol in diesel engines (Table 2).

Table 2. Summary of different methods of fueling diesel engines with alcohol.

Type of conversion	Initial investment	Diesel displaced percent	Efficiency and power	Handling by user	Capacity to reduce oper. costs if alcohol price is competitive	Possibility of conversion by skilled user
Addition of cetane improver in the alcohol	None, but cetane improvers, are very expensive	100	Same as diesel	Very easy	Doubtful	Very good
Engine total conversion	High	100	Equal or higher than diesel	Very easy	Very capable	Good
Blending diesel with alcohol	Very high	20	Equal to diesel	Confusing	Doubtful	Poor
Mixing diesel/alcohol within intake gallery of the injection pump.	Very high	50	Higher or equal to diesel	Very easy	Capable	Poor
Using two injection pumps	Very high	50	Equal or higher than diesel	Very easy	Capable	Poor

Table 2 (cont'd)

Type of conversion	Initial investment	Diesel displaced percent	Efficiency and power	Handling by user	Capacity to reduce oper. costs if alcohol price is competitive	Possibility of conversion by skilled user
Dual-fueling with carbureted alcohol	Moderate	50	Equal or higher than diesel	Can be easy	Very capable	Very good
Use of two injectors in each cylinder	Very high	80	Equal or higher than diesel	Very easy	Very capable at factory	Poor
M & W Gear kit	Moderate	30	Higher (?)	Easy	Very capable	Very good

2.5 Theory of Knock

To understand knock we first have to understand the phenomenon of combustion in internal combustion engines.

In S.I. engines the combustion normally begins at the spark plug where the molecules in and around the spark discharge are activated to a level where reaction is self-sustaining. Once the reaction is underway, a spherical flame front will advance from the spark plug. In the vicinity of the chamber walls both turbulence and temperature are low and therefore the flame speed is retarded. In this stage the unburned gas ahead of the flame front, and the burned gas behind the flame front are compressed by expansion of the burning mixture and are raised in temperature. Therefore, the pressure throughout the chamber is continually increasing. The final stage of combustion is when the flame slows down as it approaches the walls of the combustion chamber and is finally extinguished (16).

To understand knock we must first define it. However, a satisfactory definition for knock is difficult to give because of the complexity of the combustion process. In general, knock is the term used to signify any unusual sound that arises because of autoignition in the combustion process (16). If knock implies autoignition, an infinite range of severity can be present, and high speed photography of the combustion process would be necessary for identification of the knock. If knock implies pressure differences in the combustion process, then the sensitivity of the pressure-measuring equipment would be a factor in the definition. If knock implies sound, which was considered in this dissertation, then the sensitivity of the ear enters the problem.

The cause of knock or detonation in spark-ignition engines and compression ignition engines is basically the same. In both situations, compression ignition is followed by a rapid pressure rise. However, "in the spark-ignition engine it is the last part of the charge to burn, while in the compression ignition it is the first part up to, and sometimes including, the whole charge"(25).

The tendency of given fuel to detonate is measured by its octane number. A higher octane rating means the antiknock (antidetonation) qualities of the fuel are better. In the spark-ignition engine the combustion process rarely occurs without some trace of autoignition (detonation). When autoignition occurs the pressure and temperature may abruptly increase because of the sudden release of chemical energy (16). The consequent rise in pressure compresses the end gas ahead of the flame front and therefore its temperature and density increase causing autoignition at the periphery of the combustion chamber. The combustion time is always shortened by autoignition with consequent sharper rise in pressure that may lead to an audible sound called knock. Two distinct types of autoignition can be developed in the combustion process, and each with various degrees of severity (16): explosive (usual) and nonexplosive autoignition. Explosive means that the rate of chemical reaction is greater than the rate of expansion. Nonexplosive means that the rate of expansion is greater than the rate of chemical reaction so that the pressure pulse in the autoigniting region is too small to cause the audible sound called knock. In S.I. engines it is relatively easy to distinguish between knocking and non-knocking operation if only because the sensitivity of the ear can afford an acceptable distinction. It is the end portions of the mixture that may self-ignite, and, if knock appears, it will appear near the end of the combustion process (16).

In the C.I. engine, air alone is compressed and raised to a high temperature on the compression stroke. One or more jets of fuel are then introduced into the combustion chamber. The jet disintegrates into a core of fuel surrounded by a spray envelope of air and fuel particles (16). This envelope is created both by the atomization and vaporization of the fuel and the turbulence of the air in the combustion chamber as it passes across the jet and strips the fuel particles from the core. At some location in the spray envelope a mixture of air and fuel will form and oxidation becomes imminent (16). This period of physical delay is the time between the beginning of injection and the attainment of chemical reaction conditions. In the next stage, called the chemical delay, reaction starts slowly and then accelerates until inflammation or ignition takes place. At some location, or at many locations, flame appears. Rather than an orderly propagation of flame along a definite flame front, entire areas may explode or burn because of the accumulation of fuel in the chamber during the delay period (16).

In the C.I. engine, the fuel is injected into hot air and combustion begins with autoignition. Thus, if pressure disturbances are apparent, it is at the beginning of the combustion period that knocking occurs (16). Self-ignition is the essential condition for establishing either a pressure difference or a rapid pressure rise in the chamber of the C.I. engine.

The so-called knock rating of a diesel fuel is found by comparing the fuel with hexadecane, which is more often called cetane. Cetane number describes the ignition characteristics of the fuel rather than knock. Fuels with higher cetane rating have lower ignition delay. Since ignition delay is the primary factor controlling the initial auto-ignition in the compression ignition engine, it is reasonable to conclude that knock is related to the ignition delay of the fuel (16).

If the delay time is long, there will be more opportunity for the fuel and air to mix intimately before combustion starts. The air-fuel mixture in the compression ignition engine is not homogeneous. Regions exist with droplets of fuel alone, with fuel vapor alone, with air alone, and with fuel-air mixtures. When ignition begins in a region that contains fuel and air, flame will propagate if the regions of mixtures is continuous. Adjacent regions on the verge of self-ignition may ignite from heat transferred from the burning region. In any event, it would be difficult to distinguish between flame propagation and self-ignition which is aided, of course, by the high temperatures being generated in the chamber (16).

"Ricardo conceived the combustion process in the compression-ignition engine as taking place in three stages, the first of which is the delay period. The delay is always long enough that, when ignition occurs, there is an appreciable amount of evaporated and finely divided fuel well mixed with air. Once ignited, this fuel tends to burn very rapidly by reason of the multiplicity of ignition points and the high temperature already existing in the combustion chamber. This period of rapid combustion is Ricardo's second phase of the process. After the period of rapid combustion, the fuel which has not yet burned, together with any fuel subsequently injected, burns at a rate controlled principally by its ability to find the oxygen necessary for combustion. This period is Ricardo's third stage of combustion" (25).

If the ignition delay of a fuel is too large, too much fuel is injected into the combustion chamber prior to ignition. Simultaneous ignition of this large amount causes an excessive pressure rise and it will produce a sound due to the impact of the gases in the combustion chamber.

The high pressure differences created by ignition at different points in the combustion chamber cause the gas to vibrate which in turn cause vibration of the walls of the chamber producing an audible sound called knock (16).

CHAPTER 3

OBJECTIVES

The overall objective was to demonstrate the feasibility of using alcohol as alternative fuel for diesel farm tractors, and determine any associated problems. More specific objectives were:

1. To evaluate the performance through laboratory testing of a diesel tractor, dual-fueled with the use of an alcohol spray-injection kit.
2. To evaluate the conversion of diesel engines for alcohol use by attaching a carburetor to the inlet air systems of turbocharged tractors.
3. To determine the relationship of water content in the alcohol to the increase in power and fuel consumption.
4. To develop a method for injecting the right amount of alcohol into the tractor under varying load conditions.

CHAPTER 4

EXPERIMENTAL PROCEDURES

Dual-fuel tests were conducted with a four cylinder turbocharged diesel engine "Ford 7700"[1] tractor (see Specifications in Appendix A). A schematic arrangement of the apparatuses are given in Figures 1 and 2.

4.1 Testing Procedure

In the first part of the tests, a commercial kit produced by M & W Gear Company[1] was mounted on the tractor. In this system the mixture of water and alcohol was injected into the air stream by means of pressure from the turbocharger. Different water-alcohol mixtures and distilled water were used in combination with various nozzle sizes to determine optimum nozzle size and fuel combination for this system. The graduations and fuel mixtures were as follows:

1. 50 percent ethanol-water mixture, using graduated nozzle sizes of: .51 mm, .76 mm, .89 mm, 1.02 mm and 1.09 mm.
2. 84 percent ethanol-water mixture with nozzle sizes graduated at .13 mm intervals as follows: .51 mm, .64 mm, .76 mm, .89 mm and 1.02 mm.
3. 100 percent ethanol for nozzle sizes .51 mm, .64 mm and .76 mm.
4. Distilled water with nozzle sizes of .51 mm and .64 mm.

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[1]Trade names are used in the thesis solely to provide specific information. Mention of a trade name does not constitute a warranty of the product by Michigan State University or an endorsement of the product to the exclusion of other products not mentioned.

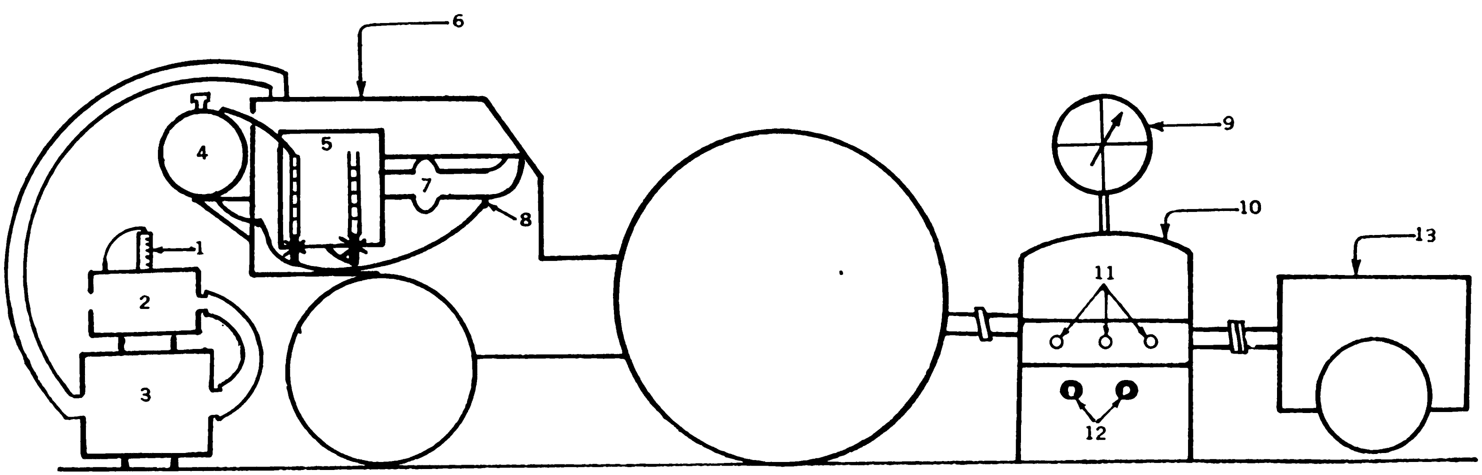
The maximum useable nozzle size in each case was limited by engine knocking.

All the tests for the spray-injection method of dual-fueling were run at 2100 engine rpm or 1000 rpm pto speed (the higher pto speed was chosen to accommodate the high speed dynamometer used). For this experiment the tractor was connected to two dynamometers through its power-take-off to load the engine (see Figure 1) because the main dynamometer would not measure more than 37 kW at 1000 rpm. The procedure used was to load the engine for 5.5 N, read at the main dynamometer scale, and subsequently increase the load by increments of 5.5 N, until the recording range of the main dynamometer was approached. The resulting total load was then transferred to an auxilliary dynamometer and the testing continued by incrementing the load by 5.5 N. The last reading was then obtained by setting the speed control lever of the tractor to full load position, followed by loading the main dynamometer until the tractor tachometer indicated 2100 engine rpm.

The second method of dual-fueling was done by attaching a carburetor to the inlet air system of the turbocharger. The tractor was connected to a dynamometer through its power-take-off (see Figure 2, 2a). In order to determine the optimum substitution of ethanol for diesel fuel via carburetion, the following feasible combinations were tried:

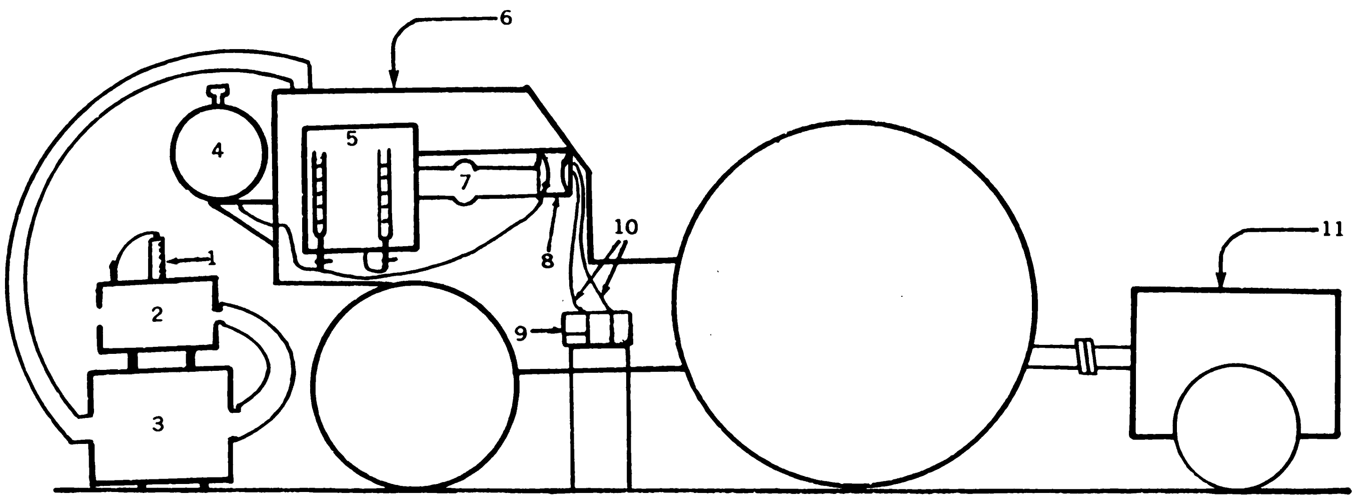
1. Varying air/diesel fuel ratios.
2. Varying air/ethanol ratios.
3. Different ethanol concentrations--100, 80 and 50 percent by volume.

The procedure used was as follows: The engine speed was set at 2100, 1600 and 1150 engine rpm, which corresponded to full, 3/4, and 1/2



1 Manometer
2 Surge tank orifice meter (95 liter tank)
3 Surge tank (208 liter tank)
4 Ethanol tank
5 Calibrated pippettes
6 Tractor
7 Turbocharger
8 Nozzle
9 Main dynamometer read out
10 G E cradle type dynamometer
11 Gages
12 Controls
13 Auxiliar dynamometer hydraulic type

Figure 1. Schematic layout of the test apparatus used to measure performance of the turbocharger diesel tractor with the spray-injection method of dual-fueling.



1	Manometer	7	Turbocharger
2	Surge tank orifice meter (95 liter tank)	8	Carburetor
3	Surge tank (208 liter tank)	9	Temperature read out
4	Ethanol tank	10	Thermocouples
5	Calibrated pipettes	11	Dynamometer (hydraulic type)
6	Tractor		

Figure 2. Schematic layout of the test apparatus used to measure performance of the turbocharger diesel tractor with the carbureted method of dual-fueling.

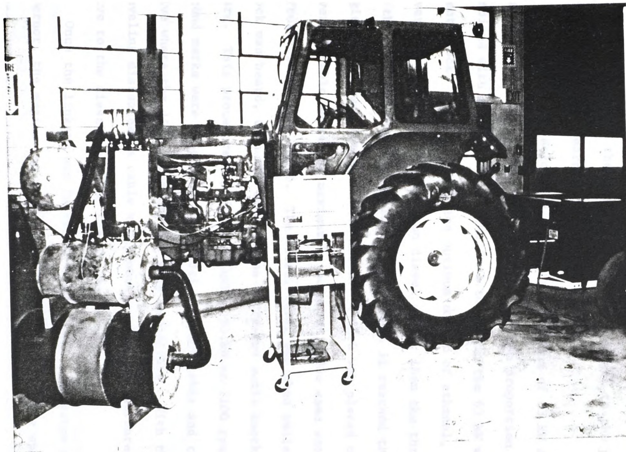


Figure 2a. Arrangements of the apparatus.

engine load respectively, using diesel fuel alone. At each of the above engine settings the ethanol/air ratio was then increased by means of the throttle valve in the carburetor until incipient knock was heard. From there, the alcohol was throttled back to the point where no more knock was heard.

Because the tractor was rated for a maximum of 63 kW at 2100 rpm engine speed, the aim was to supply the maximum proportion of ethanol combined with the minimum diesel fuel to obtain the 63 kW without incipient knock. To accomplish the optimum amount of ethanol, the diesel governor was set to full load on diesel alone, then the throttle valve on the carburetor was gradually opened, until it reached the maximum engine horsepower without undue knock. A mark was placed on the diesel governor cable where the maximum hp occurred. The same was done on the carburetor throttle valve. The diesel lever was then pulled back until knock was heard, then the ethanol was decreased until knock was no longer heard. This procedure was followed until 63 kW at 2100 rpm was obtained. Second marks were placed on the diesel governor cable and throttle valve where the diesel and ethanol were compatible. With those cable traveling distances, a cable was connected from the carburetor throttle valve to the diesel pedal on the tractor's cabin.

Once the linkage was installed between the carburetor and diesel governor, the engine was tested again at 2100 rpm engine speed. For tests with the carburetor approach, the engine was loaded to 23 kW and the dynamometer scale read, and subsequently the load increased by 7.5 kW, until the maximum increment of 7.5 kW was obtained. The last reading was then obtained by setting the engine to a full load position on dual-fueling.

4.1.1 Measurements

Power and torque were obtained using pto dynamometers. Introductory tests with the conventional tractor and dynamometer provided a maximum torque of 482 Newton-meters at 1000 rpm pto speed.

For the spray-injection method, the dynamometers used were a General Electric cradle type as the main dynamometer which provided accurate measurement of power output, and a M & W Gear hydraulic type as the auxilliary dynamometer which was used to hold the transferred load, as explained earlier in this chapter.

For the carburetor approach, the tractor was connected to a dynamometer through its power-take-off. The dynamometer used was a M & W Gear P-2000 Hydra-Gauge.

For both methods, the engine rpm was read at the tractor's tachometer located in the cabin. Because of dynamometer limitations, the tests with the spray-injection method were done only at 2100 rpm engine speed (1000 rpm pto shaft speed). With the carburetor method, a tachometer located on the dynamometer instrument panel measured the tractor pto shaft speed.

4.1.2 Temperature

The exhaust temperatures for both methods of dual-fueling were measured by a pyrometer gage installed on the tractor.

For the dual-fueling test with carbureted ethanol, copper constantan thermocouples and a "Digistrip II" recorder were used to measure the intake air temperature, which was the same as the ambient temperature, and the air temperature at the tube which connects the turbocharger to the intake manifold. The Digistrip II manufactured by Kaye

Instruments, was programmed to record temperatures at 10 second intervals.

4.1.3 Turbocharger Pressure

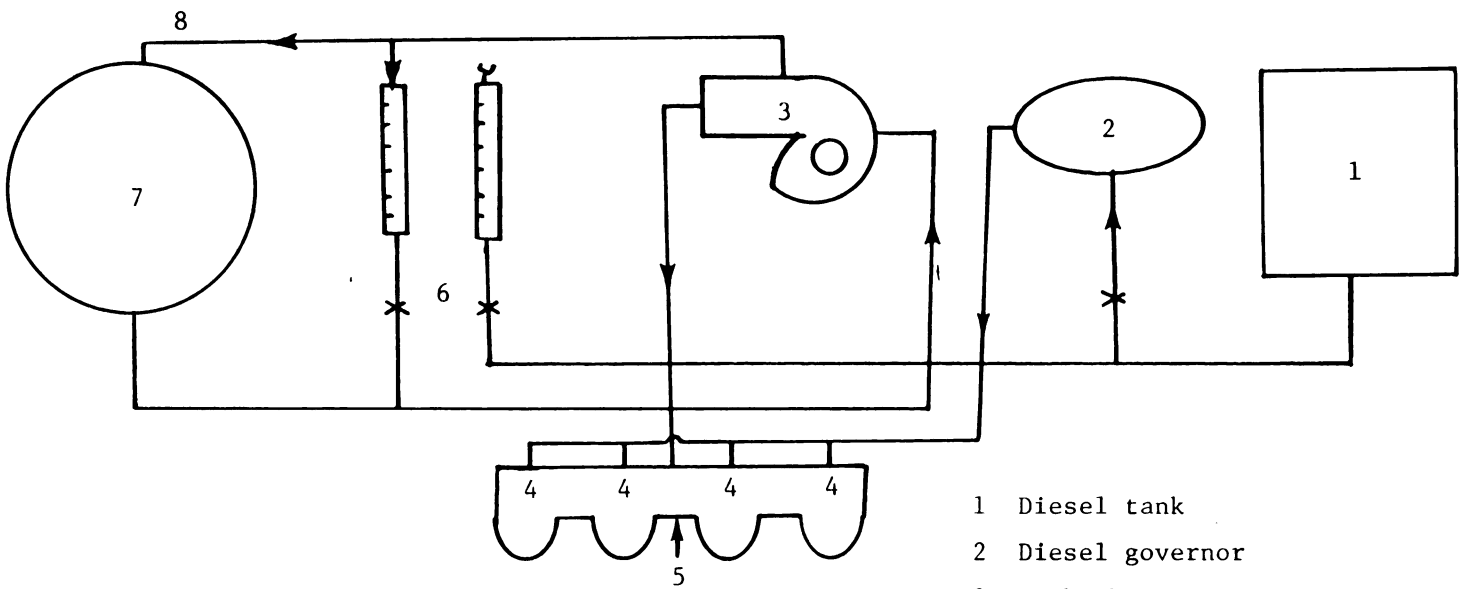
The turbocharger pressure for both methods was measured by a "Stuart-Warner" turbopressure gage number 360H82320Z2 installed on the tractor cabin. The gage read from zero to 30 pounds per square inch (psi) with a sensitivity of ±14 kPa.

4.1.4 Fuel Consumption

A fuel measuring device was built using graduated pippetes to measure the consumption of both diesel and ethanol solutions. A stop watch was used to time the output from two pippetes of calibrated volume. The fuel was then measured under various engine loads, as stated in the Testing Procedure earlier in this chapter (see Figures 3 and 4).

4.1.5 Air Flow

Air flow was measured using a 73.5 mm diameter orifice at the end of a surge tank of 95 liters capacity. Another surge tank of 208 liters capacity was placed between the tractor intake air and the smaller surge tank. The 208 liters capacity surge tank was used to buffer pulsations on the manometer. The air flow measurements were taken by using a water manometer which was used to measure pressure drop across the orifice at the inlet of the surge tank (see Figures 1 and 2).



1 Diesel tank

2 Diesel governor

3 Turbocharger

4 Engine injectors

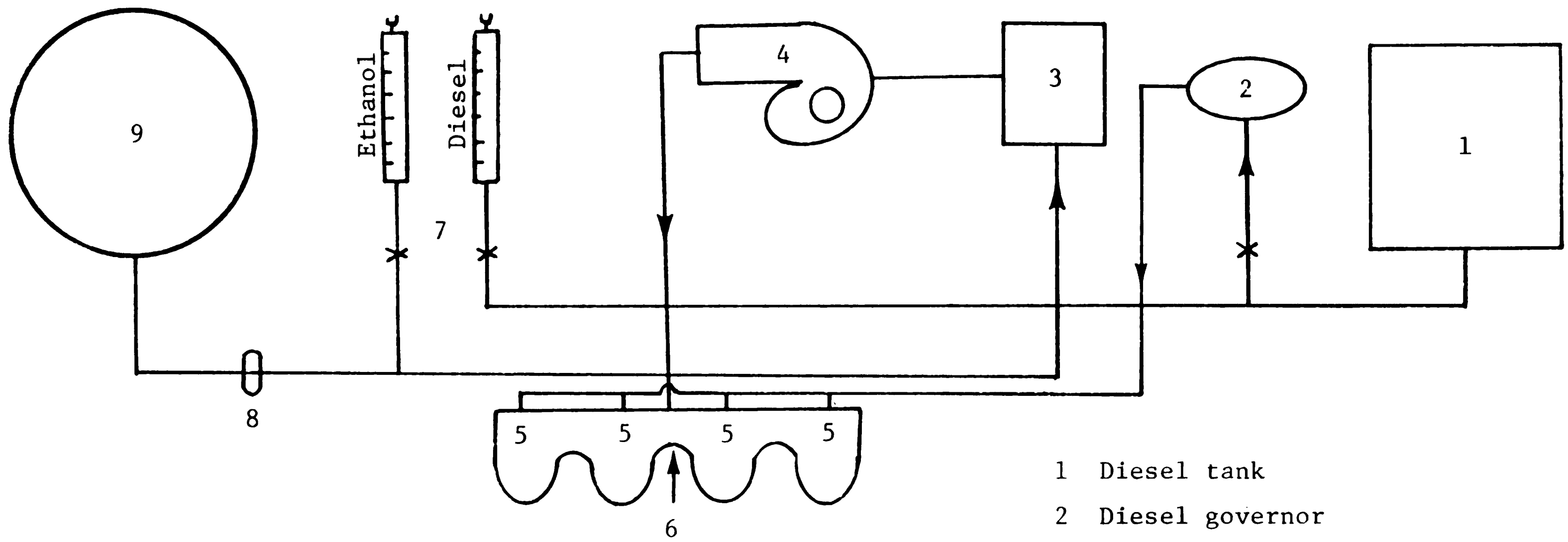
5 Engine intake manifold

6 Pipettes

7 Ethanol tank

8 Pressure line

Figure 3. Schematic of fuel measurement apparatus for spray injecting ethanol on a turbocharger tractor.



1 Diesel tank

2 Diesel governor

3 Carburetor

4 Turbocharger

5 Engine injectors

6 Engine intake manifold

7 Pipettes

8 Fuel pump

9 Ethanol tank

Figure 4. Schematic of fuel measurement apparatus for carbureting ethanol on a turbocharged tractor.

4.2 Procedure of Analysis

Because the internal combustion engine is a complex device, a careful analyses of variables is necessary to provide or to verify new design concepts. To ensure a correct analyses, the following performance factors were used.

4.2.1 Fuel Consumption and Thermal Efficiency

For every pound of air inducted into the engine, a proportionate amount of fuel should be inducted. Hence, the brake specific fuel consumption in Kg per hour is proportional to the air consumption. The fuel consumption is a comparative parameter that shows how efficiently an engine is converting fuel into work. This parameter has been used more often and is preferred by the author, rather than thermal efficiency, because all quantities are measured in standard and accepted physical units.

Any form of internal combustion engine is a steady-flow machine with air and fuel entering at atmospheric pressure and temperature and products of combustion leaving at atmospheric pressure. For these flow conditions, the heat of combustion that can be obtained from the air-fuel mixture with the use of a calorimeter, is the heat of combustion at constant pressure. The combustion engine produces work and releases heat only as a by-product. Thus, the ratio of work obtained from the engine to the energy input is called "brake thermal efficiency." Brake thermal efficiency is an indication of how well the engine is utilizing the energy contained in the fuel. This parameter was computed from the product of power in kW and a constant divided by the sum of the product of high heat value and mass flow rate of each fuel in every test run (see equation in Appendix B). The high heat values of the fuels were extracted from Table 1.

4.2.2 Volumetric Efficiency

Since both air and fuel are partners in the combustion process, it is evident that both are equally important. However, the volume occupied by a liquid or gaseous fuel is a fraction of the volume occupied by the air. For this reason, the induction of the air presents the greatest problem. If the engine does not induct the largest possible amount of air, the power output of the engine will be restricted, no matter how much fuel is added. A basic requirement for an engine is its capacity for inducting a large amount of air during each intake stroke. The mass of air inducted by the engine per intake stroke is called the unit air charge or actual air capacity.

The volumetric efficiency is the actual air capacity, when divided by the theoretical air capacity, which is the mass of air that would fill the displacement of the cylinders at inlet temperature and pressure, becomes the volumetric efficiency. The actual air capacity was calculated by entering the value of pressure difference for each test run in the continuity equation (see Appendix B). Pressure differences were read at the manometer. The theoretical air capacity was obtained from the product of engine rpm, engine displacement and density of the air.

4.2.3 Air-Fuel Ratio

The work developed by the engine depends directly upon the amount of energy released when a mixture of air and fuel burns. To understand the chemistry of combustion, the air-fuel ratio parameter was created. This mass ratio shows the relative portions of air and fuel inducted. This ratio when compared with the stoichiometric air-fuel ratio (theoretical air-fuel ratio). This mass ratio was calculated by dividing the actual air capacity, which was obtained as explained before, by the mass of fuel used at the run. This mass of fuel was obtained in accordance with the time spent to empty the pippete at the respective run, and density of the fuel used.

4.2.4 Peak-Power

For a given rpm the engine power can be shown by either torque or brake horsepower, depending on the type of work for which the engine was designed. The crankshaft power from an engine is called brake horsepower, and it is measured with all engine components functioning. Torque is not strongly dependent on the speed of the engine, but depends on the volumetric efficiency and friction losses (16). Thus, if the mass of air of the engine increases by dual-fueling, because the mass of air increases as it cools, torque also increases accordingly. Because the engine tests were done at constant engine speeds, and horsepower is proportional to the product of torque and speed, peak horsepower was one of the major parameters analyzed. Peak horsepower was read directly from the dynamometer scale with the engine running at full load.

4.2.5 Exhaust Temperature

Since the C.I. engine inducts a constant amount of air on the intake stroke at a given rpm, a small amount of fuel injected into the engine will not require all of the air in the cylinder. This occurs at part load. As the load is increased, greater amounts of fuel are injected and more and more of the air is required for combustion. At some stage, further injection of fuel leads to part of the fuel not being fully oxidized and to a more rapid increase of the exhaust temperature. Hence, the exhaust temperature is another way to analyze performance. This parameter was measured directly with a pyrometer for each respective run.

4.2.6 Turbocharger Pressure

A turbocharger, as the name implies, is a compressor coupled to a turbine which receives its driving power from the exhaust gases from the engine. The function of the turbocharger is to increase the mass of intake air, allowing more fuel to be combusted. Consequently, output power is increased. There is a direct correlation between the centrifugal compression on the turbocharger and the speed of the impeller (16). Therefore, the ratio of turbopressure and atmoshperic pressure is also related. The pressure ratio and mass flow rate are fixed by the volume demanded by the engine (16). If a greater mass flow rate is desired to obtain more power, the compressor speed must be raised (engine speed constant) with consequent increase in the pressure ratio. But with this increased inlet pressure to the engine, the volumetric efficiency increases and a greater volumetric air flow is received by the engine. Thus, the air capacity of the engine at constant speed varies with varying compressor speed. Because the carburetion of ethanol changes the intake air density, and consequently the mass flow rate and pressure ratio, the turbopressure also was an object of interest in this investigation.

4.2.7 Maximum Proportion of Fuel Energy from Ethanol

The heating value of a fuel is colloquially called the heat of combustion and is determined by burning the fuel with oxygen in a bomb calorimeter and noting the temperature rise of a cooling bath. The amount of heat transferred to the cooking bath will depend, in part, on whether all or part of the water vapor formed by combustion is condensed. If all of the water vapor can be condensed, a higher heating value is obtained. If none of the water vapor is condensed, a lower heating value is obtained.

The maximum proportion of fuel energy from ethanol was computed from the product of the high heat value of the ethanol, mass of fuel used at each run, and the percent of ethanol in the mixture, divided by the sum of the product of high heat value and mass flow rate of each fuel for each run. The high heat value of the fuels were extracted from Table 1. The procedure of determination of these values was explained above.

Ethanol has about two-thirds the energy content of diesel fuel and gasoline. However, it improves power output because of its high latent heat of vaporization.

How the variables discussed above affect engine performance will depend, in part, on the system used to dual-fuel turbocharged diesel engines.

CHAPTER 5

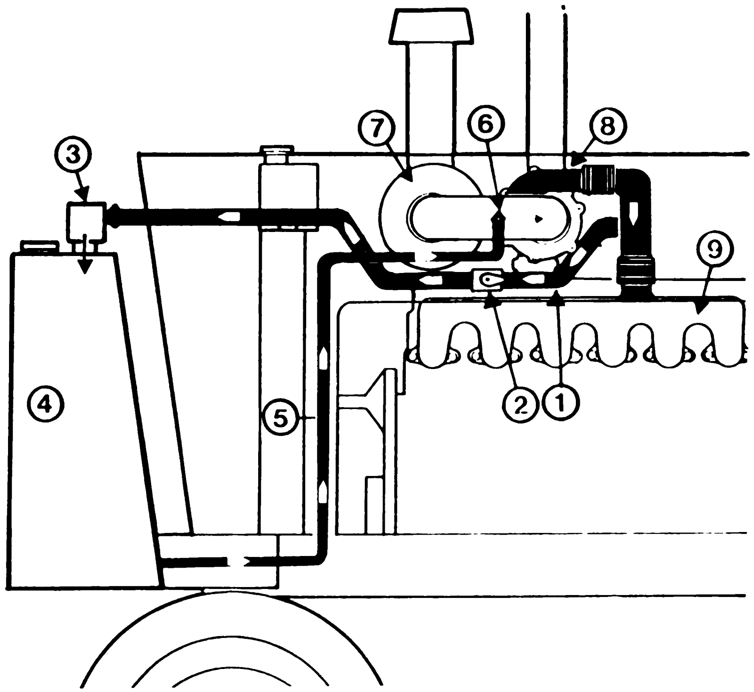
DUAL-FUELING THROUGH SPRAY-INJECTION

This system uses the pressure from the turbocharger to inject the mixture of water and ethanol through a feed line and filter. The ethanol solution flows through an orifice that meters the mixture into the air stream going into the intake side of the turbocharger.

5.1 Equipment Description

The spray-injection kit consists of a 132 liter tank, an air supply line, ethanol/water mixture (aquahol) supply line, an orifice, a fuel filter, an instrument panel, and the necessary valves and fittings. The system is automatically activated by turbocharger pressure to provide a preset flow of ethanol/water solution. When turbocharger pressure drops below a preset point, the flow of ethanol/water mixture is stopped. The turbocharger pressure varies with engine load and speed.

The air supply line connects to the air outlet pipe between the turbocharger and the intake manifold. During operation, the line carries pressurized air to an air/fuel separator on top of the fuel tank. This separator keeps the ethanol/water mixture from backing up into the air line. A fuel drain is provided at the bottom of the tank. The ethanol/water mixture supply line connects the tank through a filter to a turbocharger inlet. The system is shut off by closing a valve in the alcohol supply line (see Figure 5).



1	Pressure line	6	Check valve
2	Shut-off valve	7	Turbocharger inlet
3	Water/air separator	8	Turbocharger
4	Storage tank	9	Intake manifold
5	Alcohol feed line		

Figure **5**. Diagram of alcohol injection on a turbocharged diesel tractor, M & W Gear Company (13).

5.2 Performance*

The performance of the tractor was analyzed using a series of six parameters in order to evaluate the optimum performance in terms of maximum brake power, thermal efficiency, volumetric efficiency, exhaust temperature, fuel consumption, and turbocharger pressure. The results are described under the following appropriate sub-headings.

5.2.1 Maximum Brake Power

Although various ratios of air and fuel can be burned in the diesel engine, it was found that a definite ratio was required to obtain maximum torque at a given speed. Thus, the richest ethanol-air ratios that could be used without knock, in combination with the slightly lower maximum diesel fuel-injection, were able to produce about 36 percent more than the engine maximum power when fueling with diesel alone (see Figures 6 and 7).**

Data presented also in Figures 6 and 7 shows that the maximum power, when using ethanol for all ethanol/water solutions, peaked higher than when operated with conventional fuel. The 50 percent ethanol by volume shows the best results of maximum replaceable energy combined with best thermal efficiency. The peak power was reduced as the percentage of water in the ethanol/water solution decreased.

Maximum power values obtained by dual-fueling were higher than the 54 kW maximum power obtained with diesel fuel alone. The 54 kW of power obtained with diesel fuel alone was designated as 100 percent power.

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*System output performance of the tests are reported in data tables of Appendix D.

**All the figures in this dissertation refer to tests done at 2100 rpm engine speed, unless otherwise specified. Each point in the figures is an average of three readings.

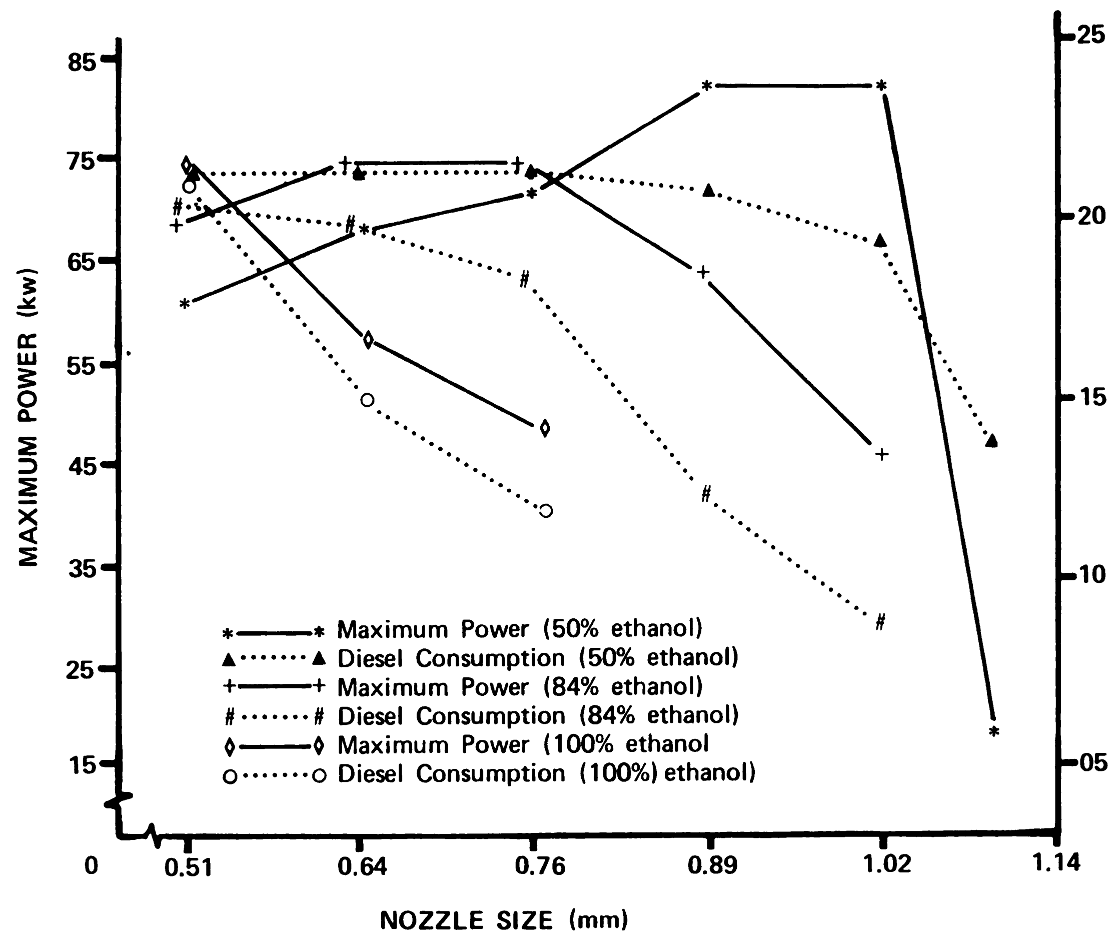


Figure 6. Effects of various nozzle sizes on engine maximum power and diesel consumption with different alcohol solutions.

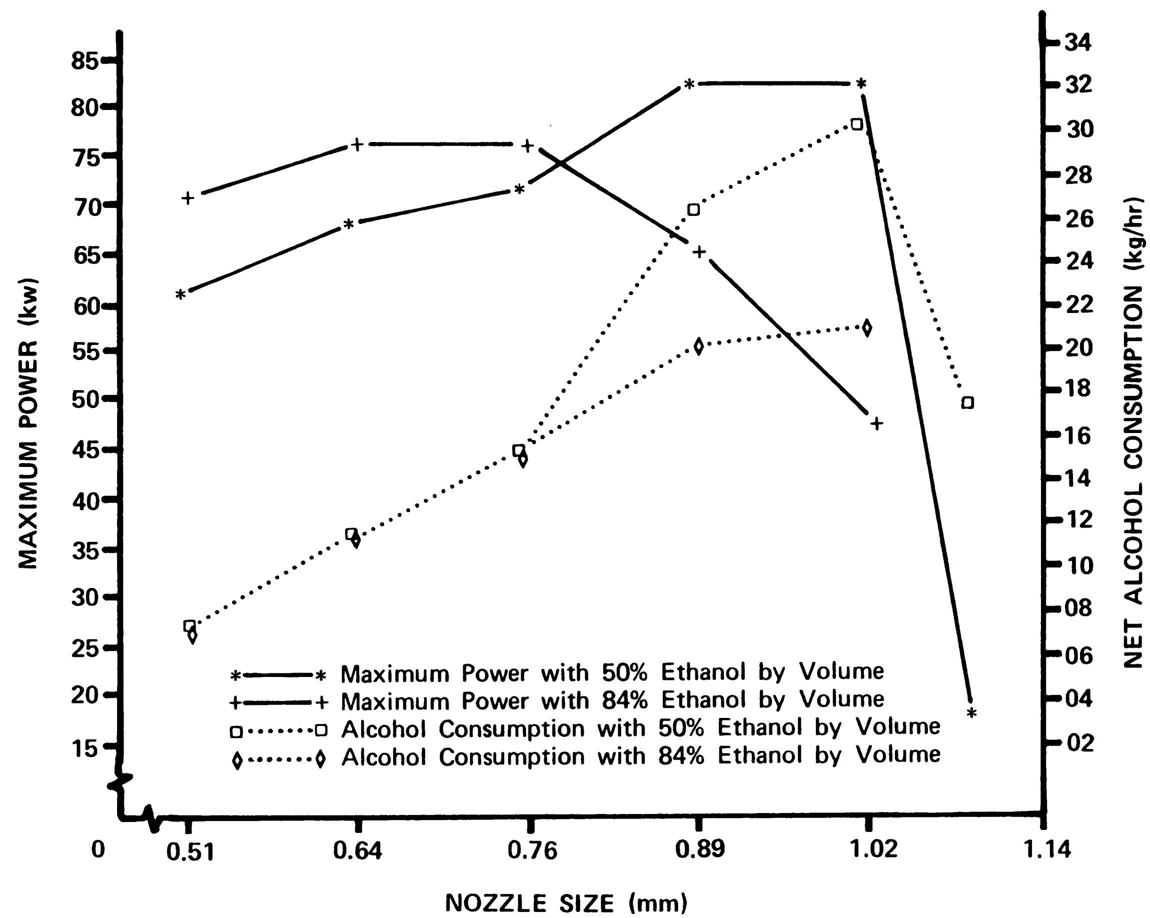


Figure 7. Effects of various nozzle sizes on engine maximum power and alcohol consumption for different alcohol solutions.

Power values obtained with different ethanol mixtures were compared with the diesel fuel standard of 100 percent (Table 3).

Table 3. Percentages of maximum tractor power obtained using different ethanol/water mixtures at different injection rates.

Spray injected liquid	Nozzle sizes (mm/10)					
	5.1	6.4	7.6	8.9	10.2	10.9
100 percent ethanol per volume	128%	96%	80%	*	--	--
84 percent ethanol per volume	121%	132%	132%	115%	83%	--
50 percent ethanol per volume	105%	119%	126%	145%	145%	33%
Distilled water	96%	96%	--	--	--	--

*The test was done by increasing torque to the point where knock was heard. The dashes indicate that no value was read, because knock restricted the measuring of power when using the larger nozzle sizes in some cases.

5.2.2 Thermal Efficiency

Data presented in Figure 8 indicates that for the same torque levels, the 84 percent ethanol by volume provided a higher brake thermal efficiency. Brake thermal efficiency for the other fuels tested was shown to be close to that of conventional diesel fuel at low torque levels.

5.2.3 Volumetric Efficiency

As indicated in Figure 9, the volumetric efficiency increases as torque increases. This is because the level of turbocharging increases

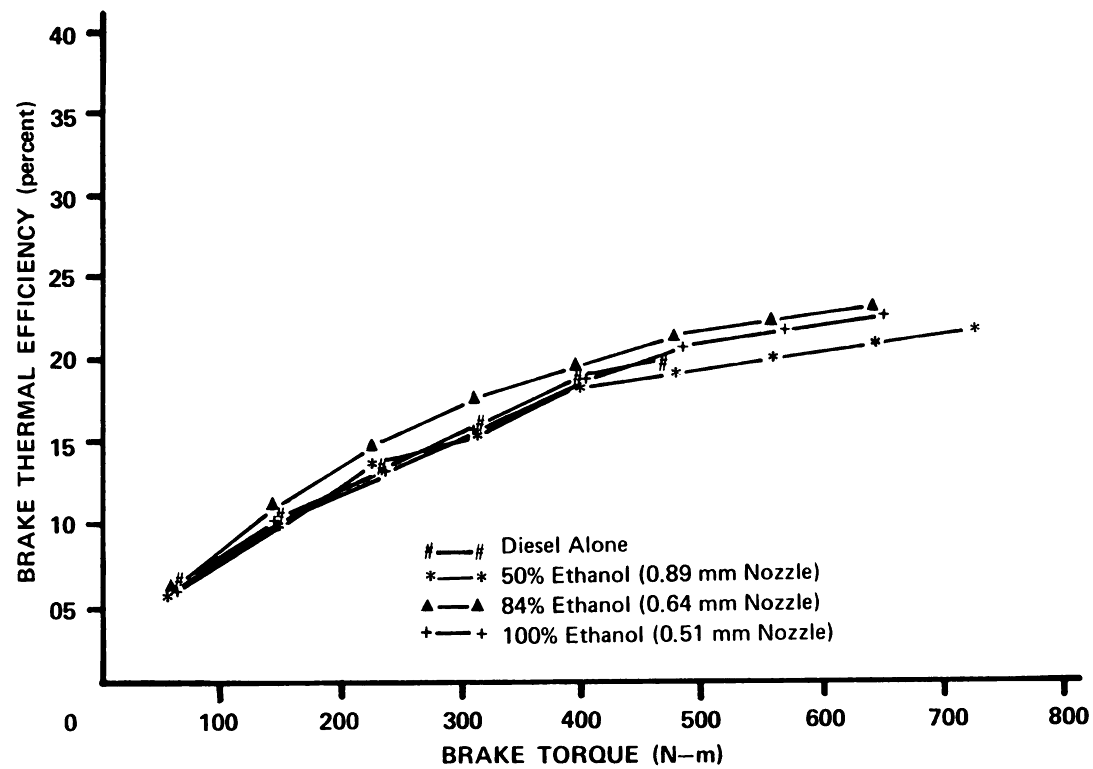


Figure 8. Effects of spray injected alcohol on brake thermal efficiency at various torque levels.

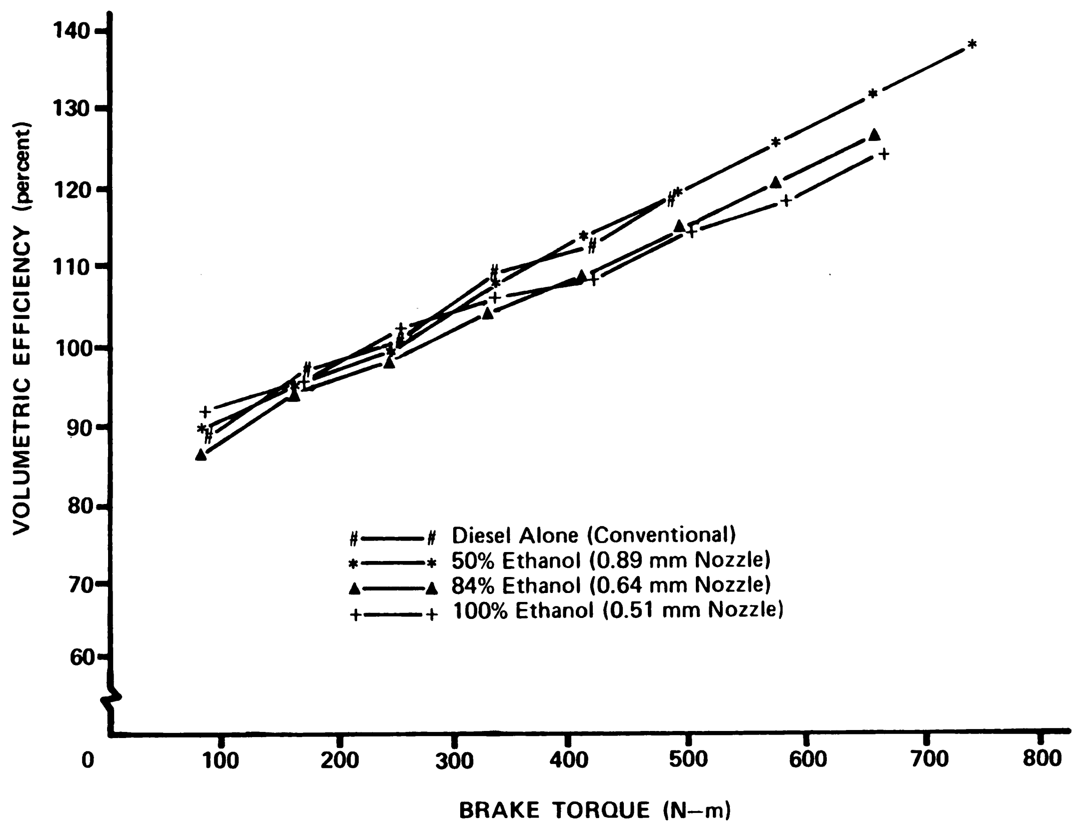


Figure 9. Volumetric efficiency at best nozzle sizes for various alcohol/water mixtures.

as the energy in the exhaust increases which drives the compressor faster.

Because of the high latent heat of vaporization of ethanol (cooling effects cause more air to be packed into the cylinder), higher volumetric efficiency was expected when using ethanol solutions; this effect, however, was not observed, except for the 50 percent ethanol solution. This was because the cooling effects of the alcohol decreases volumetric efficiency of the engine because of lower temperature and pressure in the manifold. Furthermore, the turbocharger in the tractor provided a constant-pressure increase. In the constant-pressure system, the objective is to hold the exhaust pressure at a constant and higher pressure than atmospheric pressure so that the turbine can operate at an optimum efficiency (16). The cooler burning of the ethanol solution might have affected the turbocharger efficiency, since it depends upon the exhaust-gas temperature and the turbocharger release pressure. It can also be seen in Figure 9, that the volumetric efficiency increased as the water content in the alcohol increased. This effect was expected because the higher latent heat of vaporization of the water cools down the air, thus packing more air through the turbocharger.

5.2.4 Exhaust Temperature

For all levels of torque, the exhaust temperature was lower for the spray-injected ethanol/water solutions and it decreased as the portion of water was increased (see Figure 10). The reason behind this is that alcohol burns at a lower temperature and the alcohol heat content

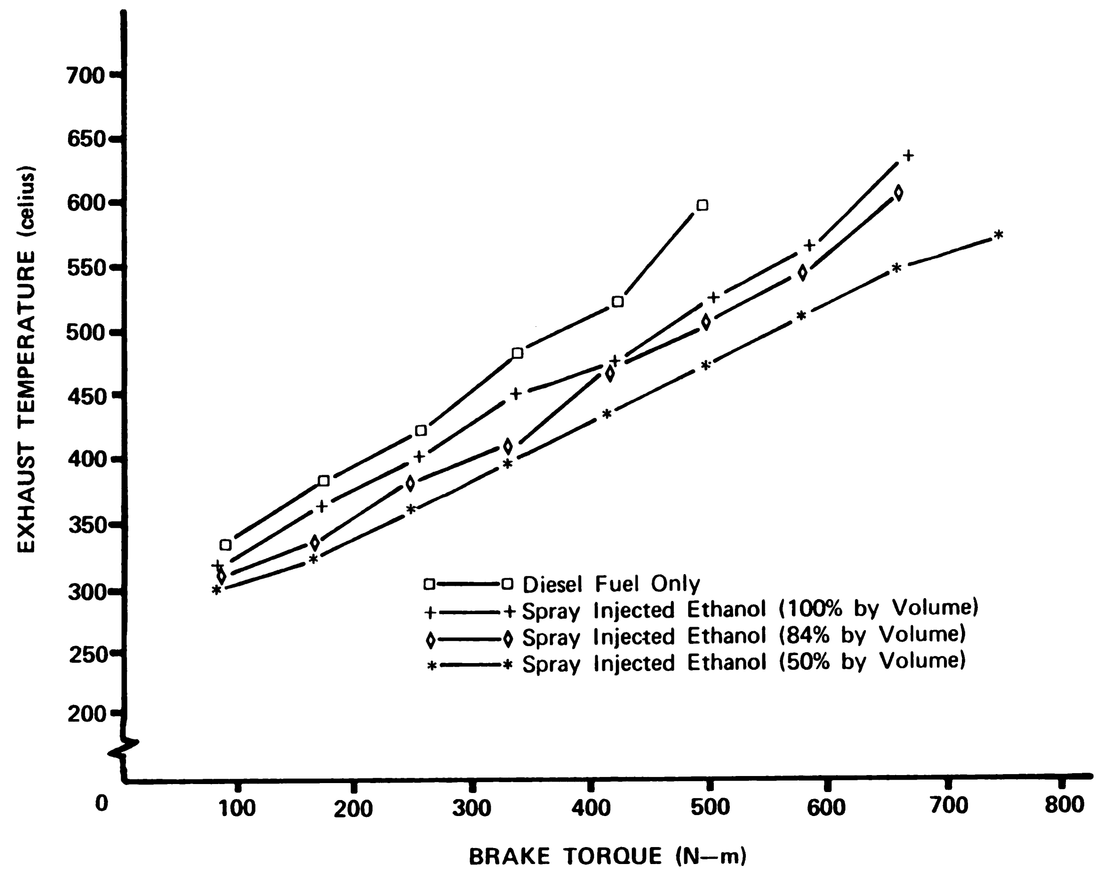


Figure 10. Exhaust temperature at various levels of engine loads for all fuel mixtures tested.

is lowered with added water. Furthermore, the cooling effects of water which has a high latent heat of vaporization, contributes even more to this phenomenon.

5.2.5 Fuel Consumption

Figures 11 through 14 illustrate the amount of diesel fuel in Kg/hr that can be saved by using the spray-injection approach. As Figures 11 and 12 show, the amount of ethanol solution used is about twice as great as the amount of diesel displaced on a mass basis. This means that there is nearly a direct trade-off of fuels in terms of energy content, since the ethanol contains about half of the diesel energy per unit volume.

Figure 13 shows the fuel consumption in conventional and dual-fueled diesel tractors by means of spray-injection with different ethanol solutions. The maximum net ethanol used to displace ethanol solution was the most suitable in reaching the maximum ethanol/water mixture consumption with engine best performance.

The maximum proportion of fuel energy supplied on different ethanol/water solutions in combination with different nozzle sizes are shown in Figures 15 through 18. As expected, the energy supplied increases as the nozzle size increases, up to the point where the maximum allowable replacement energy is reached. Then it decreases, as seen for the 10.9 mm nozzle size curve in Figure 15. It is also shown that, as the percentage of water in the ethanol decreases, the fuel energy supplied by the alcohol to the engine decreases, without affecting its performance. One exception is the 84 percent alcohol which contained

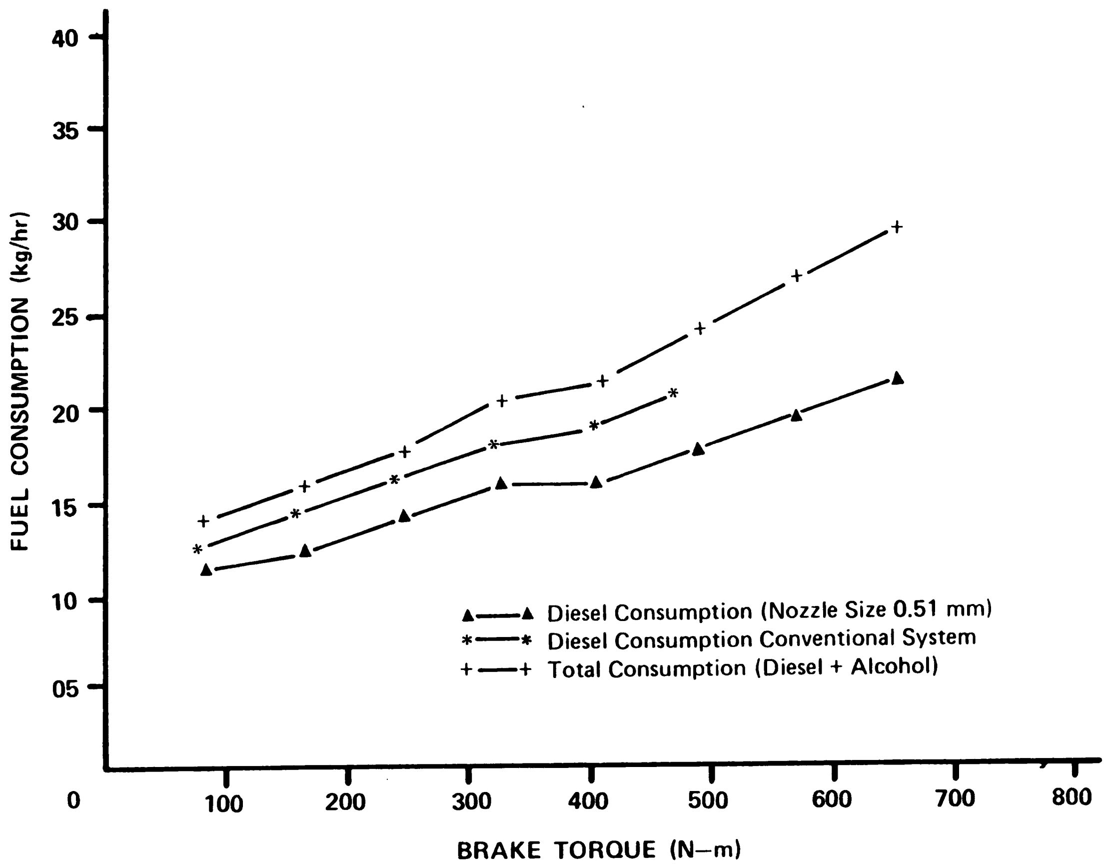


Figure 11. Fuel consumption in conventional and dual-fueled diesel tractors with 100% ethanol solution.

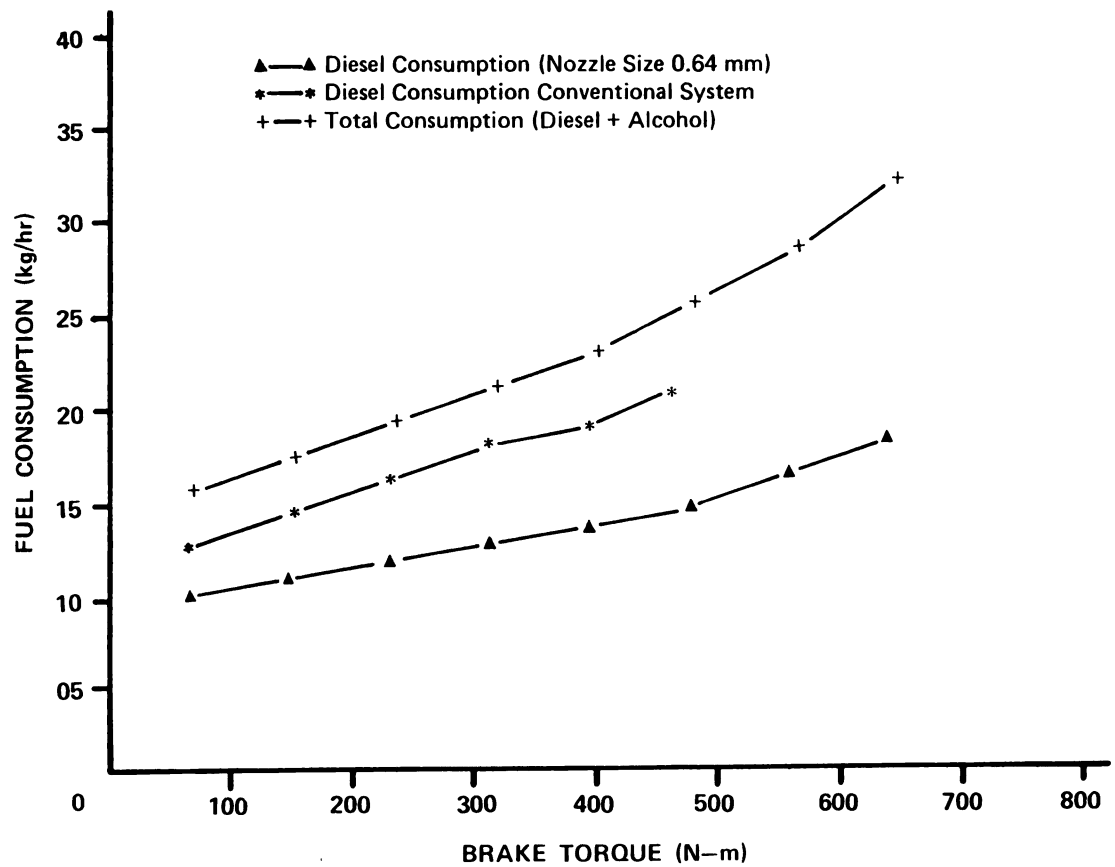


Figure 12. Fuel consumption in conventional and dual-fueled diesel tractors with 84% ethanol solution.

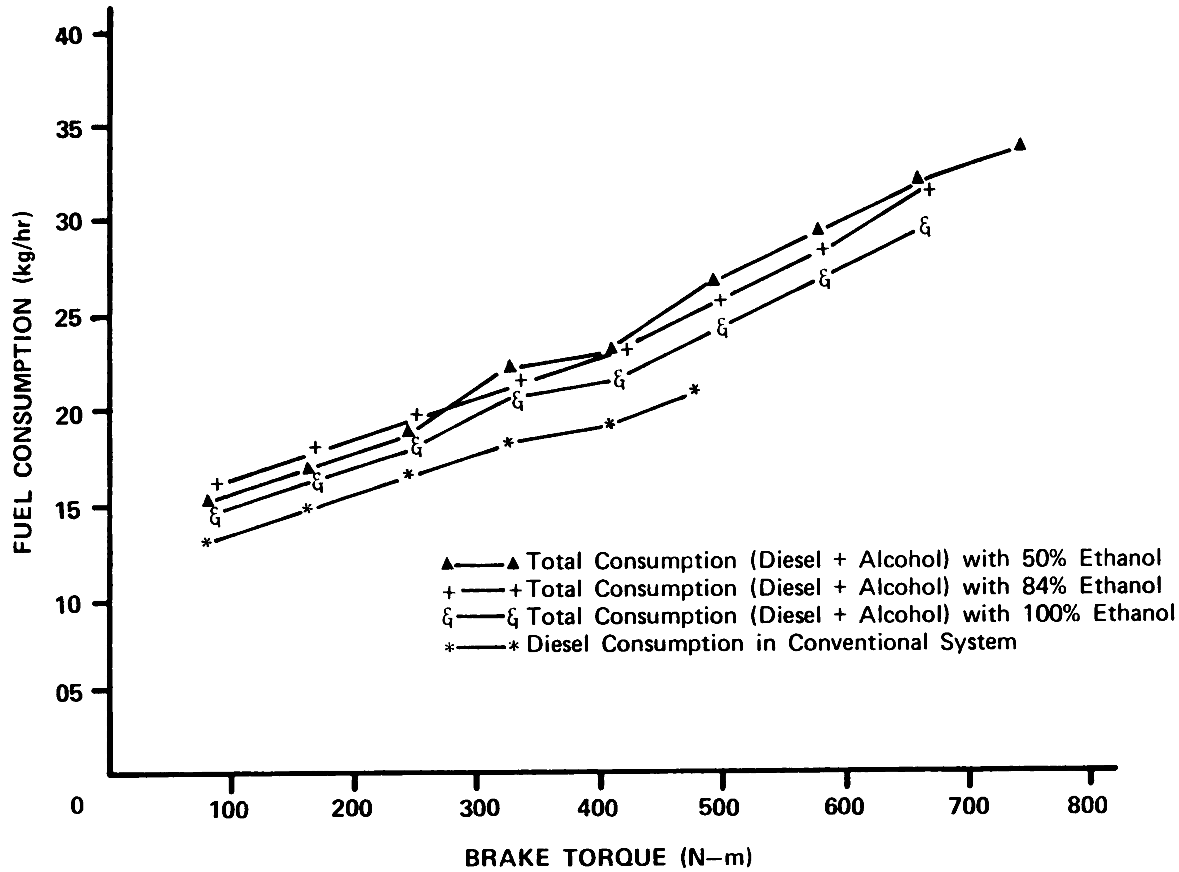


Figure 13. Fuel consumption in conventional and dual-fueled diesel tractors with different ethanol solutions.

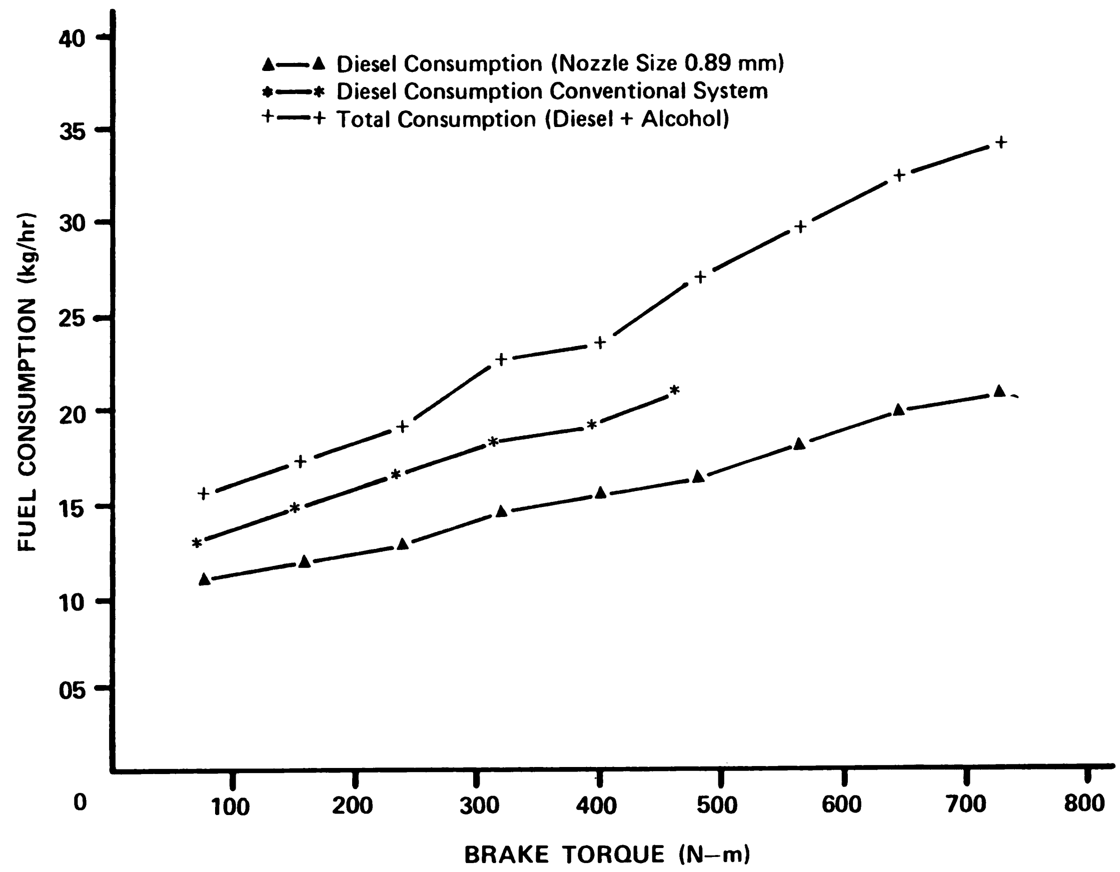


Figure 14. Fuel consumption in conventional and dual-fueled diesel tractors with 50% ethanol solution.

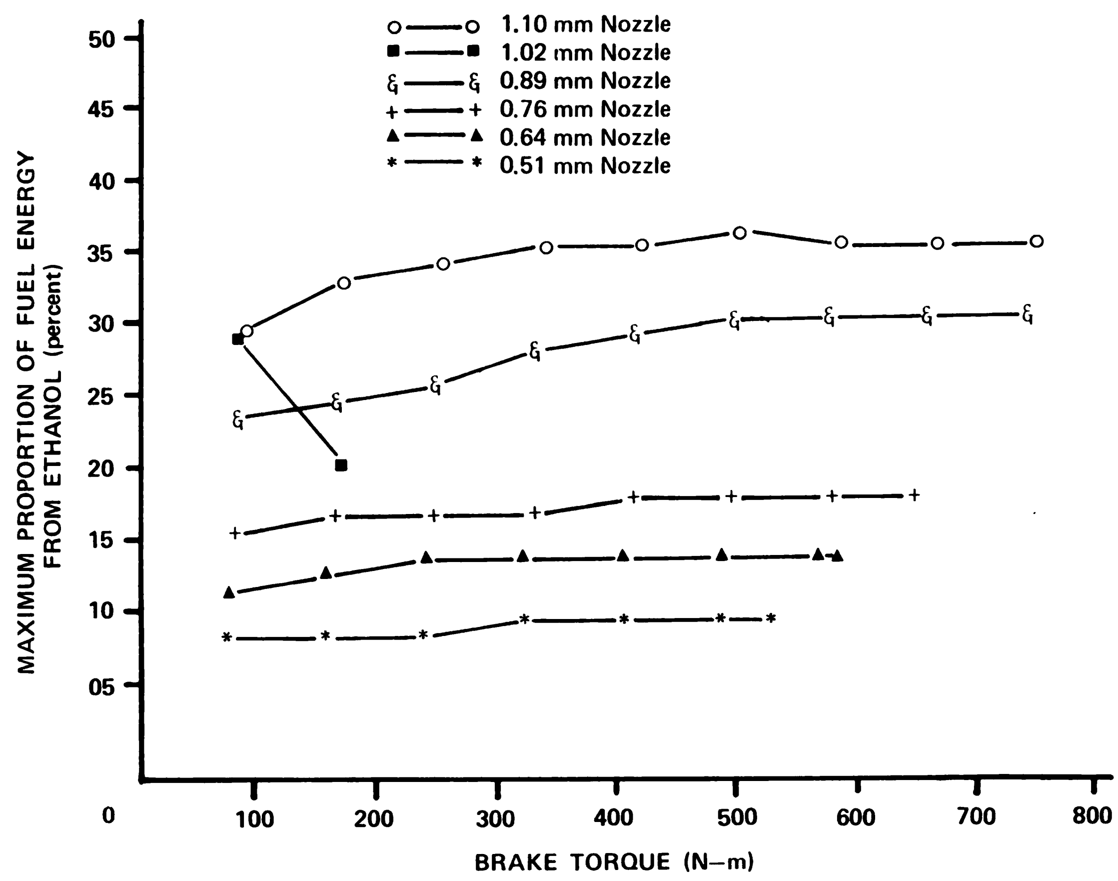


Figure 15. Maximum proportion of fuel energy displaced with a 50% ethanol solution for different nozzle sizes.

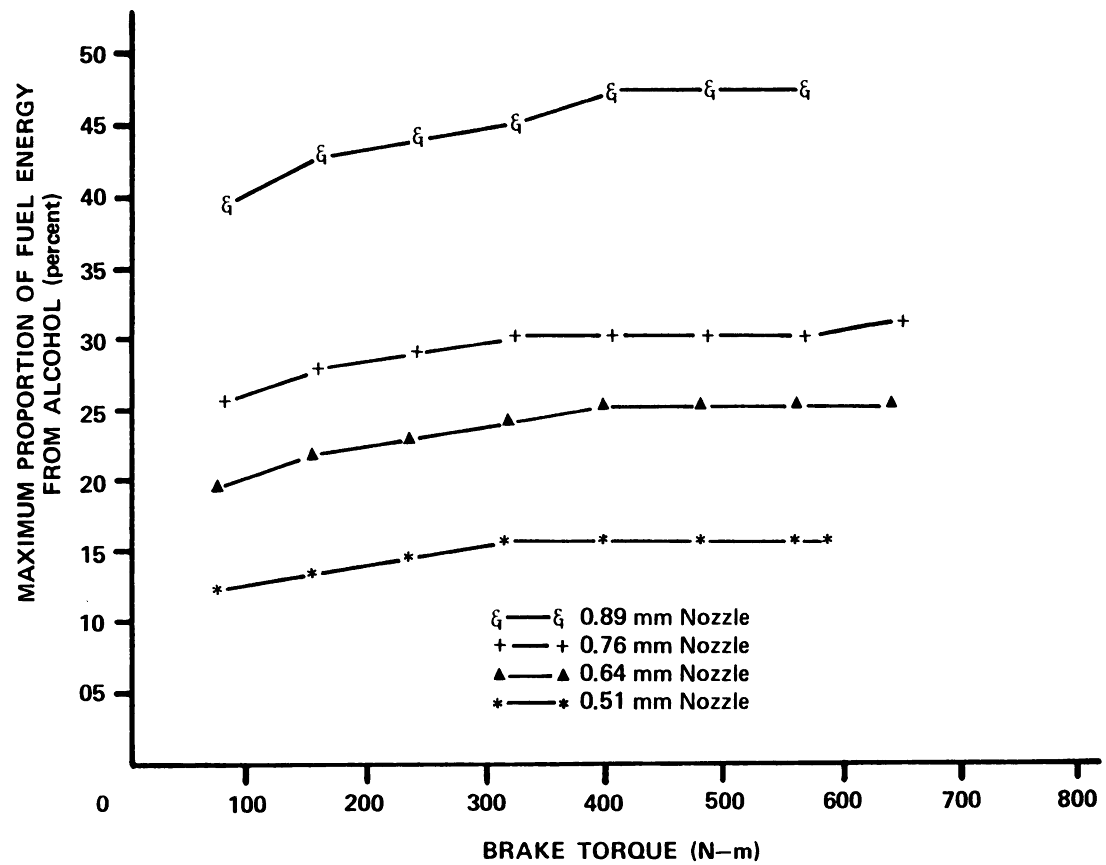


Figure 16. Maximum proportion of fuel energy displaced with an 84% ethanol solution, for different nozzle sizes.

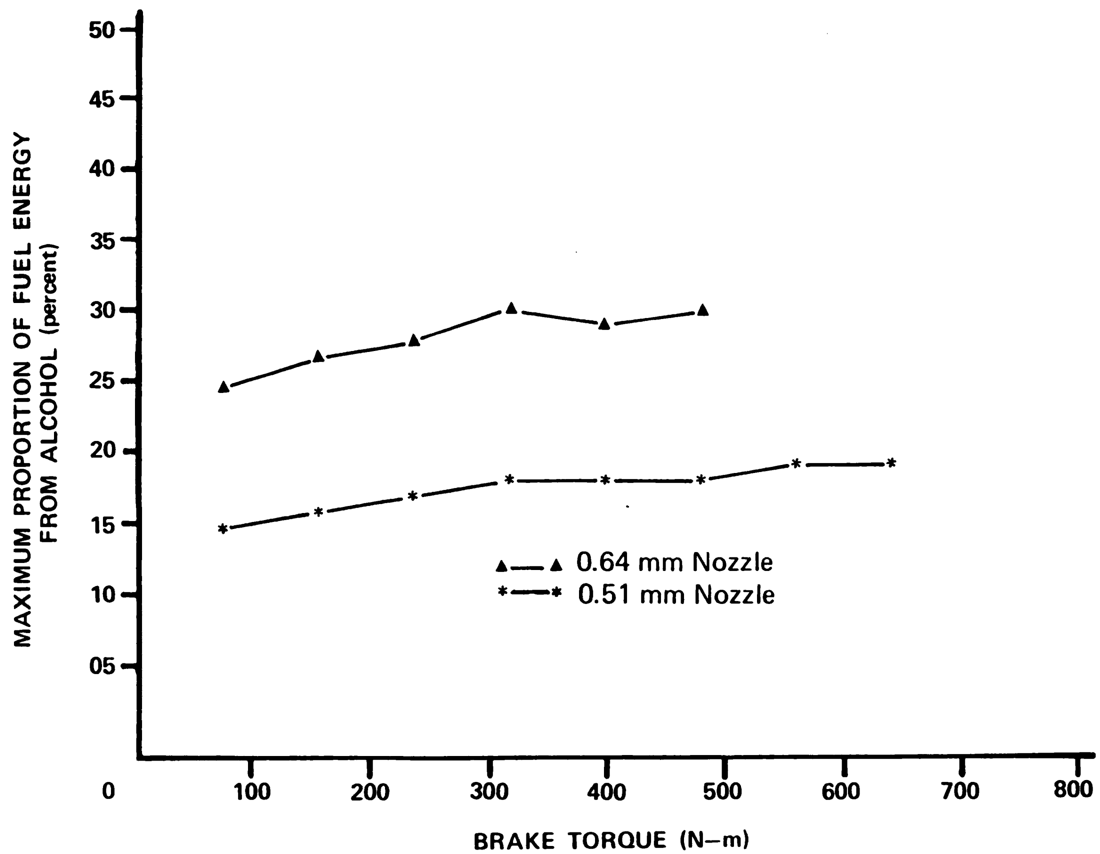


Figure 17. Maximum proportion of fuel energy displaced with 100% ethanol at different nozzle sizes.

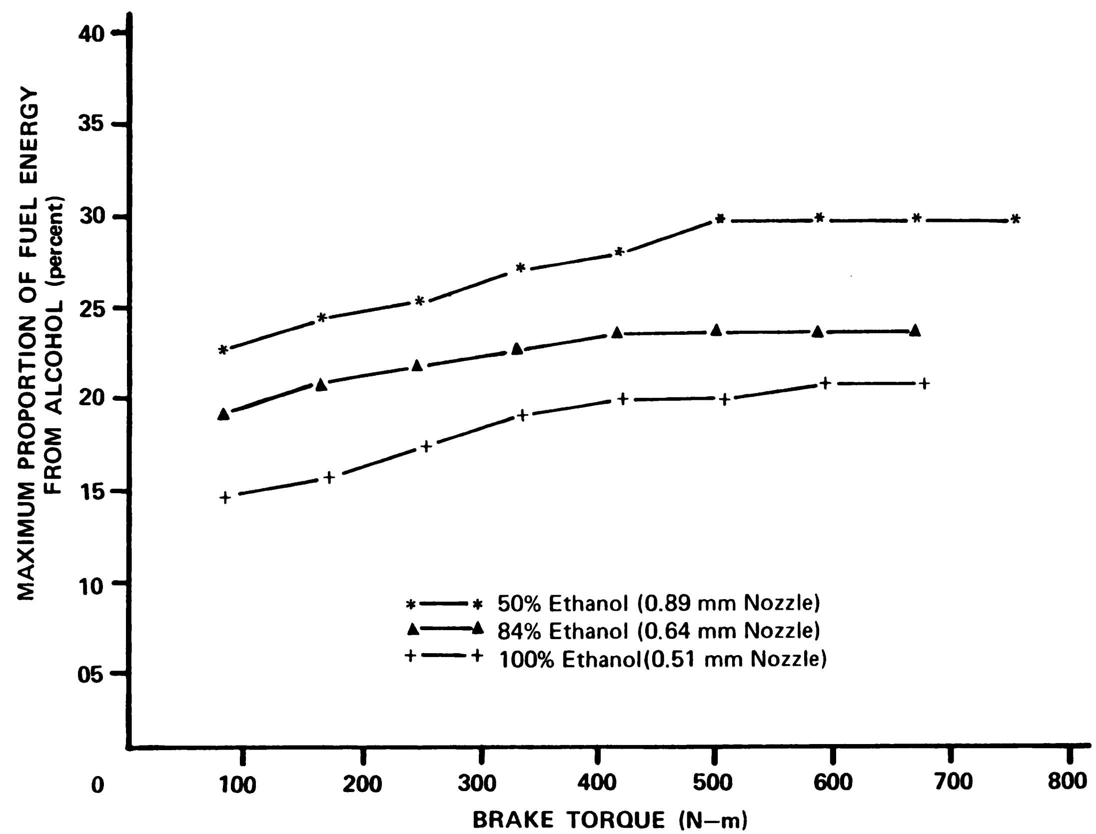


Figure 18. Maximum proportions of fuel energy at various torque levels that could be supplied by alcohol without affecting the engine maximum power for different ethanol solutions.

a lower proportion of water, but did not provide maximum torque as high as that obtained when using the 50 percent ethanol solution.

The effects of water in the ethanol without affecting the engine performance, and the maximum proportion of fuel energy from ethanol are presented in Figure 18. As it shows, the 50 percent ethanol by volume demonstrated best results. Furthermore, the maximum portion of energy that could be supplied by ethanol without sacrificing engine performance was limited to 30 percent. The energy supplied by ethanol with good performance declined as the amount of water in the alcohol increased.

5.2.6 Turbocharger Pressure

The turbocharger pressure values decreased with the use of alcohol. The values of turbocharger pressure for spray-injecting different alcohol concentrations did not change much at lower concentrations of ethanol. On the other hand, the waterless ethanol presented higher turbopressure than the ethanol/water mixtures (see Figure 19).

5.3 Summary

Table 4 presents a summary of the tractor performance with the spray-injected approach. As it shows, the 50 percent ethanol solution with the 0.89 mm nozzle size achieved the highest degree of replacement of diesel in combination with maximum torque and brake thermal efficiency.

5.4 Effects of Water Injection On the Engine Performance

Alcohols are cheaper to produce with higher water content. Since steam produced at the combustion of the diesel creates a controversy about increasing power when injected in the combustion chamber, the

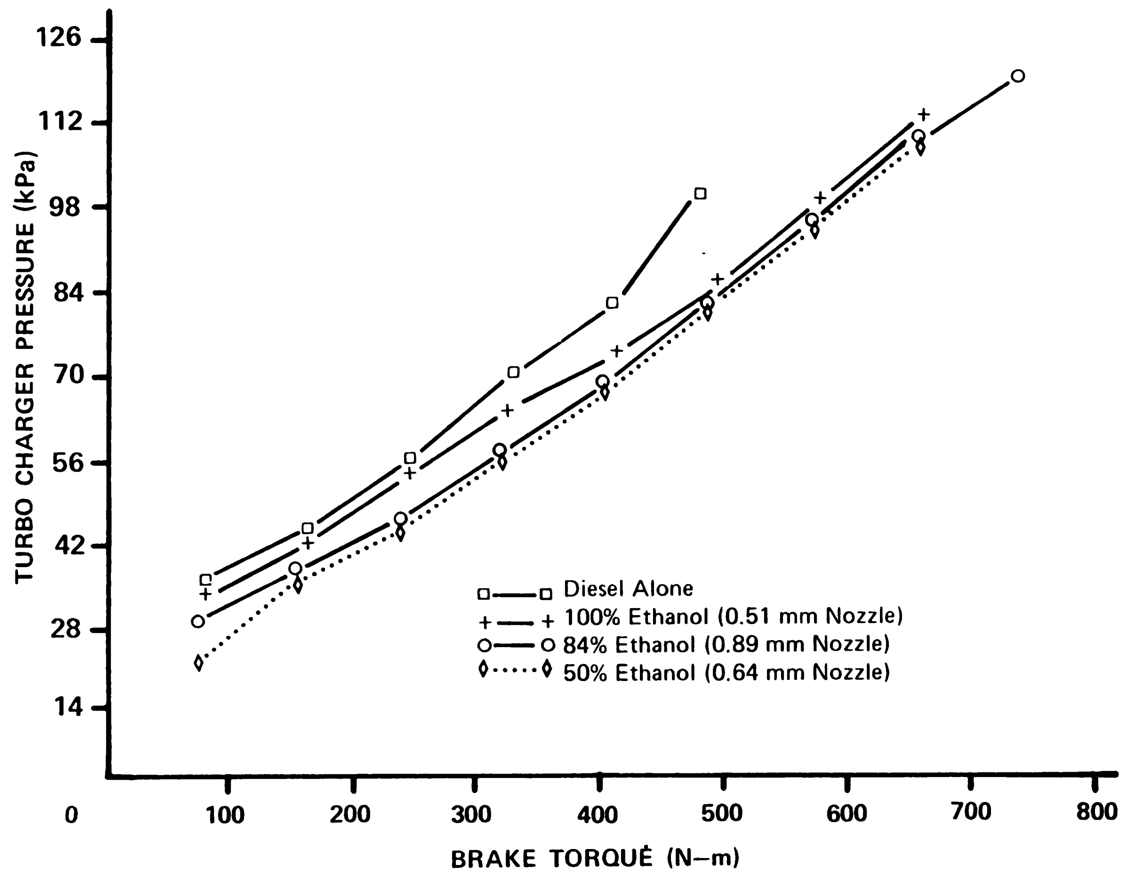


Figure 19. Effects of using different alcohol/water mixtures on turbocharger pressure.

Table 4. Effects of nozzle size and percent of ethanol/water solution on engine performance.

Nozzle size (mm)	Ethanol/water solution (%)	Maximum B.T.E. (%)	Maximum torque (N-m)	Maximum proportion of replaceable energy (%)
5.1	100	23	651	19
	84	23	598	15
	50	20	535	10
6.4	100	21	488	32
	84	23	651	23
	50	22	584	14
7.6	100	19	407	32
	84	23	651	31
	50	22	635	18
8.9	100	--	--	--
	84	21	569	46
	50	23	732	29
10.2	100	--	--	--
	84	18	407	56
	50	21	732	34
10.9	100	--	--	--
	84	--	--	--
	50	8	163	29

(Tests done at 1200 rpm engine speed)

effects of spray-injected water and diesel fuel were also investigated. As seen in Figures 20 and 21, the brake torque, brake thermal efficiency, and volumetric efficiency decreased as the water flow increased. The peak torque is comparable with conventional fueling at lower water flow, as seen on the .51 mm nozzle size curve (see Figure 20). Even though the volumetric efficiency is higher than conventional for both nozzles, an increase in water flow (nozzle size) caused a substantial decrease in equivalent brake torque compared to the low water flow. With the conventional fuel system the volumetric efficiency was also increased.

Data presented in Figure 22 shows that, as expected, the exhaust temperature decreased when distilled water was injected into the engine. The exhaust temperature for the same torque levels, was comparable with alcohol. When higher mass of water mixed with ethanol was used, the exhaust temperature dropped quite substantially.

5.5 Problems Encountered

The clogging of the smaller nozzles, as well as the alcohol line filter, were the only minor problems encountered with the spray-injection approach. However, these problems cannot be attributed to the spray-injection method, but to the impurities remaining in the alcohol after the distilling process.

This problem was encountered only at the beginning of the experiment, because the tests were done as part of a larger project at Michigan State University, which entailed the use of alcohol distilled from corn. Initially, the author's associates who were responsible for distilling alcohol from corn, encountered problems in producing alcohol without

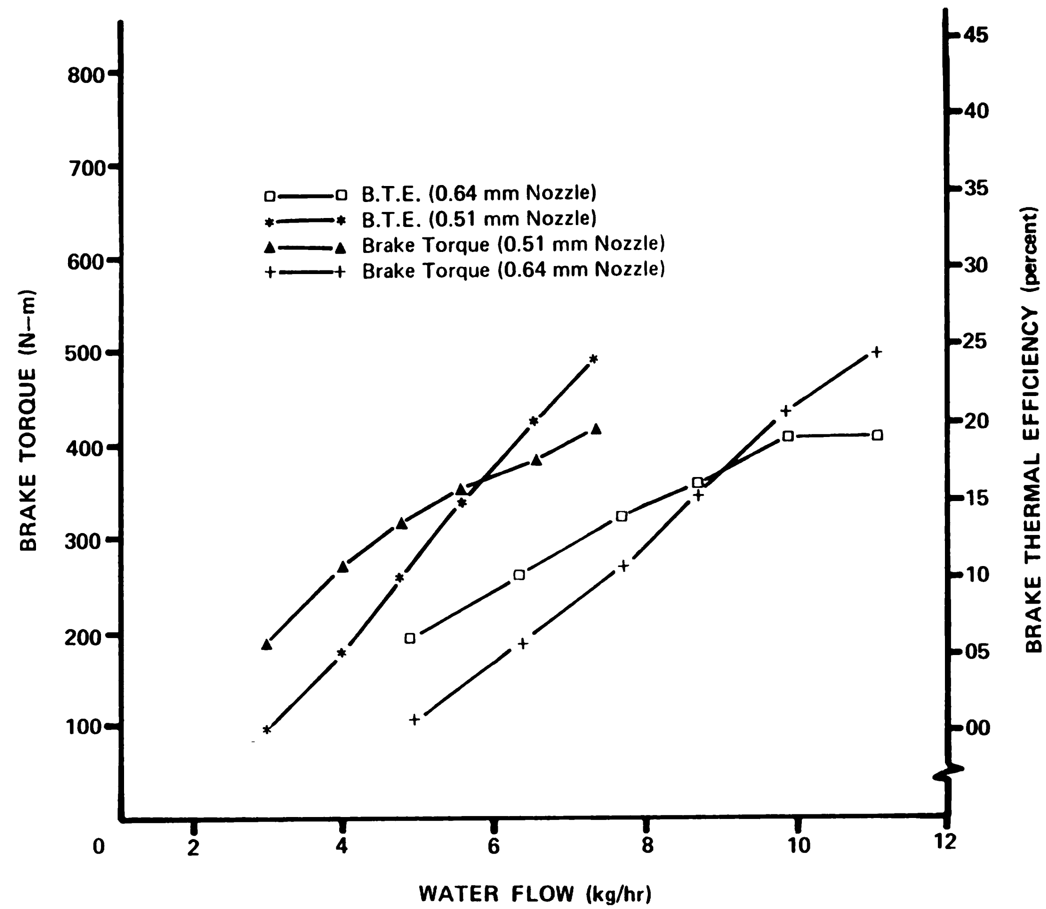


Figure 20. Effects of various proportions of water injection on engine torque and brake thermal efficiency.

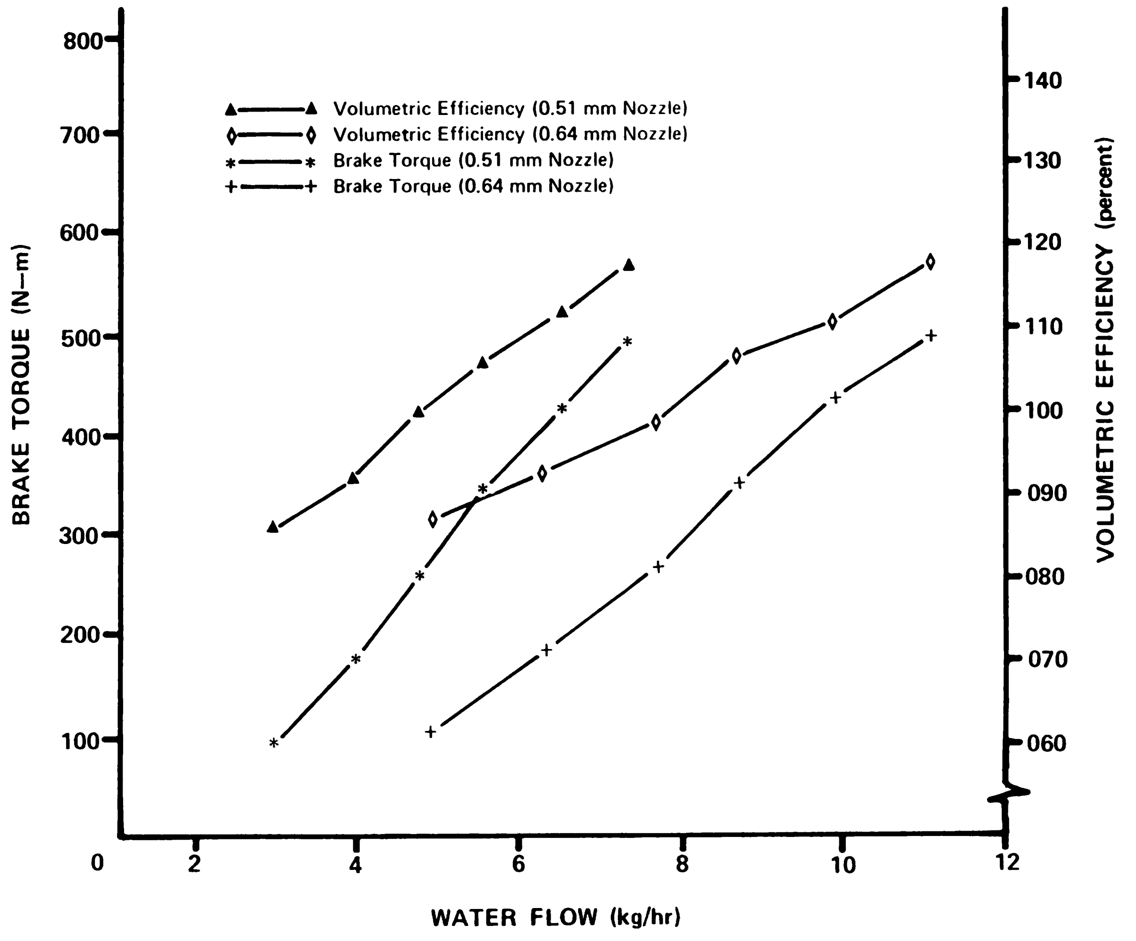


Figure 21. Effects of various proportions of water injection on engine torque and volumetric efficiency.

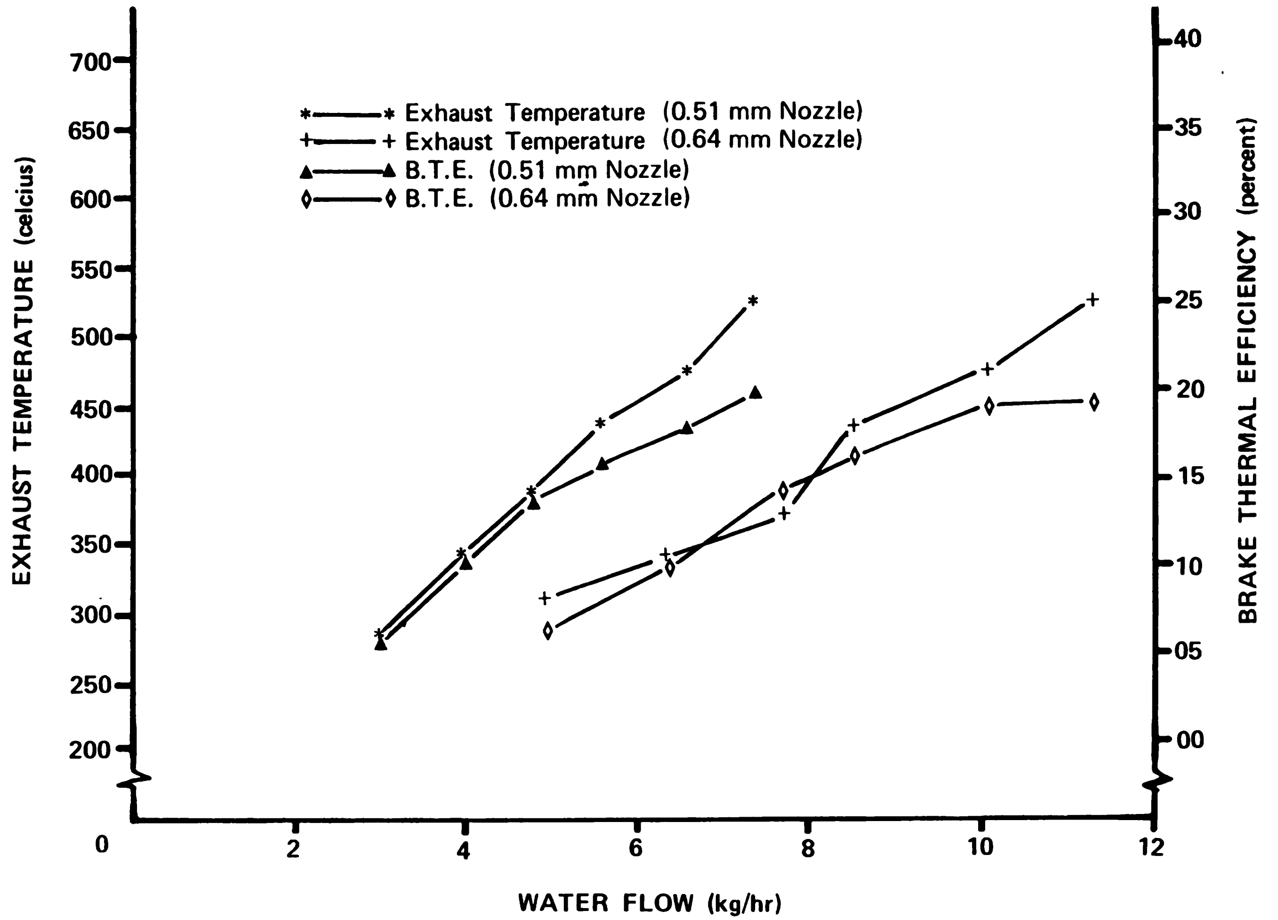


Figure 2 2. Effects of water injection on brake thermal efficiency and exhaust temperature.

impurities. As the distillation process was perfected, the problem of impurities in the alcohol was resolved.

CHAPTER 6

DUAL-FUELING THROUGH CARBURETION

The second method of dual-fueling was done by attaching a carburetor to the inlet air system of the tractor turbocharger (Figure 23).

6.1 Equipment Description

The carburetor approach consists of a carburetor, a by-pass system, choke plate/throttle valve system, control linkage, electric fuel pump, in line filter, and the ethanol fuel tank (Figure 24).

6.1.1 Carburetor

The carburetor used was "Carter YF No. 4499S" down draft type, with a 30.5 mm throat diameter and a 1.02 mm diameter main jet. This carburetor was purchased at a salvage yard at a very low price.

The sizing of the venturi for the carburetor for dual-fueling the diesel engine with carbureted alcohol, was based upon a continuity equation. From the continuity equation we have that:

$$Qair = \pi/4 \times d^2 \times Kc \times \sqrt{2g\ \Delta h_{air}} \qquad (1)$$

where:

d = orifice diameter, which becomes the diameter of the venturi

Kc = orifice coefficient. This was obtained from reference 21 page 305 for sudden expansion and D1/D2 $\simeq$ 0.0 (see Appendix B for more detail).

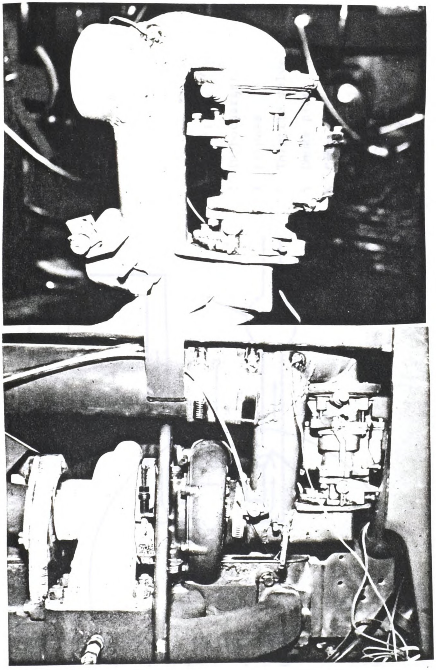
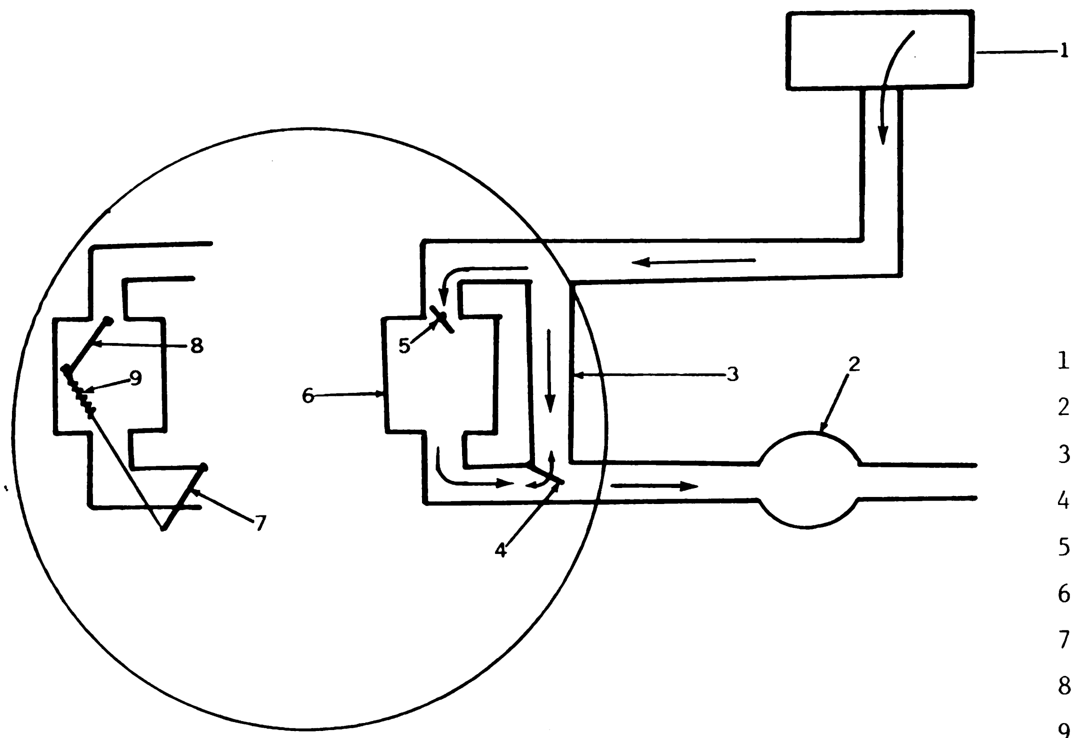


Figure 23. Intake air carburetor mounting.



1 Air filter
2 Turbocharger
3 By-pass
4 Throttle valve
5 Choke plate
6 Carburetor
7 Throttle linkage
8 Choke linkage
9 Spring apparatus

Figure 24. Carburetor/by-pass apparatus used to gradually increase ethanol flow.

g = gravitational acceleration.

Δh = the vacuum in the venturi (head of air). This data was recommended as 8.5 KPa by reference 16.

Qair = volume of intake air by the engine, and is calculated by the following formula:

$$\text{Qair} = \frac{2 \times \text{R.P.M.} \times \text{displacement}}{2 \text{ or } 4(\text{number of strokes per cycle})}$$

The sizing of the carburetor jet for dual-fueling a diesel engine with carbureted alcohol, was obtained from the following formula:

$$AF = 1.64 \left(\frac{d}{df}\right)^2 \Psi$$

where:

AF = air/alcohol ratio, obtained from results of reference 4, page 26.

Ψ = air/fuel ratio constant extracted from table in reference 16, page 390.

d = diameter of the sized venturi.

df = diameter of fuel orifice (jet).

6.1.2 By-pass Apparatus

In order to permit the engine to start and shut off on diesel alone, a by-pass system was built (Figure 25). This system was built with the inlet air divided into two parts. One part of air passed through the carburetor to draw alcohol from the carburetor jet. The other part went through the by-pass in order to assure that the engine was not lacking air. A throttle valve placed at the by-pass outlet was used to manually control the direction and amount of air flowing through the system. The by-pass apparatus allowed the engine to start carbureting the alcohol when the engine reached an optimum speed for dual-fueling operation. The system was built from pipes at the Agricultural Engineering shop (see Figure 25).

6.1.3 Fuel Pump

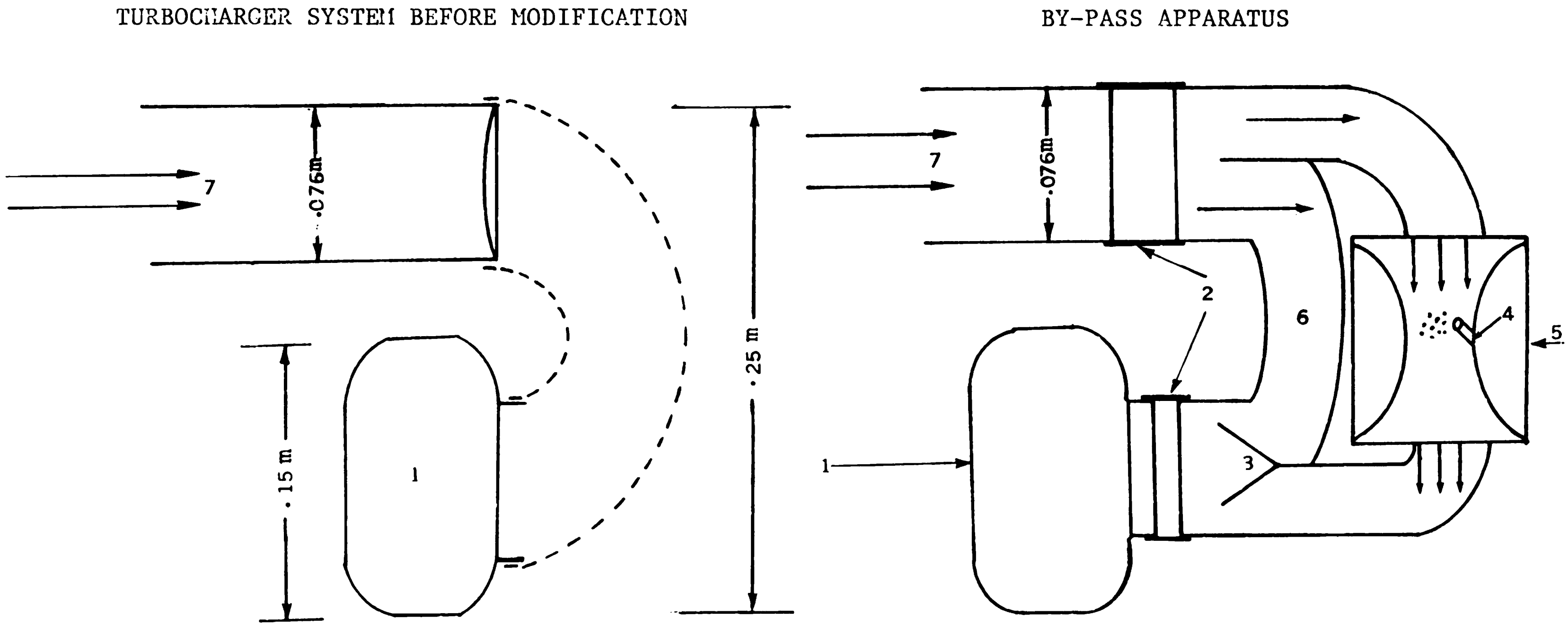
A Stuart-Warner model 235 series fuel pump electrically operated with the 12 volt tractor battery was installed to provide adequate flow of alcohol from the ethanol tank to the carburetor. The pump had a "built-in" fuel screen to catch large impurities. An additional fuel filter (Fram 3G) was incorporated in the line at the "in" port of the pump to assure clean fuel delivery to the carburetor. The fuel pump was recommended as being compatible with alcohol.

6.2 Performance

The performance of the tractor with carbureted ethanol was analyzed through the following parameters.

6.2.1 Maximum Power

Maximum power values obtained with carbureted alcohol were higher than the tractor maximum torque values obtained with diesel fuel alone, which was designated as 100 percent. The following maximum torque values are shown in Table 5.



1 Turbocharger
2 Connections
3 Throttle valve
4 Carburetor jet
5 Carburetor
6 Air by-pass
7 Engine intake air

Figure 25. Longitudinal cross-sectional view of the carburetor by-pass apparatus.

Table 5. Engine maximum torque with carbureted alcohol.

Carbureted fuel	Engine speed (PTO speed) R.P.M.		
	2100 (1000)	1600 (755)	1150 (540)
100% ethanol	136%	130%	106%
81% ethanol solution	143%	118%	128%
50% ethanol solution	159%	138%	106%

The maximum tractor power obtained when carbureting different ethanol/water mixtures is shown in Table 6. As shown at the highest speed tested, increased water in the alcohol increased the degree of diesel substitution and peak power. On the other hand, the 81% ethanol solution presented the best result throughout the different engine speeds.

Table 6. Maximum tractor power obtained when carbureting different ethanol/water mixtures.

Fuel used	Engine speed (R.P.M.)					
	2100		1600		1150	
	Power (kW)	Fuel energy (%)*	Power (kW)	Fuel energy (%)*	Power (kW)	Fuel energy (%)*
Diesel fuel only	57	--	57	--	38	--
Diesel + 100% ethanol (carbureted)	77	20	74	20	40	22
Diesel + 81% ethanol (carbureted)	81	24	68	31	48	23
Diesel + 50% ethanol (carbureted)	90	34	79	25	40	19

*Percent of total fuel energy contributed by the alcohol.

6.2.2 Thermal Efficiency

Data presented in Figure 26 indicates that when the oxygen in the cylinder begins to limit further torque, increased over-fueling (when curves start to peak) with alcohol produced higher brake thermal efficiencies than over-fueling with diesel fuel. In these tests, this characteristic was more noticeable with the 50 percent ethanol solution than with the use of other ethanol solutions. The use of the 80 percent ethanol by volume permitted an increase in brake torque without over-fueling, since a steep increase in brake thermal efficiency was noticeable.

At low torque levels brake thermal efficiencies appeared higher with diesel fuel alone than with ethanol. This reversal of relative efficiency characteristics between high-torque and low torque conditions was also noticed by Barnes, et al. (2), Panchapakesan, et al. (17), and Cruz (4). The percentages of water in the alcohol influenced the peak brake thermal efficiency along with torque levels. The 100 percent ethanol presented the highest brake thermal efficiency among the other ethanol solutions tested. The 84 percent ethanol solution showed lower results than 100 percent ethanol at lower torque levels, but was equal or greater at high torque levels.

The use of a 50 percent ethanol solution reduced the mass of oxygen inducted into the cylinder by the turbocharger on each intake stroke. At high torques, greater over-fueling was required with the 50 percent ethanol solution than with the other fuels to obtain a given torque level causing brake thermal efficiencies to be lower.

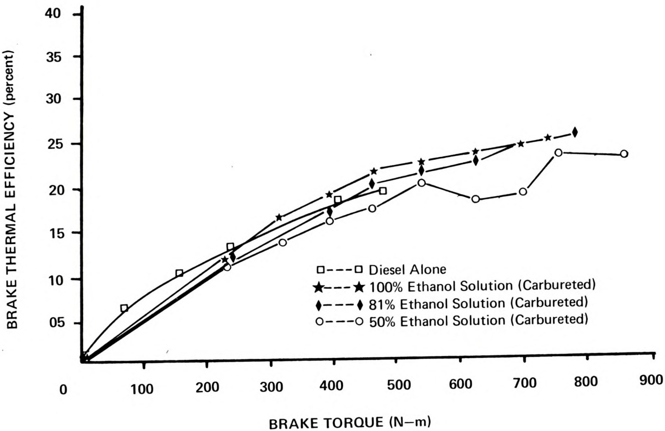


Figure 26. Effects of carbureted ethanol on brake thermal efficiency at various torque levels.

6.2.3 Volumetric Efficiency

Carbureting ethanol showed similar behavior as the spray-injection method. Lower volumetric efficiencies than diesel alone were observed for the carburetor approach (see Figure 27). The reasons are the same ones explained previously in the spray-injected method. In contrast with the spray-injection approach, no relationship was observed between water content in the ethanol and volumetric efficiency.

6.2.4 Exhaust Temperature

The low exhaust temperature with the use of the higher water concentration ethanol, in combination with lower engine torque, was a result of the typically low exhaust temperatures at low torque (see Figure 28). Exhaust temperatures at high torque levels were about 125° C lower with carbureted ethanol than with diesel fuel alone. Even though the engine developed 40 percent more brake torque with the carbureted ethanol solutions, the exhaust temperature did not go above the temperature obtained when the engine was operated on diesel alone. No relationship was observed between ethanol/water concentrations and exhaust temperature.

6.2.5 Fuel Consumption

As shown in Figure 29, the difference between the total consumption (diesel and ethanol) and the diesel consumption with carbureted ethanol is the amount of alcohol consumed. And the difference between the diesel consumption with conventional system and diesel consumption with carbureted ethanol is the amount of diesel displaced by the carburetor method.

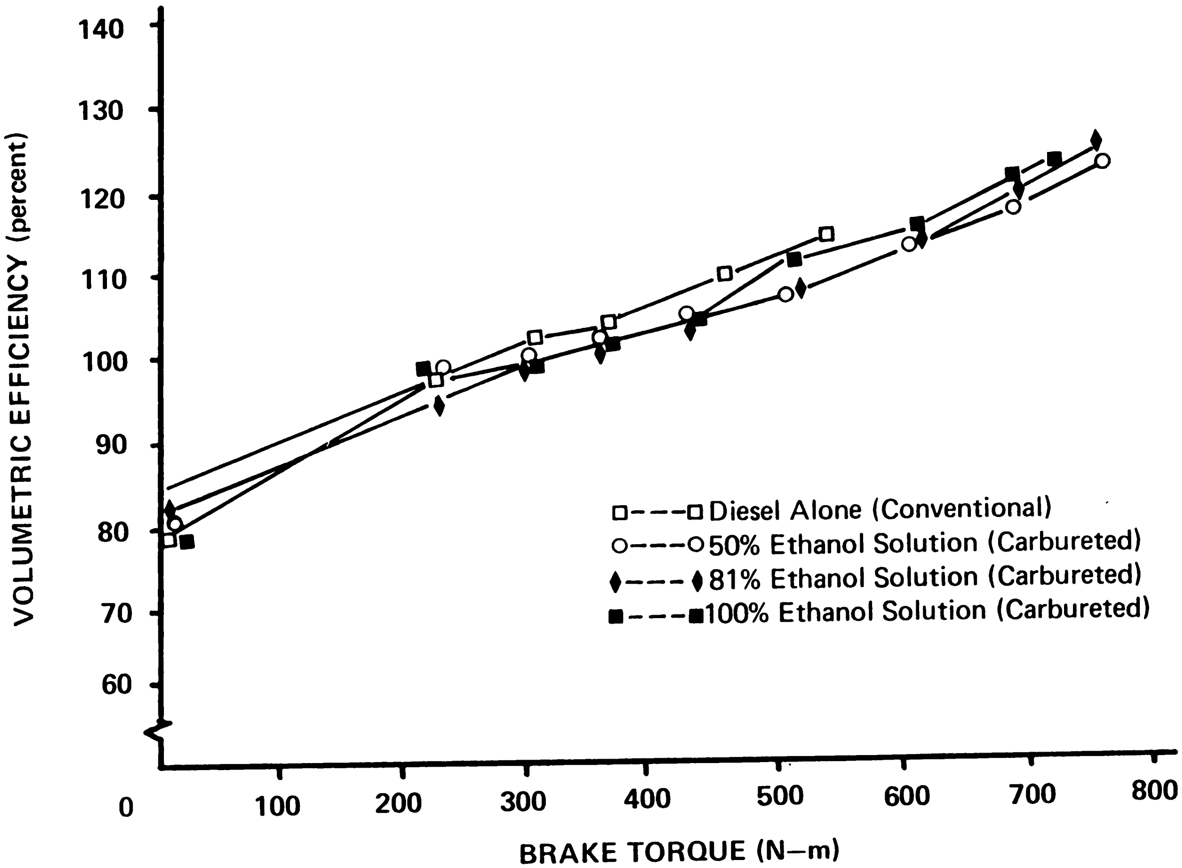


Figure 27. Effects of carbureted ethanol on volumetric efficiency at various engine torque levels and 2100 rpm engine speed.

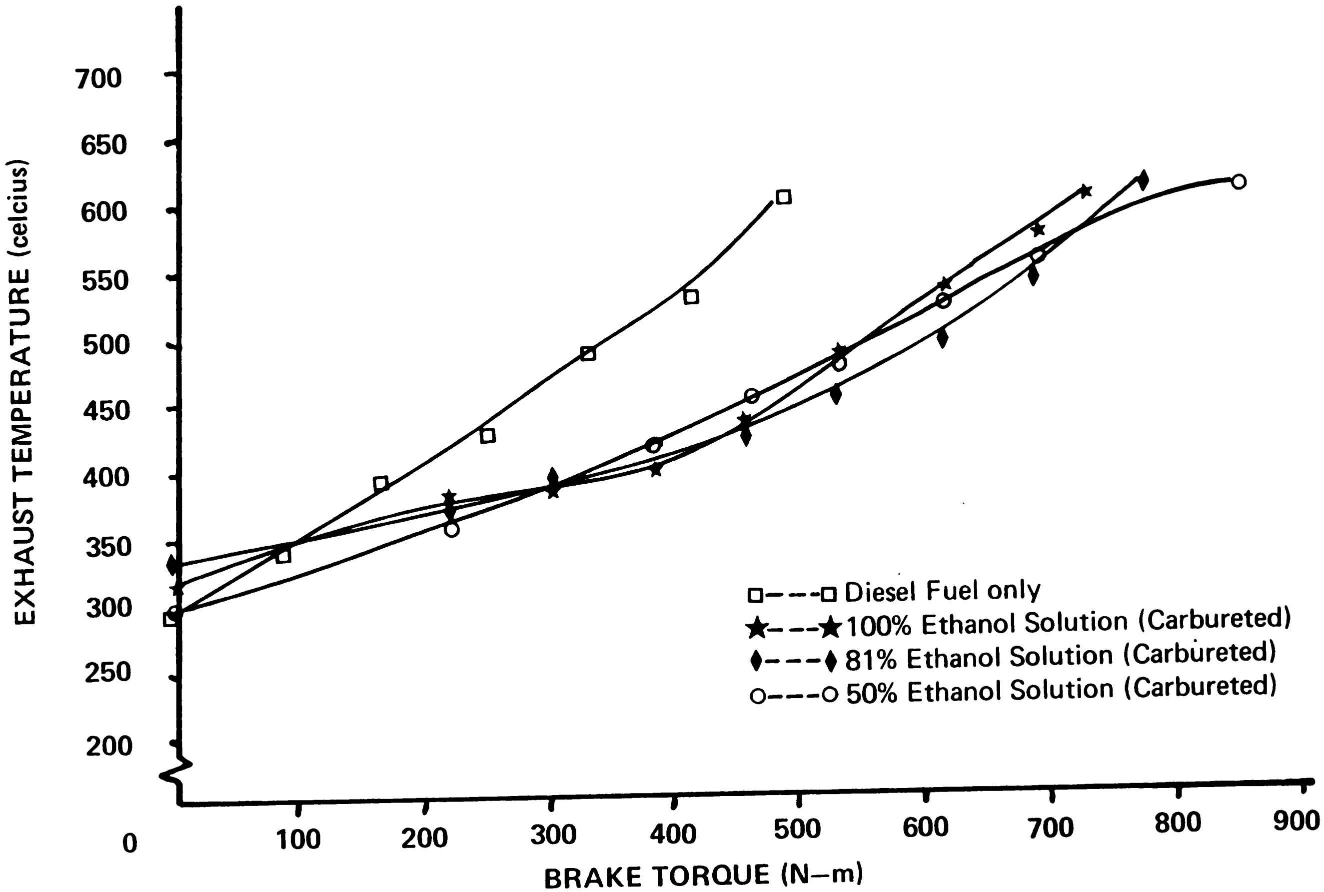


Figure 28. Exhaust temperature at various engine load for all fuel mixtures tested with carbureted ethanol.

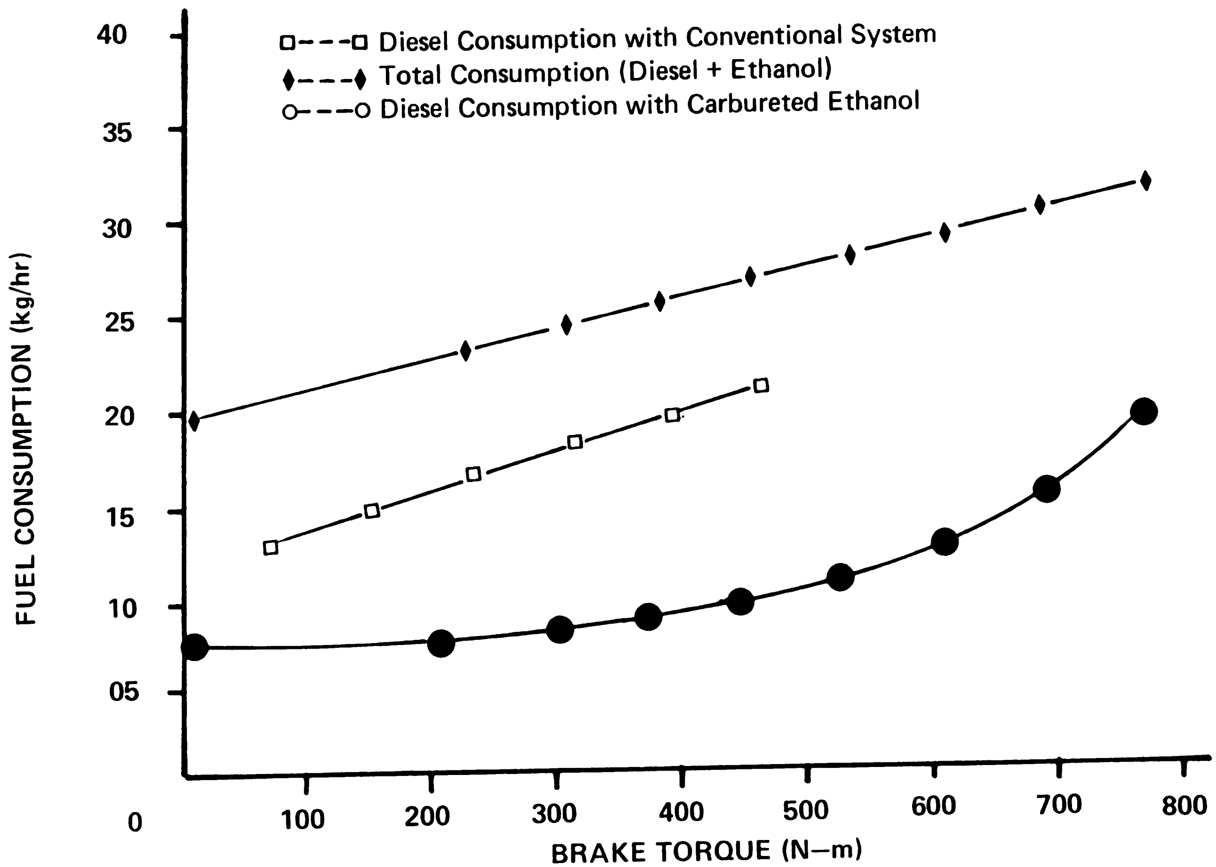


Figure 29. Fuel consumption in conventional and carbureted dual-fueled diesel tractors with 81% ethanol solution.

At low torque levels the ratio of alcohol used and diesel displaced was about two-to-one. But since alcohol has about half of the caloric value of the diesel fuel, the energy input remains the same.

As torque increased the ratio of alcohol consumed to diesel displaced was no longer two-to-one. Instead, a smaller ratio between these two parameters occurred. Therefore, more than one energy unit of diesel fuel was replaced by one energy unit of ethanol. The replacement showed best results at the peak torque for conventional fueling.

The degree of fuel substitution which can be achieved within the engine performance restrictions determined in these tests is shown in Figure 30. The behavior of the diesel replacement with different alcohol concentrations indicates that the amount of diesel fuel displaced by the ethanol varies with the torque level and percentages of water in the ethanol. At low torque levels the replacement of diesel appeared higher with the 100 percent ethanol and it decreased as the concentrations of alcohol in the alcohol/water solution decreased. A reversal of this phenomenon was observed at high torque levels. Throughout the brake torque ranges otherwise obtained with conventional fueling, the 81 percent ethanol solution presented a more constant degree of replacement (~45 percent).

The air/alcohol ratios found at various torque levels before incipient knock was heard are indicated in Table 7.

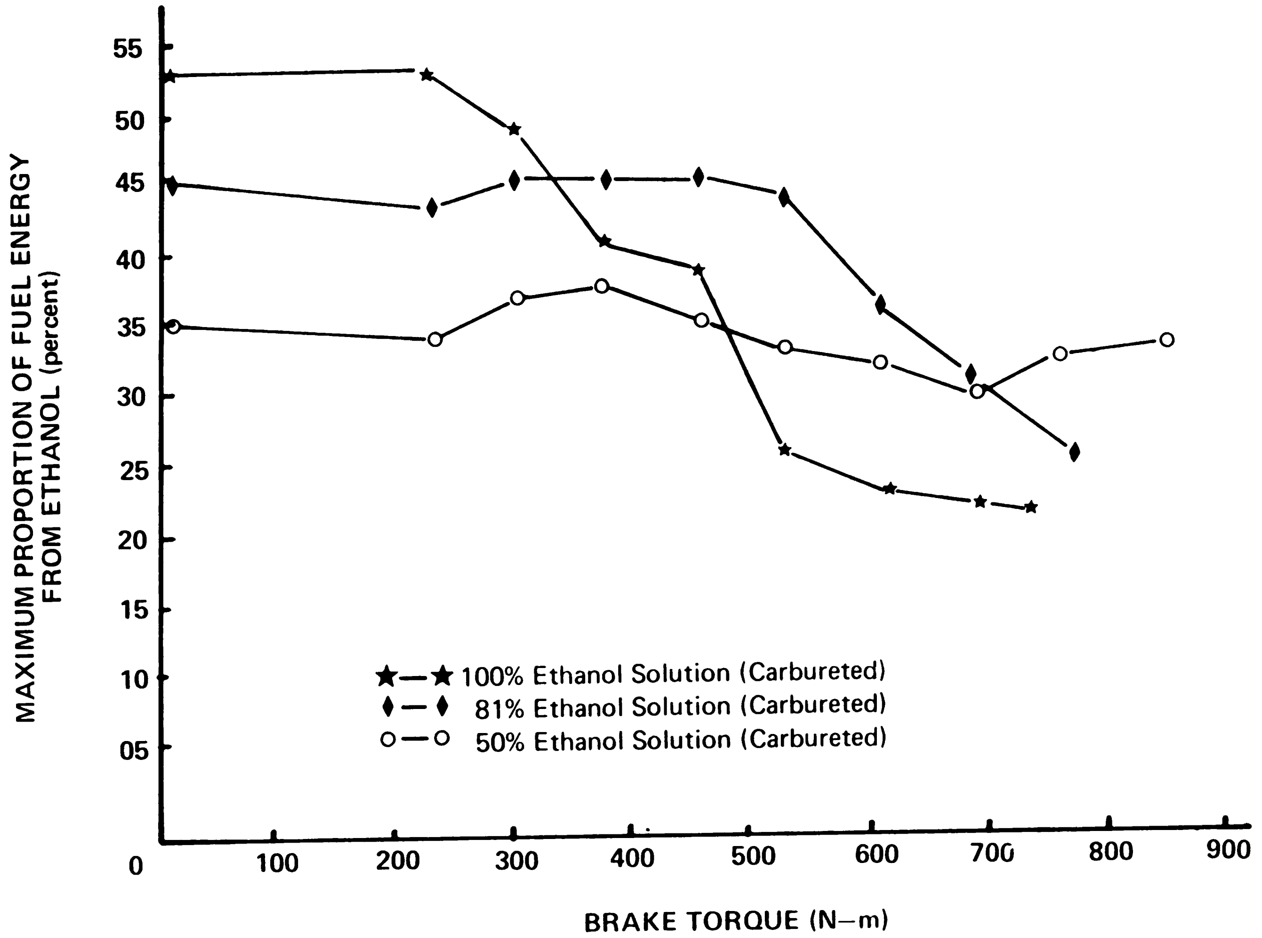


Figure 30. Maximum proportion of fuel energy displaced with carbureted ethanol for different ethanol solutions.

Table 9. Air/alcohol ratio prior to onset of knock.

Carbureted fuel	Percent of maximum torque (diesel only)					
	0*	40	60	80	100	130
100% ethanol	22[53]**	22[53]	24[49]	29[40]	44[25]	47[21]
81% ethanol solution	22[45]	22[44]	21[50]	20[46]	19[44]	26[31]
50% ethanol solution	18[35]	17[34]	15[36]	15[35]	15[33]	13[30]

* Engine at 2100 rpm and no load.
**Numbers in brackets indicate the percent of total fuel energy contributed by the ethanol. Numbers not in brackets represent the air-alcohol ratio.

These data indicate that only a limited proportion of carbureted ethanol could be used to fuel turbocharged diesel tractors, and that this proportion decreased as engine torque approached maximum levels. This proportion was restricted to air/ethanol mixtures which were sufficiently lean to avoid knock.

6.2.6 Turbocharger Pressure

Because the carburetion of alcohol changes the intake air density, and consequently the mass flow rate and pressure ratio, the turbocharger pressure also was an object of interest in this investigation (see Figure 31). The turbocharger pressure values were lower for the carburetor method of dual-fueling than the engine operating on diesel fuel alone. At low torque levels the variation of water concentrations in the ethanol solutions did not affect the turbocharger pressure. However, at high torque levels an increase of water in the ethanol solutions caused a significant decrease in turbocharger pressure.

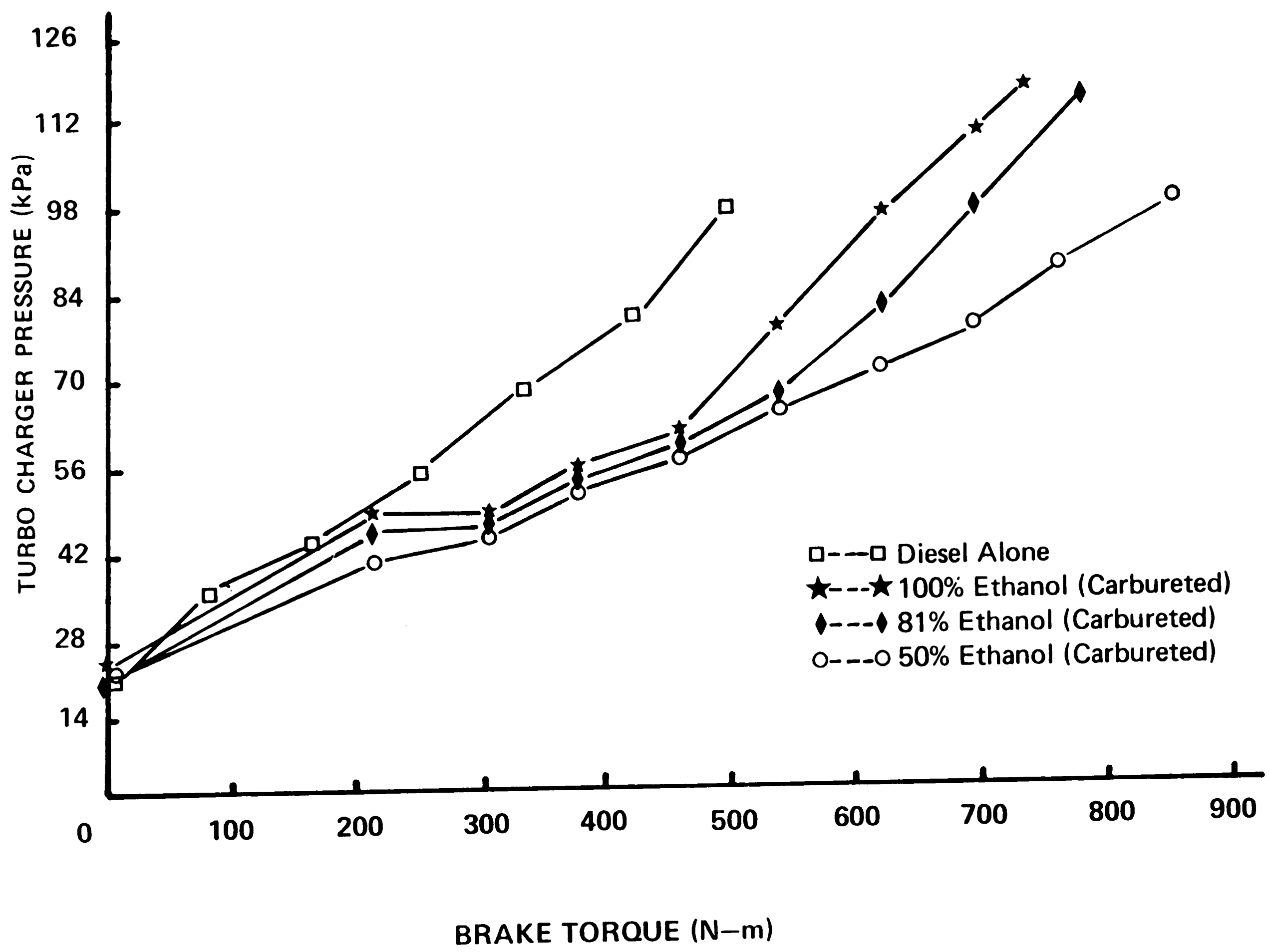


Figure 31. Effects of using different ethanol/water mixtures on turbocharger pressure when carbureting ethanol.

6.3 Effects of the Carburetor on the Engine Performance

Performance characteristics of the carburetor relative to the heat requirements for alcohol evaporation are presented in Table 8. Cooler intake air was obtained when alcohol was carbureted. At lower engine speed and load, there was insufficient heat in the air to evaporate alcohol solutions with higher concentrations of water, as demonstrated in Table 10. As load was increased the proportion of unevaporated ethanol decreased until engine inlet temperatures were approximately those of the ambient air. The amount of water in the ethanol influenced the amount of unevaporated ethanol, since the negative differences were greater for ethanol with a higher concentration of water. An exception to these findings occurred at no load for 1600, and 1150 rpm engine speed. The reason why the values did not agree on those was that higher percentages of total energy were contributed by the alcohol. At lower engine loads, an air pre-heater might be required to prevent uneven distribution of liquid particles among the cylinders. The turbocharger does more heating of the air at higher rpm or load.

These data presented in Table 8 also indicate that only a limited proportion of carbureted alcohol can be used to fuel diesel engines, and that this proportion decreased as engine torque approached maximum levels. This proportion was restricted to air/ethanol mixtures which were sufficiently lean to avoid knock.

6.4 Problems Encountered

Two problems were encountered with this approach. First of all, the engine produced exhaust backfire on sudden increases of ethanol intake. The reason was because the engine did not respond fast enough

Table 8. Air temperature difference (engine inlet-carburetor inlet) °C, at equivalent brake torque.

Fuel used	Engine speed (P.T.O. speed)					
	2100(1000)		1600(755)		1150(540)	
	No load	Full load	No load	Full load	No load	Full load
Diesel fuel only	31.3	109.2	14.2	73.3	12.4	42.4
Diesel + 100% ethanol carbureted	0.9[53]*	75 [25]	-4.3[39]	49.0[23]	-4 [26]	20.2[22]
Diesel + 81% ethanol carbureted	-1.5[45]	34.3[44]	-5.6[30]	18.5[31]	-9.9[30]	6.2[23]
Diesel + 50% ethanol carbureted	-2 [35]	17.6[33]	-2.3[21]	11.2[25]	-6.5[15]	7.9[19]

*Numbers in brackets indicate the percent of total energy contributed by the alcohol.

to completely burn all of the ethanol. Exhaust backfire in C.I. engines is caused by an unbalanced air/fuel ratio, in other words, by excess oxygen obtained from the previous stroke and unburned gases at the following stroke. These two combined were ignited by the hot carbon deposits at the exhaust pipe, causing backfire followed by a black cloud of smoke which originated from the carbon deposits released at the explosion.

When the ethanol was carbureted, a sudden opening of the throttle valve caused a larger volume of ethanol to be drawn into the cylinders, and as a result unburned fuel was released. Since the turbocharged tractor operated on a rich air/fuel ratio, some oxygen was left at the exhaust pipe from the previous stroke. Hence, exhaust backfire occurred when excess alcohol came in contact with the hot exhaust air as explained above. This problem, however, did not occur when the ethanol was throttled slowly. Based on this, the problem was solved by attaching a linkage from the throttle valve to the choke plate on the carburetor. The choke plate was installed to open and close twice as fast as the throttle valve to eliminate the sudden flow of ethanol.

Another problem encountered was that the bolts which hold the throttle valve at the throttle rod vibrated loose. They were drawn into the turbocharger which ruined the compression blades. The reason was because the intake air rushing through the opening of the throttle valve caused vibrations, and these vibrations caused the bolts to loosen. To solve this problem, the valve and choke were riveted in place.

CHAPTER 7

DISCUSSION OF RESULTS

After tests with the 7700 turbocharged diesel tractor, it became apparent that the ignition delay characteristic of the alcohol fuel was the factor responsible for the onset of the knocking phenomenon which governs diesel performance with the spray injected and carbureted ethanol. Certain factors commonly associated with reduced cylinder temperatures, such as the higher latent heat of vaporization of the ethanol and the induction of water into the cylinder, were related to an increased tendency for knock. The tests indicated that the use of richer alcohol/air mixtures prior to the onset of knock was limited to twenty-one parts of air to one part of alcohol. Furthermore, the combustion of alcohol/air mixture depended very much on the relative proportion of alcohol to the diesel oil inducted.

The ignition delay and associated knock imposed limitations on the extent of alcohol use in engines. It restricted the degree of substitution for petroleum fuel that might be accomplished by the methods of spray injection or carburetion.

7.1 Knock

When knock occurs two different types of vibration may be present. In one case a large amount of mixture may autoignite and so give rise to a very rapid increase in pressure throughout the combustion chamber which will be a direct shock on the engine structure. The ear will detect a thudding sound from the impact and consequent free vibrations of the engine parts. In the other case, a large pressure difference may exist in the combustion chamber, and the resulting gas vibrations can force the walls of the chamber to vibrate at the same frequency as the gas. An audible sound or ping is then evident (16). Any unusual sound that arose because of the dual-fueling with alcohol set the limit of the increase in flow of alcohol.

In the C.I. engine, knock becomes apparent when there are significant pressure disturbances because of autoignition. Autoignition is essential for starting combustion in the C.I. engine.

The severity of the pressure rise upon ignition will depend on the length of ignition delay (as well as on the self-ignition temperature). Injection occurs over a relatively long period. If the fuel has a long delay period, a larger amount will be injected and accumulated in the chamber during the delay period. Autoignition will tend to be uncontrollable because of the amount of high temperature mixture in the combustion chamber.

The knock which resulted from dual-fueling appears to be very complex. There are two possible explanations for this phenomenon. First of all, during the dual-fueling test, audible knock did not occur when low concentrations of alcohol were being carbureted. This was because alcohol and air were more homogeneous, and the engine was operating in a richer air/fuel mixture, making it easier for diesel to find the right proportion of air to start the ignition. As the mass of carbureted alcohol was increased, regions containing alcohol droplets or alcohol vapor alone, as well as diesel droplets or diesel vapor, existed in the combustion chamber. Air/fuel ratios were leaner, making it more difficult for the diesel to find the right proportion of air to diesel to start the ignition. In other words, the physical delay increased. When the ignition occurred, those regions of fuel previously described became mixed with the air due to turbulence, allowing the flame to propagate. The resulting rapid increase in pressure caused the sound called "knock."

The second explanation is that when alcohol was added to the inlet air, most of the alcohol ignited simultaneously as diesel ignition occurred. The cetane rating or octane rating of the alcohol may have had little or no impact on the process. The knock might have happened because of excess energy released by combustion of the alcohol in a very short period of time. For example, if a major portion of alcohol had been injected instead of carbureted, then this problem might not have occurred. In this situation, the combustion process begins as alcohol and diesel fuel are injected and this process continues at a steady rate as diesel and alcohol are being injected. As a result, a longer combustion process and a less rapid pressure rise would ensue.

Although the diesel flame continues to burn at a steady rate, the path of combustion for the diesel fuel varies with the percentage of alcohol present in the mixture. The increase of water in the alcohol aggravates the tendency for knocking. As explained before, this may be due to the high latent heat of vaporization of the mixture and the resulting longer ignition delay. Or perhaps this is better explained by the fact that, as the percentage of water in the alcohol increased, the cooling effects of the water caused the diesel to burn at the same steady rate, but along a narrower path or range (see Appendix B for sketch of the combustion process with different alcohol/water mixtures). Consequently, a larger portion of the air/alcohol mixture was available to ignite simultaneously.

As the air/diesel ratio decreased to the point where oxygen in the cylinder was insufficient for complete combustion, an earlier onset of incipient knocking was observed limiting the maximum decrease in air/alcohol ratios. Air/alcohol ratios which could be tolerated without undue knock had to be made leaner as the proportion of water in the carbureted mixture was increased. At lower air/diesel ratios, this tendency was accentuated.

With the carburetor method the engine tended to knock less than the spray-injected approach. Consequently, higher displacement of diesel fuel was possible with the alcohol. This was probably because of more uniform mixing of alcohol and air with the carburetor approach.

7.2 Performance

At part load, a small amount of fuel injected into the engine will not need all of the air in the cylinder because the C.I. engine inducts a constant amount of air on the intake stroke. As the load is increased, greater amounts of fuel are injected and more and more of the air is required for combustion. At some stage, further injection of fuel leads to part of the fuel not being oxidized and to the production of smoke. Even with this condition, part of the air in the engine may not react because of failure of the injected fuel to find air which will cause the engine torque to level off. With dual-fueling this phenomenon happened later than with conventional fueling, permitting the tractor to reach much higher torques. Table 11 presents a summary of the tractor performance both with and without dual-fueling.

The increase in power due to the use of alcohol was greater for carbureted alcohol than spray-injected alcohol. Combining the richest alcohol/air ratios that could be used without knock with the minimum diesel fuel injection, produced 59 percent more torque, when ethanol was carbureted, and about 36 percent higher torque with the spray-injection approach.

At the richest diesel/air ratio an increase in alcohol/air ratio did not cause much increase in torque, as this amounted to over-fueling with limited oxygen.

Increasing the water content in the carbureted ethanol/water mixture caused an increase in maximum power. This fact was even more noticeable for the carbureted alcohol than spray-injected alcohol.

Although alcohol caused a decrease in volumetric efficiency, it increased torque and power quite substantially. The reason was that alcohol, with a high latent heat of vaporization, acted as an intercooler when evaporated into the intake air. The reduction in temperature was beneficial to the engine performance because more dense air entering the engine allowed a greater mass flow of air. The volumetric efficiency of the engine decreased because of the lower temperature and pressure in the manifold. Intercooling reduced both compressor efficiency and the inlet pressure to the engine, but increased the density and mass flow rate which increased torque and power.

The results of the carburetor approach were plotted in Figure 32. Also, the most suitable ethanol solution and nozzle size (50 percent and

Table 9. Summary of the maximum values attained for tractor performance with spray-injection and carburetor approaches.

Engine speed	Fuel used	Air/alcohol ratio before incipient knock			
		Max. prop. of fuel energy from ethanol (%)	Maximum torque (n.m.)	Max. brake thermal effic. (%)	Maximum volumet. effic.
2100	Diesel fuel only	--	539	21.2	115.3
	Diesel + 100% ethanol (carbureted)	53.2	731.5	24.5	123.8
	Diesel + 100% ethanol (spray injected)	19	651	23	124.0
	Diesel + 81% ethanol (carbureted)	45.9	770.0	24.6	126.1
	Diesel + 84% ethanol (spray injected)	46.0	569	22	115.0
	Diesel + 50% ethanol (carbureted)	36.8	855.0	24.3	122.6
	Diesel + 50% ethanol (spray injected)	29	732	23	137.0
1600	Diesel fuel only	--	724.1	26.0	79.7
	Diesel + 100% ethanol (carbureted)	38.6	938.3	29.0	88.3
	Diesel + 81% ethanol (carbureted)	32.5	856.7	27.8	81.5
	Diesel + 50% ethanol (carbureted)	26.3	1001.0	27.8	93.1

Table 9 (cont'd)

Engine speed	Fuel used	Air/alcohol ratio before incipient knock			
		Max. prop. of fuel energy from ethanol (%)	Maximum torque (n.m.)	Max. brake thermal effic. (%)	Maximum volumet. effic.
1150	Diesel fuel only	--	670.2	21.2	51.3
	Diesel + 100% ethanol (carbureted)	26.3	713.0	23.7	51.0
	Diesel + 81% ethanol (carbureted)	29.8	855.6	24.3	56.4
	Diesel + 50% ethanol (carbureted)	20.3	706.9	21.5	51.0

Note:

1. Spray injected refers to the M & W Gear system with .89 mm nozzle size, except 100% ethanol which used .51 mm nozzle size.

2. The alcohol % refers to % by volume.

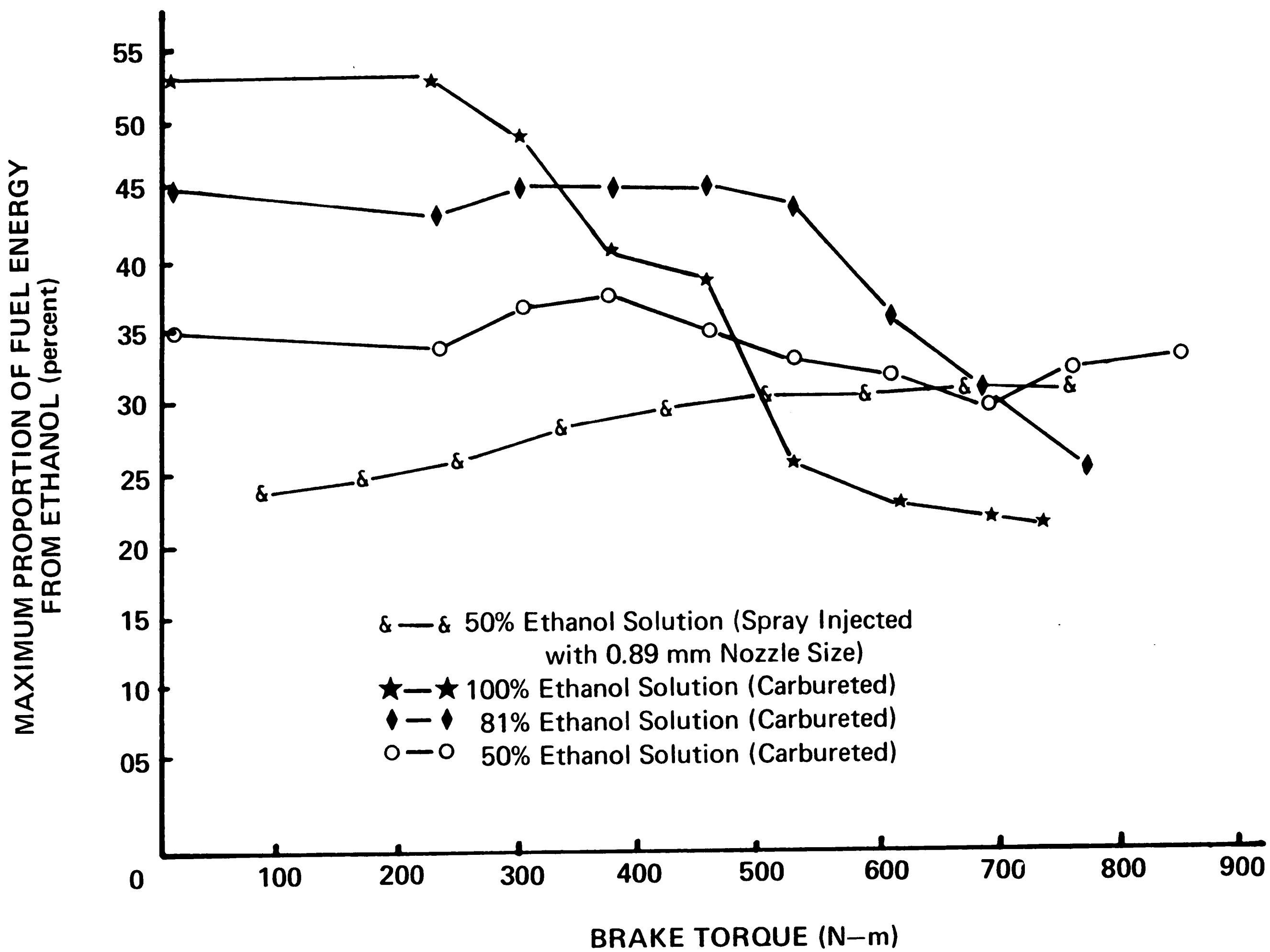


Figure 3 2. Maximum proportion of fuel energy displaced through carburetion and spray injection of the ethanol.

.89 mm) for the spray-injection approach was plotted and used for comparison. Except for the 100 percent ethanol dual-fueled by the carburetor approach at higher torque ranges, dual-fueling with carbureted ethanol solutions showed superior results in replacing diesel fuel.

The reason why the spray-injected ethanol showed lower maximum proportion of replaceable energy at low torque levels was because the rate of injection varied with turbocharger pressure. Turbocharger pressure increased as torque increased. The carbureted ethanol showed the opposite behavior because the throttle valve settings, which controlled the flow of alcohol, were a function of incipient knock rather than turbocharger pressure. At high torque levels the onset of knock with carbureted ethanol occurred earlier because of a rapid pressure rise resulting from the combustion of a greater volume of injected diesel fuel.

A higher percentage of maximum fuel energy from the carbureted waterless ethanol was possible at low torque levels. When lower amounts of diesel were injected at low torque levels, the combustion rate was not large enough to produce a rapid pressure rise. However, as torque increased the opposite occurred. As the injection rate of diesel increased, the rapidity of pressure rise due to the diesel combustion also increased. Consequently, a drop in the maximum proportion of fuel energy occurred.

As the amount of water increased in the mixture of carbureted ethanol, incipient knock happened earlier as explained before. Consequently, a smaller proportion of fuel energy was replaced by the alcohol. However, as explained earlier in this chapter, the combustion of the diesel was less violent as water in the alcohol mixture increased. As a result there was a less rapid pressure rise and a similar proportion of alcohol could be carbureted as torque increased

The maximum substitution of alcohol for diesel fuel via injection and carburetion are shown in Figures 33 and 34. As these Figures show, the carburetor used the ethanol more efficiently than the spray-injected system. The portion of diesel displaced by the carburetor method was larger. Furthermore, the total consumption (diesel and ethanol) is smaller at high torque levels with the carburetor. At low torque levels with carbureted ethanol, there is nearly a direct trade-off of fuels in terms of energy content, but as torque increases, the engine uses the alcohol more efficiently. In other words, one energy unit of alcohol displaces more than one energy unit of diesel. In addition, with this system, maximum torque is increased.

7.3 Efficiency

For all ethanol concentrations used over the high range of engine torques, higher efficiency levels were found in engine operation with carbureted ethanol than in operation with the spray-injected method. One exception was found when the 50 percent ethanol solution was spray-injected. In this case, a higher efficiency level occurred at low torque levels and an equivalent efficiency level occurred at high torque levels when compared with operation using carbureted ethanol.

The carburetion of the 50 percent ethanol solution reduced the mass of oxygen that could be inducted at higher torque levels. Greater over-fueling was then necessary to achieve a given torque level under maximum torque conditions, which caused a drop in efficiency.

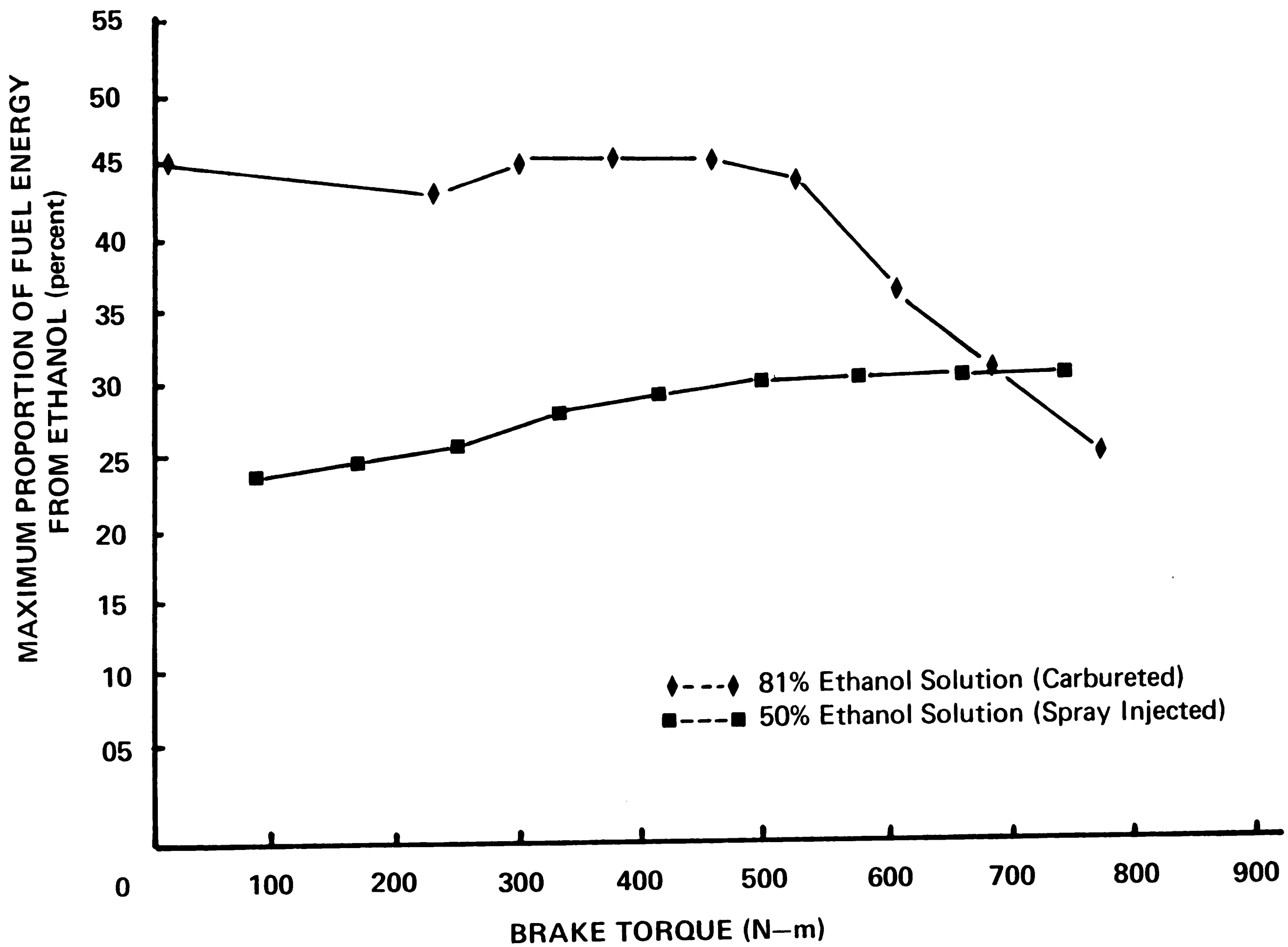


Figure 33. Maximum proportion of fuel energy displaced with carbureted ethanol and spray injected ethanol for selected ethanol solutions.

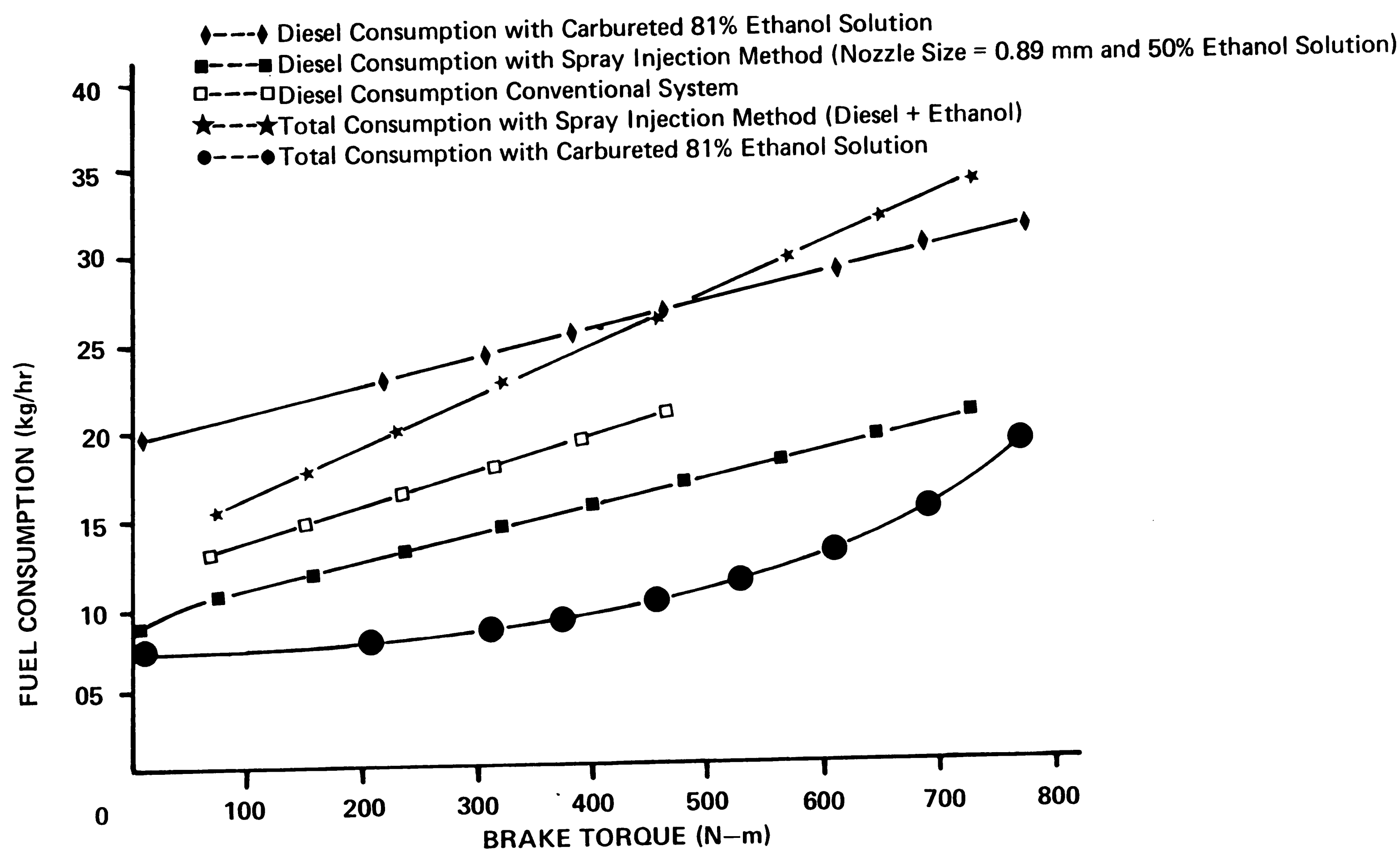


Figure 34. Fuel consumption in conventional, spray-injected, and carbureted dual-fueled diesel tractor with diesel, 50% ethanol, and 81% ethanol respectively.

The spray-injected approach presented an increase of brake thermal efficiency across the full range of torque when the 50 percent alcohol concentration was used. At higher air/alcohol ratios, which produced low engine torque levels, brake thermal efficiency with the carburetor approach appeared lower than that for operation with diesel fuel alone. The probable cause was the incomplete combustion of alcohol resulting from low cylinder temperature. However, at lower air/alcohol ratios the opposite occurred. Higher brake thermal efficiency was obtained with carbureted alcohol.

The lower air/alcohol ratios in combination with the higher air/diesel ratios caused higher efficiency than the opposite combination when different alcohol concentrations were carbureted. The addition of water in the alcohol decreased efficiency, since problems with ignition delay were accentuated. This was even more noticeable for carbureted 50 percent ethanol than with the spray-injection system.

7.4 Turbocharger Pressure

A parallel between an increase of water content in the alcohol and turbocharger pressure drop was observed when compared to conventional fueling. This phenomenon can be explained by either of the following: 1) the alcohol displaces the air that would be drawn otherwise; 2) fanning loss due to denser flow of air/alcohol mixtures; and 3) intercooling reduces both compressor efficiency and the inlet pressure to the engine.

When the engine speed is increased, compressor operation is relatively inefficient while the boost pressure strongly increases. This is undesirable for most engines, since maximum pressures on combustion are also raised (16). Therefore, alcohol improves compressor efficiency, because it lowers the outlet pressure of the turbocharger particularly at high speed and load.

7.5 Exhaust Temperature

Exhaust temperature with carbureted alcohol and spray-injected alcohol was lower than that with diesel fuel alone by approximately 90° C at high torque. This was because of the lower burning temperature of the alcohol. Even though the exhaust temperature was low with spray-injected alcohol, it gradually decreased further as the amount of water increased in the carbureted alcohol. This occurrence was not noticeable for the carbureted alcohol, because the portion of the energy contributed by the alcohol was different for each alcohol concentration at equivalent torque levels.

7.6 Control Linkage and Maximum Proportion of Energy Replaced

Based upon results obtained with the manually operated carburetor, a control linkage was built to replace the maximum allowable amount of alcohol.

The tractor specifications stated that the tractor was capable of a maximum power of 65 kW at full load. Farm tractors, however, usually work between 70 and 80 percent of the engine maximum power. At 65 kW the maximum energy displaced by the ethanol was 37 percent because of earlier onset of knock. At a lower power output of 54 kW or less, up to 47 percent of the diesel energy could be replaced by the carbureted ethanol. If the maximum allowable power it to be obtained, the linkage control can be set for a maximum of 37 percent replacement. However, the linkage system can be adjusted to replace 47 percent of the energy by a simple adjustment of the linkage. This could normally be used on a farm tractor since maximum power is seldom used.

This practical implementation of replacing the maximum allowable amount of alcohol with the carbureted ethanol was done by using a throttle valve at the carburetor and by-pass outlet. This valve varied the proportion of the inlet air drawn through a fixed jet carburetor, and it was connected to the carburetor choke plate through a linkage. This valve/choke system functioned such that the throttle valve remained partially open at no load. The choke plate opened gradually, together with the throttle valve at the rate of 2 to 1. Once the choke plate was fully opened, and the system required more alcohol fuel to meet the load, a spring system allowed the throttle valve to continue to open until the demand was met. If the system called for a reversal, the throttle valve closed gradually until it reached the point where the choke plate was fully opened. Then the choke plate started to close at a faster rate together with the throttle valve, until it was completely closed. The engine continued idling at 1000 rpm on diesel fuel alone. Thus, any stalling due to a sudden burst of alcohol was avoided. A coordinating linkage was connected from the throttle valve at the carburetor by-pass apparatus to the diesel pedal at the tractor's cabin. The coordinating linkage functioned such that:

1. The choke plate opened and closed twice as fast as the throttle valve.

2. A minimum diesel injection level sufficient for pilot ignition was maintained at all times.
3. The valve started supplying alcohol when the engine reached 1000 rpm (determined by observation of engine sound and exhaust smoke during the test), and it varied the amount of alcohol supplied to meet all torque levels up to that achieved by the minimum pilot diesel injection in conjunction with air/ethanol ratio just short of that which produced knock.
4. Torque levels beyond that described in (3), would be met by injecting increased amounts of diesel fuel beyond that required for pilot ignition.

The linkage system allowed the engine to start with diesel alone. Then when the engine was run at 1000 rpm, the valve started to increase the volume of ethanol in accordance with the torque required until it reached the maximum amount of alcohol before incipient knock.

If more torque was required and the maximum allowable air/ethanol ratio was not sufficient to supply it, the system will increase the energy input from injected diesel fuel. When a torque decrease was called for reversing the system, the linkages maintained the maximum allowable ethanol and the energy supplied by the diesel first decreased (by means of the diesel governor) until it reached the minimum diesel required. Then the energy supplied by the ethanol decreased until the desired output torque level was reached.

The throttle valve controlled the half of the intake air which entered through a fixed jet carburetor, and the other half was supplied through a by-pass. At full throttle, the by-pass was fully blocked forcing the air to go through the carburetor to fuel the maximum allowable ethanol. Under these circumstances, the diesel injection rate remained at the minimum allowable value, and the engine maximum capacity to induct air was not reduced.

CHAPTER 8

SUMMARY AND CONCLUSIONS

Tests with the converted turbocharged diesel tractor to dual-fueling by means of the spray-injection and carburetor methods led to the following conclusions:

1. Partial substitution of diesel fuel can be achieved by these approaches. The replacement of diesel with ethanol is limited to 30 percent for the spray-injection method. With direct control linkage, the carburetor approach can replace about 37 percent of the diesel with alcohol if the engine is set for 65 kW maximum power or 47 percent if the engine is set for 87 percent of the tractor maximum power.
2. An increase of water in the ethanol of up to 50 percent resulted in a decrease of the replaceable diesel fuel.
3. The spray-injection kit developed the best results for power, brake thermal efficiency and replaceable energy with the 50 percent ethanol in combination with the 0.89 mm nozzle size. This combination resulted in an increase of about 36 percent in the engine maximum power.

4. For the spray-injection method, the best nozzle sizes for use on the Ford 7700 at the respective ethanol concentrations of 50 percent by volume, 84 percent by volume, 100 percent by volume; are 0.89 mm, 0.64 mm, 0.51 mm respectively.
5. The carburetor approach showed the best performance with the 81 percent ethanol solution. And this combination developed 43 percent more power than the engine maximum power with conventional fueling. With the 50 percent ethanol solution, the carburetor approach resulted in an increase of about 59 percent in the engine maximum power.
6. The rate of torque increase with an increase in ethanol/air ratio, was greater for ethanol with larger concentrations of water. This occurred because the engine had a tendency to knock less with lower ethanol concentrations.
7. For both methods, the brake thermal efficiency of the tractor was slightly greater with ethanol and an increase of the portion of water in the ethanol caused a reduction of the thermal efficiency.
8. With the spray-injection method, except for the 50 percent ethanol solution, the volumetric efficiency was decreased by the use of ethanol solutions. Furthermore, the addition of water in the ethanol resulted in an increase of volumetric efficiency.

With the carburetor method, the volumetric efficiency was also decreased by the use of ethanol solutions.

9. The amount of diesel fuel replaced increased with an increase of water content in the spray-injected ethanol solutions tested. Among all ethanol concentrations and methods, the carburetion of the 81 percent ethanol solution presented the highest degree of replacement.
10. Spray-injected water did not increase power and had little effect on engine performance.

APPENDIX A

APPENDIX A

GENERAL ENGINE SPECIFICATIONS

General Dimensions*	Ford 7700
Height to top of Ford cab	2880 mm
Height to top of exhaust	3070 mm
Ground clearance under front axle	584 mm
Ground clearance under rear axle	679 mm
Width at minimum track	1880 mm
Overall length (to end of lower links)	3815 mm
Wheelbase (long)	2580 mm
Wheelbase (short)	2172 mm
Turning diameter (long wheelbase) without brakes	4290 mm
Turning diameter (long wheelbase) with brakes	3680 mm
Weight (with Ford cab)	
Total with fuel, oil, and water	3710 kg
On front axle (long wheelbase)	1180 kg
On front axle (short wheelbase)	1250 kg
On rear axle (long wheelbase)	2220 kg
On rear axle (short wheelbase)	2460 kg
Weight (without Ford cab	
Total with fuel, oil, and water	3220 kg
On front axle (long wheelbase)	1130 kg

*Dimensions measured with 7.50 x 16 front tires and 15.5 x 38 rear tires.

Weight (without Ford cab) cont'd	Ford 7700
On front axle (short wheelbase)	1190 kg
On rear axle (long wheelbase)	2090 kg
On rear axle (short wheelbase)	2030 kg

Engine

No. of cylinders	4
Bore	112 mm
Stroke	107 mm
Displacement	4195 cm^3
Compression ratio	15.6:1
Firing order	1-3-4-2
Idle speed	600-700 rev/min
Maximum no-load speed	2325-2375 rev/min
Rated speed	2100 rev/min
Tappet clearance intake	.355-.406 mm
Exhaust	.432-.482 mm

Cooling System

Type	Pressurized recirculating by-pass
Thermostat	
Starts to open at:	75.6° C
Fully open at:	89° C
Pressure cap	.5 (.9 with air conditioners) (bar)

Clutch

Type	Single dry plate
Pedal free travel	31-38 mm

Power Take Off	Ford 7700
Type	Independent, hydraulically actuated
Engine speed for 540 rpm PTO speed	1900 rpm
Engine speed for 1000 rpm PTO speed	2060 rpm
Hydraulic System	
Type	Live with position control, draft sensing and category II - 3-point linkage
Draft sensing systems	
Dual sensing upper link	S
Load monitor	S
Nominal system pressure	172.4 bar
Flow at rated engine speed	
@ 2100 PSI (147.6 bar) @ remote outlet	36.7 liters
Steering	
Type	Hydrostatic with tilt steering wheel
Front sheel toe-in	6-13 mm
Brakes	
Type	Mechanically actuated disc
Pedal free travel	25-32 mm
Electrical Equipment	
Alternator	12 volt, negative ground
32 amp	--
51 amp	Std.
Starter	Positive engagement, solenoid operated
Battery	128 amp./hour

Lubricants	Ford 7700
Transmission/rear axle	Ford M2C53A
Power steering reservoir	Ford M2C41A
Front wheel bearings	Ford M1C137A
Lubrication fittings	Ford M1C137B
Engine oil	Ford M2C121A (300)

Viscosity Grade and API Classification

Temperature:	Diesel:
Below 32° F (0° C)	Ford 300 or SAE 10W (CD) - low ash

APPENDIX B

APPENDIX B

CALCULATIONS

1. Torque (N, m)

   Dyno reading (lbs) x 2.0336 "or" $T = \frac{kW \times 9,549.2}{R.P.M.}$

2. Power (kW)

   $$kW\ corrected = \frac{2\pi nT}{60,000} \times \frac{P_2}{P_1} \left(\frac{t_1}{t_2}\right)^{0.5}$$

3. Specific gravity

   * Specific gravity, ethanol = 0.785 grams/cc at 25.6°C

   * Specific gravity, diesel = 0.853 grams/cc at 25.6°C

   Liquid flow

   Ethanol $\frac{ml}{sec} \times 3.22$ = kg/hr alcohol

   Diesel $\frac{ml}{sec} \times 3.071$ = kg/hr of diesel

   Distilled water $\frac{ml}{sec} \times 3.6$ = kg/hr of distilled water

4. Proportion of energy from alcohol

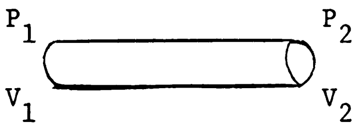
   $$\frac{\text{HHV alcohol x kg/hr x \% of alcohol in the mixture}}{\text{HHV diesel x kg/hr + HHV alcohol x kg/hr x \% of alcohol in the mixture}}$$

5. Brake thermal efficiency

   $$B.T.E. = \frac{\text{kW x 3600}}{\text{HHV diesel x kg/hr + HHV alcohol x kg/hr x \% of alcohol in the mixture}}$$

NOTE: HHV from Table 1 and reference 3.

6. Air flow



$$\frac{P_1}{\partial air} + Z_1 + \frac{V_1^2}{2g} = \frac{P_2}{\partial air} + Z_2 + \frac{V_2^2}{2g} + K\frac{V_2^2}{2g}$$

$V_1A = V_2A$ — $V_1 = V_2$

$V_1 = \frac{Q}{A}$ $\qquad Z_1 = Z_2$

K = frictional loss coefficient

$$\frac{P_1 - P_2}{\partial air} = \frac{Q^2}{A_2 \cdot 2_g}$$

$$Q = AK\sqrt{2g\,\frac{\Delta P}{\partial air}}$$

$$Q = AK\sqrt{2g \times \Delta h}$$

$Q(kg/hr) = \frac{\pi}{4} \times (\text{orifice diameter})^2 \times \text{orifice coefficient} \times \partial air \times$

$\sqrt{2 \times g \times \Delta head\ (air)}$

Orifice coefficient = 0.62 from reference 21 pp 531

$$\Delta h(air) = \Delta h(water) \times \frac{\partial(water)}{\partial(air)}$$

$$\partial(air) = \frac{P \cdot g \cdot M}{Rt}\ N/m^3$$

P = Pabs = Patm(std) + Pgage

$$Tabs = tstd + thermometer$$

$$R = 8311.4\ N \cdot m/kg\ mole\ °K$$

7. Air fuel ratio (AFR)

$$AFR = \frac{Kg/hr\ air}{Kg/hr\ fuel}$$

8. Volumetric efficiency ($\dot{\eta}$)

$$\dot{\eta} = \frac{\text{Actual air capacity (Kg/hr)}}{\text{Theoretical air capacity (TAC)}}$$

$$TAC = \frac{rev.}{min.} \times \frac{min.}{hr} \times \text{No. cylinders} \times \frac{1}{2}\ \frac{\text{intake stroke}}{\text{rev. cylinder}} \times \frac{m^3}{\text{intake stroke}}$$

$$\times \frac{Kg}{m^3}$$

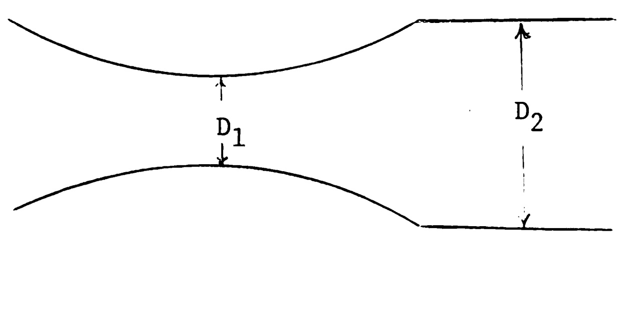
$$\text{Displacement} = \text{No. cylinders} \times \frac{m^3}{\text{intake stroke}}$$

$$\therefore\ TAC = r.p.m. \times 60 \times \text{engine displacement} \times \frac{1}{2} \times \text{density}$$

9. Sizing the venturi for the carburetor for dual-fueling diesel engine with carbureted alcohol

$$Q_{air} = \pi/4 \times d^2 \times K_e \times \sqrt{2g\ \Delta h(air)}$$

K_e = frictional loss coefficient



Sudden expansion

D_1/D_2	Ke
0.0	1.0
0.1	0.98
0.2	0.94
0.4	0.71
0.6	0.41
0.7	0.22
0.8	0.13
0.9	0.04

From reference 21 pp 305.

Assume $D_1/D_2 \quad 0.0 \quad \simeq \quad Ke = 1.0$

Δh = 2 - 3 in Hg (from personal conversation with Ford)

Take Hg = 2.5 in Hg

$$\frac{2.5 \text{ in Hg}}{7.355 \times 10^{-2} \frac{\text{in Hg}}{\text{in HeO}}} = 34.0 \text{ in } H_2O$$

$$\Delta h(\text{air}) = \frac{34 \text{ in } H_2O \times \frac{62.4 \text{ lb}}{\text{ft}^3 (\text{water})}}{0.0752 \frac{\text{lb}}{\text{ft}^3 (\text{air})}} = 28{,}213 \text{ in air}$$

Q_{air} = Volume of intaked air by the engine

$$Q_{air} = \frac{2(\text{cylinders}) \times \text{RPM} \times \text{displacement}}{4 \text{ (strokes)}} \ (\text{in}^3/\text{min})$$

Ford 7700 displacement 256 in^3

RPM = 2060 engine or 1000 PTO

$$Q_{air} = \frac{2060 \times 256}{2 \times 60} = 4,394.67 \text{ in}^3/\text{sec}$$

Since $Q = \pi/4\ d^2 * K_e \times \sqrt{2g\Delta air}$

$$4,394.67 = \frac{\pi}{4} d^2 \times 1.0 \sqrt{2 \times 32 \cdot 2 \times 12 \times 28,213}$$

$$d^2 = 1.1983$$

d = 1.20 in or 30.5 mm diameter of the venturi.

10. Sizing the jet for the carburetor for dual-fueling diesel engine with carbureted alcohol (equation from reference 16)

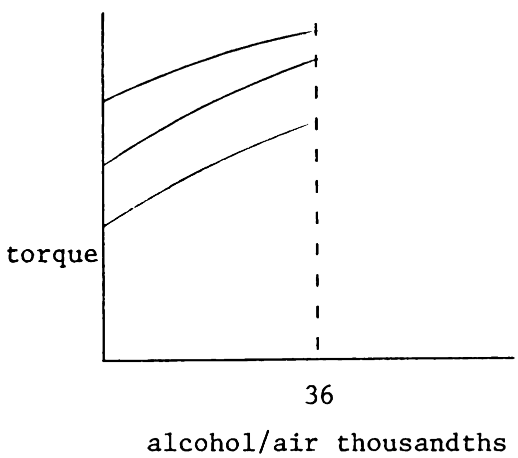
$$AF = 1.64 \left(\frac{d}{df}\right)^2 \Psi$$

Ψ = from table in reference 16 pp 390.

d = diameter of venturi

df = diameter of fuel orifice (jet)

AF = from results of reference 4 pp 26



alcohol/air ratio thousandths = 36

inverse = 27 air/alcohol
(this air/alcohol ratio is the maximum allowed alcohol combined with the minimum diesel)

For ΔP = 34 in of water, it was from table 11-1 reference 16 that

= 0.0250.

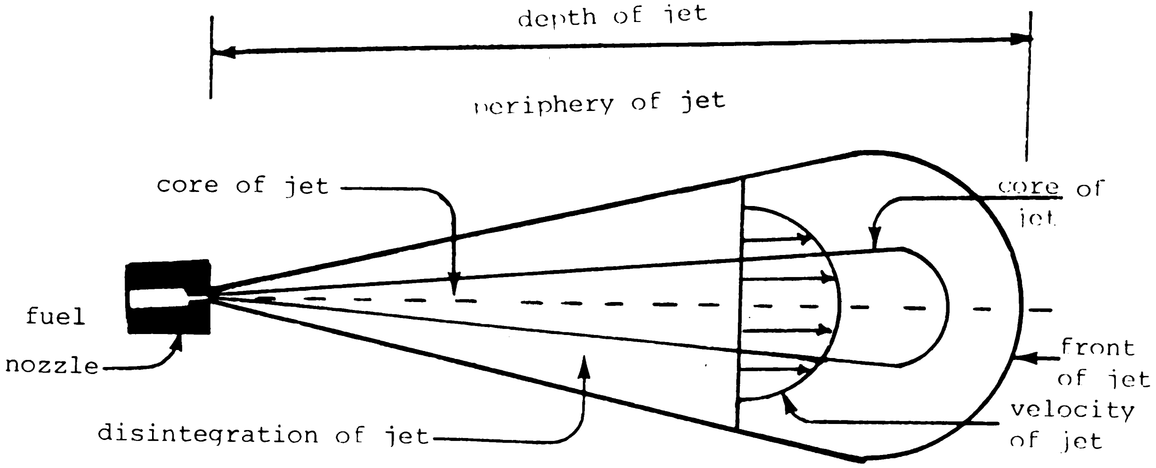
$$AF = 1.64(\frac{d}{df})^2 \ \Psi$$

$$37 = 1.64(\frac{1.2}{df})^2 \times 0.0250$$

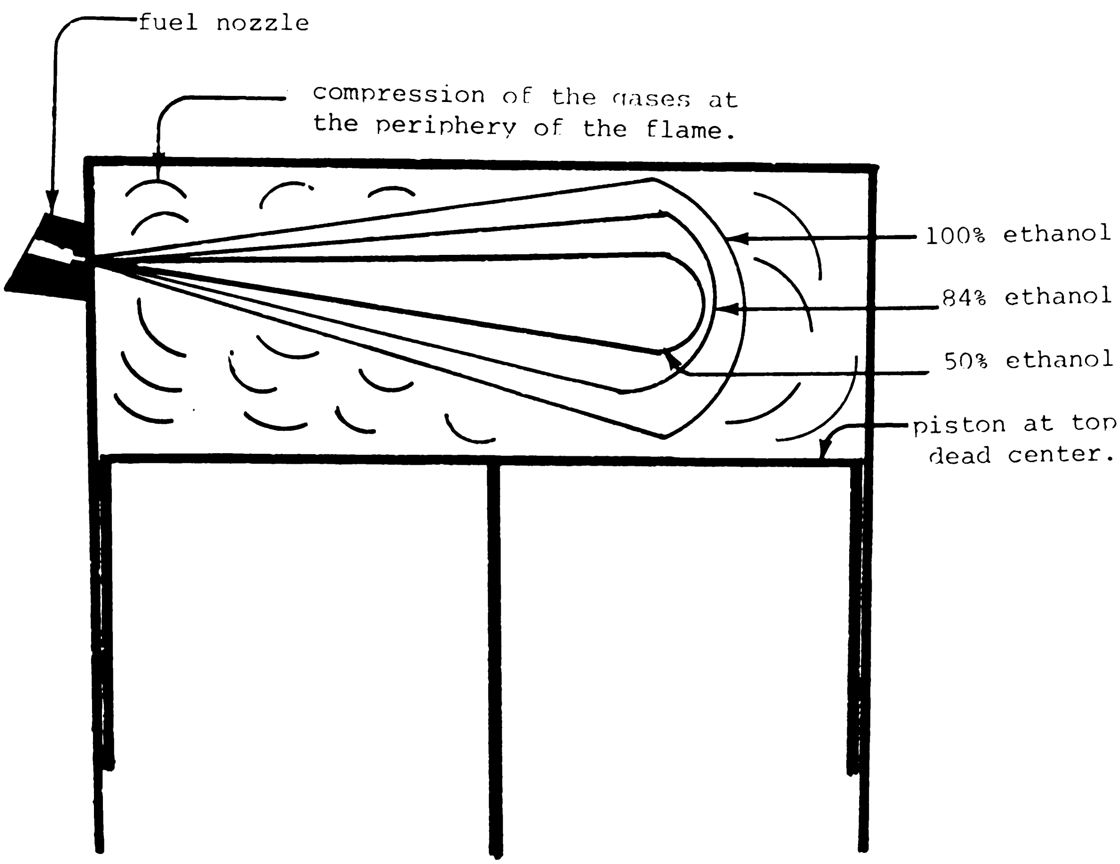
$$df = \sqrt{\frac{1.64 \times (1.2)^2 \times 0.025}{27}}$$

df = 0.043 in diameter of fuel orifice (jet)

SCHEMATIC OF THE DISINTEGRATION OF FUEL JET (16)



PATH OF DIESEL FLAME WHEN DUAL-FUELING DIFFERENT ALCOHOL/WATER MIXTURES.



APPENDIX C

APPENDIX C

STOICHIOMETRIC BALANCE

Ethanol

$C_2 H_5 OH + 3O_2 + 3 (3.76) N_2 \rightarrow 2 CO_2 + 3 H_2O + 11.28 N_2$

$(24 + 6 + 16) + (3 \times 32) + (11.28 \times 28) \rightarrow (24 + 64) + (6 + 48) +$

$(11.28 \times 28) \quad 467.84 \rightarrow 467.84$

96 kg O_2 required for 46 kg fuel

$O_2 + N_2 \rightarrow Air \rightarrow 96 + 327.14 = 423.12$ kg of air

$\frac{air}{fuel} = \frac{423.12}{46} = 9.11 \qquad \therefore AF_{stoich} = 9.11$

Methanol

$CH_3 OH + 1.5O_2 + 1.5 (3.76) N_2 \rightarrow CO_2 = 2 H_2O + 5.64 N_2$

$32 + 48 + 157.32 \rightarrow 44 + 36 + 157.92 + 237.92 \rightarrow 237.92$

48 kg O_2 required for 32 kg of fuel

$O_2 + N_2 \rightarrow Air \rightarrow 48 + 157.92 = 205.92$ kg of air

$\frac{air}{fuel} = \frac{205.92}{32} = 6.44 \qquad \therefore AF_{stoich} = 6.44$

Diesel Fuel

$C_{16} H_{34} + 24.5 O_2 + 24.5 \times 3.76 N_2 \rightarrow 16 CO_2 + 17 H_2O + 92.12 N_2$

$(226) + 784 + 2679.36 \rightarrow 704 + (306) + (2573.36)$

$3589.36 \rightarrow 3589.36$

784 kg O_2 required for 226 kg of fuel

$N_2 + O_2 = \text{air} \quad 784 + 2479.36 = 3363.36$

$$\frac{\text{air}}{\text{fuel}} = \frac{3363.36}{226} = 14.89 \qquad \therefore AF_{stoich} = 14.89$$

APPENDIX D

TEST AND COMPUTED DATA

Diesel Fuel Test with a Turbocharged Ford 7700 Diesel Tractor

Engine Test Data

Run #	Power hp	Diesel fuel ml	Diesel fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Temperature before carb. °C	Temperature after carb. °C
1	0.0	100	35.0	280	3.0	1.10	25.3	56.6
2	30.0	100	20.0	360	8.1	1.65	25.7	96.4
3	40.0	100	19.0	460	9.0	1.80	24.9	101.7
4	50.0	100	17.5	500	10.4	1.90	24.9	110.1
5	60.0	100	16.0	530	12.0	2.10	25.0	119.5
6	70.0	100	14.5	570	14.0	2.30	24.8	134.0

Notes:
1. Average ambient temperature 20° C.
2. Barometric pressure 28.85 in/hr.
3. Engine speed 2100.
4. Shaft PTO speed 1000.

Dual-fueling a Turbocharged Diesel Tractor with Spray Injection Approach

Computed Test Parameters

Run #	Diesel flow	Air flow kg/hr	Brake torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Volumetric efficiency %
1	8.8	246.1	0.0	0.0	0.0	28.1	79.7
2	15.4	301.4	231.0	24.2	12.6	19.6	97.7
3	16.2	314.8	308.0	32.3	15.9	19.5	102.0
4	17.6	323.4	385.0	40.3	18.3	18.4	104.8
5	19.2	340.0	462.0	48.4	20.1	17.7	110.2
6	21.2	355.9	539.0	56.5	21.2	16.8	115.3

Notes:
1. Average ambient temperature 20° C.
2. Barometric pressure 28.85 in/hg.
3. Engine speed 2100.
4. Shaft PTO speed 1000.
5. Date 02/28/81.

Diesel Fuel Test with a Turbocharged Ford 7700 Diesel Tractor

Engine Test Data

Run #	Power hp	Diesel fuel ml	Diesel fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Temperature before carb. °C	Temperature after carb. °C
1	0.0	100	35.0	180	~1.0	0.60	26.6	40.8
2	43.0	100	21.5	350	4.9	0.85	25.4	63.0
3	50.0	100	20.0	400	5.9	0.90	25.7	72.6
4	57.0	100	19.5	440	7.1	0.95	25.7	73.2
5	64.0	100	18.5	480	8.5	1.05	26.6	91.5
6	71.0	100	17.5	520	9.8	1.10	37.2	100.5

Notes:
1. Average ambient temperature 20° C.
2. Barometric pressure 28.85 in/hg.
3. Engine speed 1600.
4. Shaft PTO speed 755.

Dual-fueling a Turbocharged Diesel Tractor with Spray Injection Approach

Computed Test Parameters

Run #	Diesel flow	Air flow kg/hr	Brake torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Volumetric efficiency %
1	8.8	181.8	0.0	0.0	0.0	20.7	58.9
2	14.3	216.3	438.6	34.7	19.4	15.2	70.1
3	15.4	222.6	509.9	40.3	20.9	14.5	72.1
4	15.8	228.7	581.3	46.0	23.3	14.5	74.1
5	16.6	240.4	652.7	51.6	24.8	14.5	77.9
6	17.6	246.1	724.1	57.3	26.0	14.0	79.7

Notes:
1. Average ambient air temperature 20° C.
2. Barometric condition 28.85 in/hg.
3. Engine speed 1600.
4. Shaft PTO speed 755.

Diesel Fuel Test with a Turbocharged Ford 7700 Diesel Tractor

Engine Test Data

Run #	Power hp	Diesel fuel ml	Diesel fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Temperature before carb. °C	Temperature after carb. °C
1	0.0	100	45.5	180	~ .5	0.30	27.7	40.1
2	10.0	100	37.5	230	~1.0	0.30	26.0	38.4
3	20.0	100	33.0	280	1.8	0.35	25.1	40.6
4	30.0	100	29.0	350	2.5	0.40	24.8	45.7
5	40.0	100	24.0	420	3.9	0.45	25.0	55.2
6	47.0	100	21.5	490	5.0	0.45	23.6	66.0

Notes:
1. Average ambient temperature 20° C.
2. Barometric pressure 28.85 in/hg.
3. Engine speed 1150.
4. Shaft PTO speed 540.

Dual-fueling a Turbocharged Diesel Tractor with Spray Injection Approach

Computed Test Parameters

Run #	Diesel flow	Air flow kg/hr	Brake torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Volumetric efficiency %
1	6.8	128.5	0.0	0.0	0.0	19.0	41.6
2	8.2	129.6	142.6	8.1	7.9	15.8	42.0
3	9.3	138.8	285.2	16.1	13.8	14.9	45.0
4	10.6	148.4	427.8	24.2	18.2	14.0	48.1
5	12.8	157.4	570.4	32.3	20.1	12.3	51.0
6	14.3	158.3	670.2	37.9	21.2	11.1	51.3

Notes:
1. Average ambient temperature 20° C.
2. Barometric pressure 28.85 in/hg.
3. Engine speed 1150.
4. Shaft PTO speed 540.
5. Date 2/28/81.

Alcohol Fuel Test (Dual-fueling with Carbureted Alcohol)

Engine Test Data

Run #	Power hp	Diesel fuel ml	Diesel fuel sec.	Alcohol fuel ml	Alcohol fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Temperature before carb. °C	Temperature after carb. °C
1	0.0	100	45.5	100	23.5	310	3.5	1.25	22.4	23.3
2	30.0	100	41.0	100	21.0	370	6.8	1.65	22.8	55.1
3	40.0	100	37.5	100	22.5	390	7.3	1.70	23.0	60.9
4	50.0	100	31.0	100	25.5	410	8.2	1.80	23.6	71.1
5	60.0	100	28.5	100	25.5	430	9.2	1.90	24.1	77.6
6	70.0	100	21.0	100	36.0	480	11.6	2.15	25.3	101.1
7	80.0	100	18.0	100	37.0	530	13.9	2.35	25.9	117.1
8	90.0	100	16.5	100	35.5	580	15.8	2.55	26.6	130.3
9	95.0	100	15.5	100	35.0	600	17.0	3.65	27.5	138.6

Notes:
1. The water content of the alcohol 0.0% by volume.
2. Average ambient temperature 20° C.
3. Barometric pressure 28.85 in/hg.
4. Engine speed 2100.
5. Shaft PTO speed 1000.

Dual-fueling a Turbocharged Diesel Tractor with Carbureted Alcohol

Computed Test Parameters

Run #	Diesel flow kg/hr	Alcohol flow kg/hr	Air flow kg/hr	Brake torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	6.8	12.0	262.3	0.0	0.0	0.0	38.9	21.8	53.0	85.0
2	7.5	13.5	301.4	231.0	24.2	12.1	40.2	22.4	53.2	97.7
3	8.2	12.6	305.9	308.0	32.3	16.0	37.4	24.4	49.2	99.1
4	9.9	10.7	314.8	385.0	40.3	19.3	31.8	29.5	40.5	102.0
5	10.8	11.1	323.4	462.0	48.4	21.7	30.0	29.2	39.4	104.8
6	14.6	7.9	344.1	539.0	56.5	23.0	23.5	43.8	25.3	111.5
7	17.1	7.6	359.7	616.0	64.5	23.5	21.1	47.1	22.1	116.6
8	18.6	8.0	374.7	693.0	72.6	24.5	20.1	47.1	21.3	121.4
9	19.8	8.1	382.0	731.5	76.6	24.5	19.3	47.3	20.5	123.8

Notes:
1. The water content of the alcohol 0.0% by volume.
2. Average ambient temperature 20° C.
3. Barometric pressure 28.85 in/hg.
4. Engine speed 2100.
5. Shaft PTO speed 1000.
6. Date 02/28/81.

Alcohol Fuel Test (Dual-fueling with Carbureted Alcohol)

Engine Test Data

Run #	Power hp	Diesel fuel ml	Diesel fuel sec.	Alcohol fuel ml	Alcohol fuel sec.	Exhaust temp. °C	Turbo Pressure psi	ΔP in/H_2O	Temperature before carb. °C	Temperature after carb. °C
1	0.0	100	40.0	100	37.0	250	2.0	0.65	25.4	19.8
2	43.0	100	31.5	100	36.5	350	4.5	0.85	24.5	41.7
3	50.0	100	30.0	100	35.5	370	5.2	0.90	24.7	45.8
4	57.0	100	28.5	100	34.5	400	5.9	0.90	24.6	50.0
5	64.0	100	24.5	100	44.0	450	7.1	0.95	25.6	66.9
6	71.0	100	22.0	100	43.5	480	8.4	1.10	25.9	74.9
7	77.5	100	21.5	100	46.0	520	9.9	1.15	26.7	85.8
8	84.0	100	21.0	100	44.5	540	10.9	1.20	26.9	91.7
9	92.0	100	18.5	100	44.0	600	13.4	1.35	27.2	107.8

Notes:
1. The water content of the alcohol 0.0% by volume.
2. Average ambient temperature 20° C.
3. Barometric pressure 28.85 in/hg.
4. Engine speed 755.
5. Shaft PTO speed 1600.

Dual-fueling a Turbocharged Diesel Tractor with Carbureted Alcohol

Computed Test Parameters

Run #	Diesel flow kg/hr	Alcohol flow kg/hr	Air flow kg/hr	Brake torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	7.7	7.6	189.2	0.0	0.0	0.0	24.6	24.8	38.6	61.3
2	9.8	7.7	216.3	438.6	34.7	18.9	22.2	27.9	33.4	70.1
3	10.2	8.0	222.6	509.9	40.3	21.1	21.8	28.0	32.9	72.1
4	10.8	8.2	222.6	581.3	46.0	23.0	20.7	27.2	32.4	72.1
5	12.5	6.4	228.7	652.7	51.6	24.8	18.3	35.6	24.5	74.1
6	14.0	6.5	246.1	724.1	57.3	25.3	17.6	37.9	22.7	79.7
7	14.3	6.1	251.6	790.4	62.5	27.4	17.6	41.0	21.4	81.5
8	14.6	6.4	257.0	856.7	67.7	29.0	17.6	40.5	21.5	83.3
9	16.6	6.4	272.6	938.3	74.2	28.6	16.4	42.5	19.6	88.3

Notes:
1. The water content of the alcohol 0.0% by volume.
2. Average ambient temperature 20° C.
3. Barometric pressure 28.85 in/hg.
4. Engine speed 1600.
5. Shaft PTO speed 755.
6. Date 02/28/81.

Alcohol Fuel Test (Dual-fueling with Carbureted Alcohol)

Engine Test Data

Run #	Power hp	Diesel ml	Diesel fuel sec.	Alcohol ml	Alcohol fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Temperature before carb. °C	Temperature after carb. °C
1	0.0	100	50.0	100	81.5	210	~0.5	0.30	24.9	20.9
2	10.0	100	46.5	100	80.5	220	~1.0	0.35	25.0	21.0
3	20.0	100	41.5	100	76.5	250	~1.5	0.35	24.6	23.5
4	30.0	100	36.0	100	74.0	320	2.3	0.35	24.8	30.2
5	40.0	100	32.5	100	67.5	370	3.1	0.40	25.2	37.7
6	50.0	100	29.0	100	60.0	440	4.4	0.45	25.6	45.8

Notes:
1. The water content of the alcohol 0.0% by volume.
2. Average ambient temperature 20° C.
3. Barometric pressure 28.85 in/hg.
4. Engine speed 1150.
5. Shaft PTO speed 540.

Dual-fueling a Turbocharged Diesel Tractor with Carbureted Alcohol

Computed Test Parameters

Run #	Diesel flow kg/hr	Alcohol flow kg/hr	Air flow kg/hr	Brake torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	6.1	3.5	128.5	0.0	0.0	0.0	20.9	37.0	26.3	41.6
2	6.6	3.5	138.8	142.6	8.1	7.3	21.0	39.5	25.1	45.0
3	7.4	3.7	139.8	285.2	16.1	13.2	18.9	37.9	24.0	45.3
4	8.5	3.8	139.8	427.8	24.2	17.6	16.4	36.6	22.1	45.3
5	9.5	4.2	148.4	570.4	32.3	21.3	15.7	35.4	21.9	48.1
6	10.6	4.7	157.4	713.0	40.3	23.7	14.9	33.4	21.9	51.0

Notes:
1. The water content of the alcohol 0.0% by volume.
2. Average ambient air temperature 20° C.
3. Barometric pressure 28.85 in/hg.
4. Engine speed 1150.
5. Shaft PTO speed 540.
6. Date 02/28/81.

Alcohol Fuel Test (Dual-fueling with Carbureted Alcohol)

Engine Test Data

Run #	Power hp	Diesel ml	Diesel fuel sec.	Alcohol ml	Alcohol fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Temperature before carb. °C	Temperature after carb. °C
1	0.0	100	41.0	100	25.0	320	3.0	1.20	23.3	21.8
2	30.0	100	34.5	100	22.5	360	6.2	1.60	23.2	34.1
3	40.0	100	33.5	100	20.0	385	7.1	1.75	22.9	37.1
4	50.0	100	31.5	100	19.0	400	8.0	1.80	23.0	40.2
5	60.0	100	30.0	100	18.0	420	8.9	1.90	23.2	44.2
6	70.0	100	27.5	100	17.5	450	9.8	2.00	23.6	57.9
7	80.0	100	21.5	100	20.0	480	11.8	2.25	23.8	78.5
8	90.0	100	18.5	100	21.0	540	13.8	2.45	24.2	94.3
9	100.0	100	15.5	100	24.5	600	16.9	2.75	24.9	121.3

Notes:
1. The water content of the alcohol 19% by volume.
2. Average ambient temperature 20° C.
3. Barometric pressure 28.85 in/hg.
4. Engine speed 2100.
5. Shaft PTO speed 1000.

Dual-fueling a Turbocharged Diesel Tractor with Carbureted Alcohol

Computed Test Parameters

Run #	Diesel flow kg/hr	Alcohol flow kg/hr	Air flow kg/hr	Brake torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	7.5	11.9	257.0	0.0	0.0	0.0	34.3	21.6	45.3	83.3
2	8.9	13.2	296.8	231.0	24.2	12.2	33.3	22.5	43.7	96.2
3	9.2	14.9	310.4	308.0	32.3	15.2	33.9	20.9	45.9	100.6
4	9.8	15.7	314.8	385.0	40.3	17.9	32.3	20.1	45.6	102.0
5	10.2	16.5	323.4	462.0	48.4	20.5	31.6	19.6	45.7	104.8
6	11.2	17.0	331.8	539.0	56.5	22.5	29.7	19.5	44.3	107.5
7	14.3	14.9	352.0	616.0	64.5	23.3	24.6	23.7	35.21	114.0
8	16.6	14.2	367.3	693.0	72.6	24.1	22.1	25.9	30.8	119.0
9	19.8	12.1	383.1	770.0	80.6	24.6	19.6	32.0	24.2	126.1

Notes:
1. The water content of the alcohol 19% by volume.
2. Average ambient air temperature 20° C.
3. Barometric pressure 28.85 in/hg.
4. Engine speed 2100.
5. Shaft PTO speed 1000.
6. Date 02/28/81.

Alcohol Fuel Test (Dual-fueling with Carbureted Alcohol)

Engine Test Data

Run #	Power hp	Diesel fuel ml	Diesel fuel sec.	Alcohol fuel ml	Alcohol fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Temperature before carb. °C	Temperature after carb. °C
1	0.0	100	38.0	100	44.0	200	1.5	0.60	23.7	19.4
2	43.0	100	30.5	100	32.0	320	4.3	0.85	23.7	19.3
3	50.0	100	28.5	100	31.0	360	5.1	0.90	23.8	29.1
4	57.0	100	26.5	100	30.0	400	6.1	0.95	23.9	35.3
5	64.0	100	25.5	100	29.0	430	7.0	1.00	24.4	39.1
6	71.0	100	24.5	100	28.0	450	7.5	1.05	23.9	42.4
7	84.0	100	22.5	100	27.0	480	8.8	1.05	24.3	50.1

Notes:
1. The water content of the alcohol 19% by volume.
2. Average ambient temperature 20° C.
3. Barometric pressure 28.85 in/hg.
4. Engine speed 1600.
5. Shaft PTO speed 755.

Dual-fueling a Turbocharged Diesel Tractor with Carbureted Alcohol

Computed Test Parameters

Run #	Diesel flow kg/hr	Alcohol flow kg/hr	Air flow kg/hr	Brake torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	8.1	6.8	181.8	0.0	0.0	0.0	22.5	26.9	30.4	58.9
2	10.1	9.3	216.3	438.6	34.7	18.5	21.5	23.3	32.5	70.1
3	10.8	9.6	222.6	509.9	40.3	20.4	20.7	23.3	31.7	72.1
4	11.6	9.9	228.7	581.3	46.0	21.9	19.7	23.1	30.9	74.1
5	12.0	10.3	234.6	652.7	51.6	23.7	19.5	22.9	30.8	76.0
6	12.5	10.6	240.4	724.1	57.3	25.2	19.2	22.6	30.7	77.9
7	13.7	11.0	251.6	856.7	67.7	27.8	18.4	22.9	29.6	81.5

Notes:
1. The water content of the alcohol 19% by volume.
2. Average ambient air temperature 20° C.
3. Barometric pressure 28.85 in/hg.
4. Engine speed 1600.
5. Shaft PTO speed 755.
6. Date 02/28/81.

Alcohol Fuel Test (Dual-fueling with Carbureted Alcohol)

Engine Test Data

Run #	Power hp	Diesel ml	fuel sec.	Alcohol ml	fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Temperature before carb. °C	Temperature after carb. °C
1	0.0	100	52.0	100	62.0	180	~ .5	0.30	23.6	13.6
2	10.0	100	46.0	100	60.0	230	~1.0	0.35	22.4	16.4
3	20.0	100	41.0	100	61.0	240	1.7	0.35	22.4	18.1
4	30.0	100	36.0	100	57.5	290	2.2	0.40	23.1	21.2
5	40.0	100	32.0	100	53.0	360	3.1	0.40	23.8	25.4
6	50.0	100	28.5	100	48.5	420	4.0	0.45	24.4	30.6
7	60.0	100	25.0	100	43.0	500	6.0	0.55	24.8	39.3

Notes:
1. The water content of the alcohol 19% by volume.
2. Average ambient temperature 20° C.
3. Barometric pressure 28.85 in/hg.
4. Engine speed 1150.
5. Shaft PTO speed 540.

Dual-fueling a Turbocharged Diesel Tractor with Carbureted Alcohol

Computed Test Parameters

Run #	Diesel flow kg/hr	Alcohol flow kg/hr	Air flow kg/hr	Brake torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	5.9	4.8	128.5	0.0	0.0	0.0	21.8	26.8	29.8	41.6
2	6.7	5.0	138.8	142.6	8.1	6.9	20.8	28.0	27.9	45.0
3	7.5	4.9	139.8	285.2	16.1	12.8	18.7	28.7	25.4	45.3
4	8.5	5.2	148.4	427.8	24.2	17.2	17.4	28.7	24.0	48.1
5	9.6	5.6	149.3	570.4	32.3	20.5	15.6	26.6	23.4	48.4
6	10.8	6.1	157.4	713.0	40.3	23.0	14.6	25.7	22.9	51.0
7	12.3	6.9	174.0	855.6	48.4	24.3	14.2	25.2	22.7	56.4

Notes:
1. The water content of the alcohol 19% by volume.
2. Average ambient air temperature 20° C.
3. Barometric pressure 28.85 in/hg.
4. Engine speed 1150.
5. Shaft PTO speed 540.
6. Date 02/28/81.

Alcohol Fuel Test (Dual-fueling with Carbureted Alcohol

Engine Test Data

Run #	Power hp	Diesel ml	fuel sec.	Alcohol ml	fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Temperature before carb. °C	Temperature after carb. °C
1	0.0	100	36.0	100	22.5	270	2.6	1.15	23.7	21.7
2	30.0	100	28.5	100	18.5	360	5.7	1.60	23.3	31.6
3	40.0	100	27.0	100	16.0	390	6.4	1.65	23.6	33.4
4	50.0	100	25.0	100	14.5	420	7.5	1.80	23.9	36.4
5	60.0	100	23.0	100	14.5	450	8.6	1.95	24.0	39.0
6	70.0	100	21.5	100	14.5	480	9.7	2.05	24.2	41.8
7	80.0	100	16.5	100	12.0	520	10.2	2.15	24.3	42.7
8	90.0	100	15.0	100	12.0	550	11.4	2.25	24.4	45.5
9	100.0	100	17.5	100	12.0	580	12.7	2.40	24.6	48.9
10	112.0	100	14.5	100	9.5	600	14.5	2.60	25.1	56.7

Notes:
1. The water content of the alcohol 50% by volume.
2. Average ambient temperature 20° C.
3. Barometric pressure 29.1 in/hg.
4. Engine speed 2100.
5. Shaft PTO speed 1000.

Dual-fueling a Turbocharged Diesel Tractor with Carbureted Alcohol

Computed Test Parameters

Run #	Diesel flow kg/hr	Alcohol flow kg/hr	Air flow kg/hr	Brake torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	8.5	14.3	251.6	0.0	0.0	0.0	29.5	17.6	35.0	81.5
2	10.8	17.4	296.8	229.0	24.0	11.7	27.5	17.1	34.2	96.2
3	11.4	20.1	301.4	305.4	32.0	14.3	26.5	15.0	36.3	97.7
4	12.3	22.2	314.8	381.7	40.0	16.4	25.6	14.2	36.8	102.0
5	13.4	22.2	327.7	458.0	48.0	18.7	24.5	14.8	34.9	106.2
6	14.3	22.2	336.4	534.4	56.0	20.8	23.6	15.2	33.3	109.0
7	18.6	26.8	344.5	610.7	64.0	18.7	18.5	12.9	31.7	111.6
8	20.5	26.8	352.0	687.1	72.0	19.7	17.2	13.2	29.7	114.0
9	17.6	26.8	363.9	763.4	79.4	24.3	20.7	13.6	33.0	117.9
10	21.2	33.8	378.3	855.0	89.5	22.3	17.9	11.2	34.0	122.6

Notes:
1. The water content of the alcohol 50% by volume.
2. Average ambient air temperature 20° C.
3. Barometric pressure 29.1 in/hg.
4. Engine speed 2100.
5. Shaft PTO speed 1000.
6. Date 03/03/81.

Alcohol Fuel Test (Dual-fueling with Carbureted Alcohol)

Engine Test Data

Run #	Power hp	Diesel ml	fuel sec.	Alcohol ml	fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Temperature before carb. °C	Temperature after carb. °C
1	0.0	100	37.0	100	48.5	180	1.5	0.60	21.2	18.9
2	43.0	100	27.0	100	25.5	340	4.0	0.80	21.7	24.8
3	50.0	100	25.0	100	24.5	380	5.0	0.85	22.2	27.3
4	57.0	100	24.0	100	23.5	410	6.0	0.90	22.8	30.3
5	64.0	100	23.0	100	22.5	440	6.7	0.95	23.1	31.9
6	71.0	100	21.5	100	21.5	480	7.8	1.05	23.2	34.4
7	78.0	100	20.5	100	21.0	510	8.9	1.15	24.0	37.2
8	85.0	100	20.0	100	20.0	530	9.9	1.20	24.2	40.5
9	92.0	100	19.0	100	19.0	560	11.2	1.30	24.8	45.4
10	99.0	100	18.0	100	18.5	600	12.8	1.50	24.6	51.2

Notes:
1. The water content of the alcohol 50% by volume.
2. Average ambient temperature 20° C.
3. Barometric pressure 29.1 in/hg.
4. Engine speed 1600.
5. Shaft PTO speed 755.

Dual-fueling a Turbocharged Diesel Tractor with Carbureted Alcohol

Computed Test Parameters

Run #	Diesel flow kg/hr	Alcohol flow kg/hr	Air flow kg/hr	Brake torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	8.3	6.6	181.8	0.0	0.0	0.0	21.9	27.4	20.5	58.9
2	11.4	12.6	209.9	434.8	34.4	17.8	18.5	16.7	26.3	68.0
3	12.3	13.1	216.3	505.6	40.0	19.3	17.6	16.5	25.6	70.0
4	12.8	13.7	222.6	576.3	45.6	21.1	17.4	16.3	25.6	72.1
5	13.4	14.3	228.7	647.1	51.2	22.7	17.1	16.0	25.6	74.1
6	14.3	14.9	240.4	717.9	56.8	23.7	16.8	16.1	25.2	77.9
7	15.0	15.3	251.6	788.7	62.4	25.0	16.8	16.5	24.8	81.5
8	15.4	16.1	257.0	859.5	68.0	26.4	16.7	16.0	25.2	83.3
9	16.2	16.9	267.5	930.2	73.6	27.1	16.6	15.8	25.2	86.7
10	17.1	17.4	287.4	1001.0	79.1	27.8	16.8	16.6	24.7	93.1

Notes:
1. The water content of the alcohol 50% by volume.
2. Average ambient air temperature 20° C.
3. Barometric pressure 29.1 in/hg.
4. Engine speed 1600.
5. Shaft PTO speed 755.
6. Date 03/03/81.

Alcohol Fuel Test (Dual-fueling with Carbureted Alcohol)

Engine Test Data

Run #	Power hp	Diesel ml	fuel sec.	Alcohol ml	fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Temperature before carb. °C	Temperature after carb. °C
1	0.0	100	43.0	100	82.5	130	~ .5	0.30	20.6	14.1
2	10.0	100	39.5	100	54.5	180	~1.0	0.30	20.4	15.5
3	20.0	100	35.5	100	47.0	240	~1.5	0.30	20.7	17.7
4	30.0	100	31.5	100	45.0	290	2.2	0.35	20.9	20.4
5	40.0	100	28.5	100	41.0	370	3.0	0.40	21.3	24.2
6	50.0	100	25.5	100	37.0	450	4.6	0.45	22.2	30.1

Notes:
1. The water content of the alcohol 50% by volume.
2. Average ambient temperature 20° C.
3. Barometric pressure 29.1 in/hg.
4. Engine speed 1150.
5. Shaft PTO speed 540.

Dual-fueling a Turbocharged Diesel Tractor with Carbureted Alcohol

Computed Test Parameters

Run #	Diesel flow kg/hr	Alcohol flow kg/hr	Air flow kg/hr	Brake torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	7.1	3.9	128.5	0.0	0.0	0.0	18.0	33.0	15.0	41.6
2	7.8	5.9	128.5	141.4	8.0	6.6	16.5	21.8	19.6	41.6
3	8.7	6.8	129.6	282.7	16.0	11.7	15.0	19.0	20.3	42.0
4	9.8	7.1	138.8	424.1	24.0	16.0	14.2	19.4	19.1	45.0
5	10.8	7.8	148.4	565.5	32.0	19.2	13.8	18.9	19.0	48.1
6	12.0	8.7	157.4	706.9	40.0	21.5	13.1	18.1	18.9	51.0

Notes:
1. The water content of the alcohol 50% by volume.
2. Average ambient temperature 20° C.
3. Barometric pressure 29.1 in/hg.
4. Engine speed 1150.
5. Shaft PTO speed 540.
6. Date 03/03/81.

Alcohol Fuel Test (Dual-fueling with Carbureted Alcohol)

Engine Test Data
Fully Automatic

Run #	Power hp	Diesel fuel ml	Diesel fuel sec.	Alcohol fuel ml	Alcohol fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Temperature before carb. °C	Temperature after carb. °C
1	0.0	100	42.0	100	35.5	230	2.9	1.05	24.0	23.2
2	30.0	100	30.0	100	26.5	340	6.2	1.50	24.0	37.7
3	40.0	100	28.0	100	25.5	380	7.6	1.65	24.0	54.0
4	50.0	100	26.5	100	23.5	410	8.5	1.75	24.0	60.6
5	60.0	100	25.5	100	22.0	430	9.5	1.85	24.0	66.3
6	70.0	100	23.5	100	21.0	460	10.5	1.95	24.0	71.7
7	80.0	100	21.0	100	20.5	490	11.6	2.05	24.0	72.6
8	84.0	100	20.5	100	19.5	510	12.5	2.10	24.0	85.4

Notes:
1. The water content of the alcohol 20% by volume.
2. Average ambient temperature 20° C.
3. Barometric pressure 29.32 in/hg.
4. Engine speed 2100.
5. Shaft PTO speed 1000.

Dual-fueling a Turbocharged Diesel Tractor with Carbureted Alcohol

Computed Test Parameters
Fully Automatic

Run #	Diesel flow kg/hr	Alcohol flow kg/hr	Air flow kg/hr	Brake torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	7.3	8.4	240.4	0.0	0.0	0.0	33.0	28.6	37.0	78.0
2	10.2	11.3	287.4	227.3	23.8	11.8	28.1	25.6	36.2	93.1
3	11.0	11.7	301.4	303.1	31.8	15.0	27.5	25.8	35.5	97.7
4	11.6	12.7	310.4	378.8	39.7	17.4	26.8	24.5	36.1	100.6
5	12.0	13.6	319.2	454.6	47.6	19.9	26.5	23.6	36.7	103.4
6	13.1	14.2	327.7	530.4	55.5	21.7	25.1	23.1	35.9	106.2
7	14.6	14.5	336.0	606.1	63.5	23.0	23.0	23.1	34.0	108.9
8	15.0	15.2	340.0	636.4	66.7	23.2	22.7	22.3	34.5	110.2

Note:
1. The water content of the alcohol 20% by volume.
2. Average ambient air temperature 20° C.
3. Barometric pressure 29.32 in/hg.
4. Engine speed 2100.
5. Shaft PTO speed 1000.
6. Date 03/14/81.

Alcohol Fuel Test (Dual-fueling with Spray Injection Approach)

Engine Test Data

Run #	Force lbs	Diesel ml	fuel sec.	Alcohol ml	fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Diesel flow Kg/hr	Alcohol flow Kg/hr	Air flow Kg/hr
1	40	100	28.0	30	31.0	300	4.5	1.40	10.97	2.74	277.63
2	80	100	25.0	30	26.0	350	5.6	1.55	12.28	3.26	292.13
3	120	100	22.5	30	21.0	340	7.0	1.75	13.65	4.04	310.40
4	160	100	20.5	30	17.0	440	8.5	1.90	14.98	4.99	323.43
5	200	100	20.0	30	16.0	460	9.5	2.00	15.36	5.30	331.83
6	240	100	18.0	30	14.5	515	11.25	2.25	17.06	5.85	351.96
7	280	100	16.5	20	8.5	550	13.1	2.45	18.61	6.65	367.27
8	320	100	15.0	20	7.8	620	15.2	2.70	20.47	7.25	385.56

Notes:
1. The water content of the alcohol 0.0% by volume.
2. Average ambient temperature 20° C.
3. Barometric pressure 29.45 in/hg
4. M & W Gear nozzle 0.51 mm.
5. Engine speed 2100.
6. Shaft PTO speed 1000.

Dual-fueling a Turbocharged Diesel Tractor with Spray Injection Approach

Computed Test Parameters

Run #	Brake Torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	81.34	8.96	5.61	25.31	101.52	13.85	89.95
2	162.69	17.93	9.93	23.78	89.59	14.61	94.65
3	244.03	26.89	13.19	22.74	76.89	16.01	100.57
4	325.38	35.86	15.71	21.59	64.85	17.67	104.79
5	406.72	44.82	19.03	21.61	62.63	18.20	107.52
6	488.06	53.79	20.58	20.63	60.20	18.09	114.04
7	569.41	62.75	21.84	19.73	55.23	18.72	119.00
8	650.75	71.71	22.73	18.83	53.21	18.58	124.92

Notes:
1. The water content of the alcohol 0.0% by volume.
2. Average ambient air temperature 20° C.
3. Barometric pressure 29.45 in/hg.
4. M & W Gear nozzle 0.51 mm.
5. Engine speed 2100 rpm.
6. Shaft PTO speed 1000.
7. Date 11/14/80.

Alcohol Fuel Test (Dual-fueling with Spray Injection Approach)

Engine Test Data

Run #	Force lbs	Diesel ml	fuel sec.	Alcohol ml	fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Diesel flow Kg/hr	Alcohol flow Kg/hr	Air flow Kg/hr
1	40	100	30.5	30	17.5	340	4.6	1.35	10.07	4.85	272.63
2	80	100	28.0	30	14.5	360	5.6	1.55	10.97	5.85	292.13
3	120	100	26.0	30	13.0	390	6.8	1.65	11.81	6.52	301.40
4	160	100	24.0	30	11.0	420	7.9	1.85	12.80	7.71	319.15
5	200	100	23.5	20	7.5	440	8.5	1.90	13.07	7.54	323.43
6	240	100	21.0	20	6.5	480	9.9	2.10	14.62	8.70	340.03

Notes:
1. The water content of the alcohol 0.0% by volume.
2. Average ambient temperature 18° C.
3. Barometric pressure 29.45 in/hg.
4. M & W Gear nozzle 0.63 mm.
5. Engine speed 2060.

Dual-fueling a Diesel Tractor with M & W Gear Kit

Computed Test Parameters

Run #	Brake Torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	81.34	8.96	5.42	27.08	56.28	23.67	88.33
2	162.69	17.93	9.70	26.63	49.96	25.57	94.65
3	244.03	26.89	13.39	25.52	46.22	26.25	97.66
4	325.38	35.86	16.09	24.94	41.41	27.97	103.41
5	406.72	44.82	19.93	24.75	42.92	21.10	104.79
6	488.06	53.79	21.19	23.25	39.11	27.71	110.17

Notes:
1. The water content of the alcohol 0.0% by volume
2. Average ambient air temperature 18° C.
3. Barometric condition 29.45 in/hg.
4. M & W Gear nozzle 0.63 mm.

Alcohol Fuel Test (Dual-fueling with Spray Injection Approach)

Engine Test Data

Run #	Force lbs	Diesel ml	fuel sec.	Alcohol ml	fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Diesel flow Kg/hr	Alcohol flow Kg/hr	Air flow Kg/hr
1	40	100	33.0	30	13.0	320	4.4	1.35	9.31	6.52	272.63
2	80	100	31.0	30	11.0	350	5.6	1.55	9.91	7.71	292.13
3	120	100	29.5	20	6.8	380	6.4	1.70	10.41	8.31	305.94
4	160	100	27.5	20	5.8	410	7.6	1.85	11.17	9.75	319.15
5	200	100	26.0	20	5.5	440	8.5	2.00	11.81	10.28	331.83

Notes:
1. The water content of the alcohol 0.0% by volume.
2. Average ambient temperature 18° C.
3. Barometric pressure 29.45 in/hg.
4. M & W Gear nozzle 0.76 mm.
5. Engine speed 2060.

Dual-fueling a Diesel Tractor with M & W Gear Kit

Computed Test Parameters

Run #	Brake Torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	81.34	8.96	5.29	29.30	41.80	31.12	88.33
2	162.69	17.93	9.61	29.49	37.90	33.40	94.65
3	244.03	26.89	13.60	29.39	36.81	33.98	99.13
4	325.38	35.86	16.38	28.58	32.75	36.00	103.41
5	406.72	44.82	19.38	28.09	32.29	35.93	107.52

Notes:
1. The water content of the alcohol 0.0% by volume
2. Average ambient air temperature 18° C.
3. Barometric condition 29.45 in/hg.
4. M & W Gear nozzle 0.76 mm.

Alcohol Fuel Test (Dual-fueling with Spray Injection Approach)

Engine Test Data

Run #	Force lbs	Diesel ml	fuel sec.	Alcohol ml	fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Diesel flow Kg/hr	Alcohol flow Kg/hr	Air flow Kg/hr
1	40	100	28.5	80	90.5	300	4.1	1.25	10.78	2.61	262.34
2	80	100	25.5	80	74.0	340	5.2	1.45	12.04	3.19	282.55
3	120	100	23.5	80	61.5	380	6.4	1.65	13.07	3.84	301.40
4	160	100	21.5	80	53.0	425	7.9	1.85	14.28	4.45	319.15
5	200	100	19.0	80	45.5	480	9.7	2.05	16.16	5.19	335.96
6	240	100	17.0	80	40.0	540	12.0	2.40	18.07	5.90	363.51
7	280	100	15.5	80	36.5	590	14.0	2.50	19.81	6.47	371.00
8	294	100	15.0	80	35.5	610	14.5	2.60	20.47	6.64	378.35

Notes:
1. The water content of the alcohol 16% by volume.
2. Average ambient temperature 24° C.
3. Barometric pressure 28.75 in/hg.
4. M & W Gear nozzle 0.51 mm.
5. Engine speed 2060.

Dual-fueling a Diesel Tractor with M & W Gear Kit

Computed Test Parameters

Run #	Brake Torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	81.34	9.25	6.05	24.35	100.61	11.58	85.00
2	162.69	18.49	10.70	23.46	88.60	12.54	91.55
3	244.03	27.74	14.60	23.06	78.55	13.72	97.66
4	325.38	36.98	17.66	22.34	71.68	14.44	103.41
5	406.72	46.23	19.42	20.79	64.78	14.80	108.85
6	488.06	55.47	20.80	20.12	61.62	15.03	117.78
7	569.41	64.72	22.13	18.73	57.38	15.02	120.21
8	592.88	67.95	22.50	18.48	56.92	14.95	122.59

Notes:
1. The water content of the alcohol 16% by volume
2. Average ambient air temperature 24° C.
3. Barometric condition 28.75 in/hg.
4. M & W Gear nozzle 0.51 mm.

Alcohol Fuel Test (Dual-fueling with Spray Injection Approach)

Engine Test Data

Run #	Force lbs	Diesel ml	fuel sec.	Alcohol ml	fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Diesel flow Kg/hr	Alcohol flow Kg/hr	Air flow Kg/hr
1	40	100	29.5	80	53.0	300	3.0	1.25	10.41	4.45	262.34
2	80	100	27.5	80	440	320	5.1	1.50	11.17	5.36	287.38
3	120	100	25.5	80	38.0	365	6.2	1.65	12.04	6.21	301.40
4	160	100	23.0	80	33.0	395	7.6	1.85	13.35	7.15	319.15
5	200	100	21.0	80	29.0	460	9.1	2.00	14.62	8.14	331.83
6	240	100	19.0	80	26.0	500	11.0	2.25	16.16	9.08	351.96
7	280	100	17.0	80	23.5	535	13.1	2.50	18.07	10.04	371.00
8	320	100	15.5	80	21.5	600	15.1	2.75	19.81	10.98	383.11

Notes:
1. The water content of the alcohol 16% by volume.
2. Average ambient temperature 24° C.
3. Barometric pressure 28.75 in/hg.
4. M & W Gear nozzle 0.63 mm.
5. Engine speed 2060.

Dual-fueling a Diesel Tractor with M & W Gear Kit

Computed Test Parameters

Run #	Brake Torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	81.34	9.25	5.75	25.20	58.92	18.80	85.00
2	162.69	18.49	10.47	25.73	53.58	20.64	93.11
3	244.03	27.74	14.35	25.03	48.53	21.83	97.66
4	325.38	36.98	17.11	23.90	44.63	22.48	103.41
5	406.72	46.23	19.36	22.69	40.78	23.15	107.52
6	488.06	55.47	20.98	21.78	38.78	23.32	114.04
7	569.41	64.72	21.95	20.54	36.95	23.14	120.21
8	650.75	73.96	22.89	19.64	35.45	23.07	126.07

Notes:
1. The water content of the alcohol 84% by volume.
2. Average ambient air temperature 24° C.
3. Barometric condition 28.75 in/hg.
4. M & W Gear nozzle 0.63 mm.

Alcohol Fuel Test (Dual-fueling with Spray Injection Approach)

Engine Test Data

Run #	Force lbs	Diesel ml	fuel sec.	Alcohol ml	fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Diesel flow Kg/hr	Alcohol flow Kg/hr	Air flow Kg/hr
1	40	100	30.5	80	39.0	320	4.6	1.40	10.07	6.05	277.63
2	80	100	29.0	80	33.0	350	5.5	1.60	10.59	7.15	296.80
3	120	100	29.0	80	28.5	380	6.6	1.75	11.37	8.28	310.40
4	160	100	25.0	80	25.5	410	7.8	1.90	12.28	9.25	323.43
5	200	100	23.0	80	23.0	450	9.0	2.05	13.35	10.26	335.96
6	240	100	21.0	80	20.5	490	10.4	2.20	14.62	11.51	348.03
7	280	100	19.0	80	18.5	520	12.4	2.45	16.16	12.76	367.27
8	320	100	17.0	80	16.0	600	15.1	2.80	18.07	14.75	392.63

Notes:
1. The water content of the alcohol 16% by volume.
2. Average ambient temperature 24° C.
3. Barometric pressure 28.75 in/hg.
4. M & W Gear nozzle 0.76 mm.
5. Engine speed 2060.

Dual-fueling a Diesel Tractor with M & W Gear Kit

Computed Test Parameters

Run #	Brake Torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	81.34	9.25	5.52	27.57	45.88	24.55	89.95
2	162.69	18.49	10.19	28.03	41.50	26.77	96.17
3	244.03	27.74	13.94	27.29	37.49	28.27	100.57
4	325.38	36.98	17.04	26.33	34.95	28.97	104.79
5	406.72	46.23	19.49	25.16	32.74	29.38	108.85
6	488.06	55.47	21.20	23.80	30.23	29.88	112.76
7	569.41	64.72	22.36	22.72	28.79	29.94	119.00
8	650.75	73.96	22.63	21.74	26.62	30.66	127.22

Notes:
1. The water content of the alcohol 16% by volume.
2. Average ambient air temperature 24° C.
3. Barometric condition 28.75 in/hg.
4. M & W Gear nozzle 0.76 mm.

Alcohol Fuel Test (Dual-fueling with Spray Injection Approach)

Engine Test Data

Run #	Force lbs	Diesel fuel ml	Diesel fuel sec.	Alcohol fuel ml	Alcohol fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Diesel flow Kg/hr	Alcohol flow Kg/hr	Air flow Kg/hr
1	40	100	33.5	60	17.0	300	4.5	1.45	9.17	10.41	282.55
2	80	100	32.5	60	14.0	320	5.7	1.65	9.45	12.64	301.40
3	120	100	31.0	50	10.5	340	6.7	1.80	9.91	14.05	314.81
4	160	100	30.0	50	9.5	370	7.6	1.95	10.24	15.53	327.66
5	200	100	29.0	50	8.5	405	8.4	2.05	10.59	17.35	335.96
6	240	100	27.5	40	6.5	455	9.6	2.20	11.17	18.15	348.03
7	280	100	25.0	40	6.0	470	10.9	2.30	12.28	19.67	355.85

Notes:

1. The water content of the alcohol 16% by volume.
2. Average ambient temperature 24° C.
3. Barometric pressure 29.01 in/hg.
4. M & W Gear nozzle 0.89 mm.
5. Engine speed 2060.

Dual-fueling a Diesel Tractor with M & W Gear Kit

Computed Test Parameters

Run #	Brake Torque N.M.	Power kW	Break thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	81.34	9.16	4.93	30.82	27.14	38.08	91.55
2	162.69	18.32	8.96	31.90	23.84	42.01	97.66
3	244.03	27.49	12.51	31.78	22.41	43.43	102.00
4	325.38	36.65	15.67	32.01	21.11	45.09	106.16
5	406.72	45.81	18.27	31.73	19.36	47.01	108.85
6	488.06	54.97	20.87	31.17	19.17	46.81	112.76
7	569.41	64.14	22.29	28.97	18.10	46.43	115.30

Notes:

1. The water content of the alcohol 16% by volume.
2. Average ambient air temperature 24° C.
3. Barometric condition 27.75 in/hg.
4. M & W Gear nozzle 0.89 mm.

Alcohol Fuel Test (Dual-fueling with Spray Injection Approach)

Engine Test Data

Run #	Force lbs	Diesel ml	fuel sec.	Alcohol ml	fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Diesel flow Kg/hr	Alcohol flow Kg/hr	Air flow Kg/hr
1	40	100	38.0	80	17.5	310	4.8	1.45	8.08	13.49	282.55
2	80	100	36.5	70	13.0	320	5.8	1.65	8.41	15.88	301.40
3	120	100	35.0	70	11.5	360	6.8	1.80	8.77	17.96	314.81
4	160	100	35.0	60	9.0	385	7.8	1.95	8.77	19.67	327.66
5	200	100	35.0	60	8.5	400	8.9	2.10	8.77	20.82	340.03

Notes:
1. The water content of the alcohol 16% by volume.
2. Average ambient temperature 24° C.
3. Barometric pressure 28.75 in/hg.
4. M & W Gear nozzle 1.0 mm.
5. Engine speed 2060.

Dual-fueling a Diesel Tractor with M & W Gear Kit

Computed Test Parameters

Run #	Brake Torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air diesel ratio	% of energy from alcohol	Volumetric efficiency %
1	81.34	9.25	4.79	34.96	20.95	47.46	91.55
2	162.69	18.49	8.66	35.82	18.98	50.55	97.66
3	244.03	27.74	11.95	35.88	17.53	52.56	102.00
4	325.38	36.98	15.18	37.34	16.66	54.82	106.16
5	406.72	46.22	18.38	38.75	16.33	56.24	110.17

Notes:
1. The water content of the alcohol 16% by volume
2. Average ambient air temperature 24° C.
3. Barometric condition 28.75 in/hg.
4. M & W Gear nozzle 1.0 mm.

Alcohol Fuel Test (Dual-fueling with Spray Injection Approach)

Engine Test Data

Run #	Force lbs	Diesel ml	Diesel fuel sec.	Alcohol ml	Alcohol fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Diesel flow Kg/hr	Alcohol flow Kg/hr	Air flow Kg/hr
1	40	100	26.0	100	108.0	300	4.7	1.50	11.81	2.98	287.38
2	80	100	22.5	90	80.0	360	5.8	1.65	13.65	3.62	301.40
3	120	100	20.0	90	67.5	410	7.2	1.80	15.36	4.29	314.81
4	160	100	18.5	90	58.0	460	9.0	2.05	16.60	5.00	335.96
5	200	100	16.5	90	50.5	520	10.8	2.25	18.61	5.74	351.96
6	240	100	15.0	90	44.0	560	13.0	2.55	20.47	6.59	374.69
7	263	100	14.5	90	42.5	590	13.9	2.60	21.18	6.82	378.35

Notes:
1. The water content of the alcohol 50% by volume.
2. Average ambient temperature 20° C.
3. Barometric pressure 29.45 in/hg.
4. M & W Gear nozzle 0.51 mm.
5. Engine speed 2060.

Dual-fueling a Diesel Tractor with M & W Gear Kit

Computed Test Parameters

Run #	Brake Torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	81.34	8.96	5.59	24.33	96.39	7.51	93.11
2	162.69	17.93	9.65	22.08	83.20	7.87	97.66
3	244.03	26.89	12.81	20.50	73.32	8.26	102.00
4	325.38	35.86	15.70	20.24	67.24	8.83	108.85
5	406.72	44.82	17.46	18.91	61.33	9.03	114.04
6	488.06	53.79	18.98	18.30	56.89	9.38	121.40
7	534.84	58.94	20.10	17.86	55.49	9.39	122.59

Notes:
1. The water content of the alcohol 50% by volume
2. Average ambient air temperature 20° C.
3. Barometric condition 29.45 in/hg.
4. M & W Gear nozzle 0.51 mm.
5. Engine speed 2060 rpm.
6. Date 11/14/80.

Alcohol Fuel Test (Dual-fueling with Spray Injection Approach)

Engine Test Data

Run #	Force lbs	Diesel ml	fuel sec.	Alcohol ml	fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Diesel flow Kg/hr	Alcohol flow Kg/hr	Air flow Kg/hr
1	40	100	26.5	100	74.5	310	4.5	1.45	11.59	4.32	282.55
2	80	100	24.0	100	62.5	350	5.5	1.60	12.80	5.15	296.80
3	120	100	21.5	100	51.0	390	6.9	1.80	14.28	6.31	314.81
4	160	100	19.5	100	44.0	435	8.4	2.00	15.75	7.32	331.83
5	200	100	18.0	100	40.0	480	10.0	2.25	17.06	8.05	351.96
6	240	100	16.5	100	35.0	520	11.7	2.50	18.61	9.20	371.00
7	280	100	15.0	100	31.0	575	13.9	2.75	20.47	10.39	389.12
8	287	100	14.5	100	30.5	590	14.1	2.80	21.18	10.56	392.63

Notes:
1. The water content of the alcohol 50% by volume.
2. Average ambient temperature 23° C.
3. Barometric pressure 28.62 in/hg.
4. M & W Gear nozzle 0.63 mm.
5. Engine speed 2060.

Dual-fueling a Diesel Tractor with M & W Gear Kit

Computed Test Parameters

Run #	Brake Torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	81.34	9.29	5.70	24.38	65.37	10.72	91.55
2	162.69	18.57	10.24	23.20	57.61	11.45	96.17
3	244.03	27.86	13.61	22.04	49.86	12.46	102.00
4	325.38	37.15	16.36	21.07	45.34	13.01	107.52
5	406.72	46.43	18.83	20.63	43.72	13.18	114.04
6	488.06	55.72	20.59	19.93	40.33	13.73	120.21
7	569.41	65.01	21.76	19.01	37.46	14.04	126.07
8	583.64	66.63	21.61	18.54	37.19	13.83	127.22

Notes:
1. The water content of the alcohol 50% by volume.
2. Average ambient air temperature 23° C.
3. Barometric pressure 28.62 in/hg.
4. M & W Gear nozzle 0.63 mm.

Alcohol Fuel Test (Dual-fueling with Spray Injection Approach)

Engine Test Data

Run #	Force lbs	Diesel ml	fuel sec.	Alcohol ml	fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Diesel flow Kg/hr	Alcohol flow Kg/hr	Air flow Kg/hr
1	40	100	28.0	90	51.0	285	4.0	1.40	10.97	5.68	277.63
2	80	100	25.5	90	41.0	330	5.2	1.55	12.04	7.07	292.13
3	120	100	23.0	90	35.5	355	6.5	1.70	13.35	8.16	305.94
4	160	100	20.5	90	31.0	410	8.2	2.00	14.98	9.35	331.83
5	200	100	19.0	90	26.0	450	9.5	2.20	16.16	11.15	348.03
6	240	100	17.0	90	23.0	500	12.6	2.50	18.07	12.60	371.00
7	280	100	15.5	90	21.0	540	13.4	2.80	19.81	13.80	392.63
8	312	100	14.5	90	19.5	580	15.0	2.95	21.18	14.86	403.01

Notes:

1. The water content of the alcohol 50% by volume.
2. Average ambient temperature 23° C.
3. Barometric pressure 29.36 in/hg.
4. M & W Gear nozzle 0.76 mm.
5. Engine speed 2060.

Dual-fueling a Diesel Tractor with M & W Gear Kit

Computed Test Parameters

Run #	Brake Torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	81.34	9.05	5.64	25.31	48.86	14.29	89.05
2	162.69	18.11	10.08	24.26	41.33	15.89	94.65
3	244.03	27.16	13.55	22.91	37.48	16.44	99.13
4	325.38	36.21	16.05	22.15	35.50	16.73	107.52
5	406.72	45.26	18.27	21.53	31.22	18.16	112.76
6	488.06	54.32	19.57	20.54	29.45	18.33	120.21
7	569.41	63.37	20.83	19.82	28.45	18.31	127.22
8	634.48	70.61	21.68	19.03	27.12	18.42	130.58

Notes:

1. The water content of the alcohol 50% by volume
2. Average ambient air temperature 23° C.
3. Barometric pressure 29.36 in/hg.
4. M & W Gear nozzle 0.76 mm.

Alcohol Fuel Test (Dual-fueling with Spray Injection Approach)

Engine Test Data

Run #	Force lbs	Diesel ml	fuel sec.	Alcohol ml	fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Diesel flow Kg/hr	Alcohol flow Kg/hr	Air flow Kg/hr
1	40	100	30.0	90	32.0	240	4.0	1.35	10.24	9.06	272.63
2	80	100	27.0	90	26.5	320	5.0	1.55	11.37	10.94	292.13
3	120	100	25.0	90	23.5	360	6.1	1.70	12.28	12.33	305.94
4	160	100	22.0	90	18.5	400	7.5	1.95	13.96	15.67	327.66
5	200	100	20.5	90	19.0	440	9.2	2.20	14.98	15.25	348.03
6	240	100	19.0	80	12.5	480	11.0	2.45	16.16	20.61	367.27
7	280	100	17.5	80	11.5	520	12.5	2.70	17.55	22.40	385.56
8	320	100	16.0	80	10.5	555	14.5	3.0	19.19	24.53	406.41
9	360	100	15.0	80	10.0	580	16.0	3.25	20.47	25.76	423.01

Notes:
1. The water content of the alcohol 50% by volume.
2. Average ambient temperature 23° C.
3. Barometric pressure 29.36 in/hg.
4. M & W Gear nozzle 0.89 mm.
5. Engine speed 2060.

Dual-fueling a Diesel Tractor with M & W Gear Kit

Computed Test Parameters

Run #	Brake Torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	81.34	9.05	5.49	26.63	30.10	22.16	88.33
2	162.69	18.11	9.69	25.68	26.71	23.63	94.64
3	244.03	27.16	13.32	24.91	24.81	24.42	99.13
4	325.38	36.21	15.19	23.47	20.92	26.54	106.16
5	406.72	45.26	18.14	23.23	19.82	27.68	112.76
6	488.06	54.32	18.99	22.72	17.82	29.10	119.00
7	569.41	63.37	20.40	21.97	17.21	29.12	124.92
8	650.75	72.42	21.31	21.17	16.57	29.15	131.68
9	732.10	81.48	22.58	20.66	16.42	28.83	137.06

Notes:
1. The water content of the alcohol 50% by volume
2. Average ambient air temperature 23° C.
3. Barometric pressure 29.36 in/hg.
4. M & W Gear nozzle 0.89 mm.

Alcohol Fuel Test (Dual-fueling with Spray Injection Approach)

Engine Test Data

Run #	Force lbs	Diesel ml	fuel sec.	Alcohol ml	fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Diesel flow Kg/hr	Alcohol flow Kg/hr	Air flow Kg/hr
1	40	100	30.0	100	26.5	300	4.4	1.45	10.25	12.15	282.55
2	80	100	28.0	100	21.5	340	5.7	1.60	10.97	14.98	296.80
3	120	100	25.5	100	18.5	390	6.8	1.75	12.04	17.41	310.40
4	160	100	24.0	90	15.0	405	8.1	1.95	12.80	19.32	327.66
5	200	100	21.5	80	12.0	440	9.4	2.25	14.28	21.47	351.96
6	240	100	20.0	80	10.5	470	10.9	2.45	15.36	24.53	367.27
7	280	100	18.5	80	10.0	510	12.4	2.65	16.60	25.76	381.97
8	320	100	17.0	70	8.0	540	14.1	2.95	18.07	28.18	403.01
9	360	100	16.0	70	7.5	570	15.6	3.10	19.19	30.05	413.13

Notes:
1. The water content of the alcohol 50% by volume.
2. Average ambient temperature 23° C.
3. Barometric pressure 29.36 in/hg.
4. M & W Gear nozzle 1.0 mm.
5. Engine speed 2060.

Dual-fueling a Diesel Tractor with M & W Gear Kit

Computed Test Parameters

Run #	Brake Torque N.M.	Power kW	Brake thermal efficiency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	81.34	9.05	5.10	27.60	23.25	27.64	91.55
2	162.69	18.11	9.14	27.06	19.82	30.53	96.17
3	244.03	27.16	12.27	25.77	17.83	31.75	100.57
4	325.38	36.21	15.18	25.61	16.96	32.70	106.16
5	406.72	45.26	17.02	24.64	16.40	32.60	114.04
6	488.06	54.32	18.62	23.92	14.97	33.96	113.00
7	569.41	63.37	20.29	23.01	14.83	33.31	123.76
8	650.75	72.42	21.28	22.31	14.30	33.42	130.58
9	732.10	81.48	22.50	21.52	13.75	33.51	133.86

Notes:
1. The water content of the alcohol 50% by volume.
2. Average ambient air temperature 23° C.
3. Barometric pressure 29.36 in/hg.
4. M & W Gear nozzle 1.0 mm.

Alcohol Fuel Test (Dual-fueling with Spray Injection Approach)

Engine Test Data

Run #	Force lbs	Diesel ml	Diesel fuel sec.	Alcohol ml	Alcohol fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Diesel flow Kg/hr	Alcohol flow Kg/hr	Air flow Kg/hr
1	40	100	27.0	100	24.5	320	4.1	1.30	11.37	13.14	267.53
2	80	100	22.5	90	17.0	420	5.4	1.65	13.65	17.05	301.40

* No further increase on load was possible because the tractor was sounding very different and the level of knocking increased quite substantially.

Notes:
1. The water content of the alcohol 50% by volume.
2. Average ambient temperature 23° C.
3. Barometric pressure 28.62 in/hg.
4. M & W Gear nozzle 1.1 mm.
5. Engine speed 2060.

Dual-fueling a Diesel Tractor with M & W Gear Kit

Computed Test Parameters

Run #	Brake Torque N.M.	Power kW	Brake thermal effiiciency %	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	81.34	9.29	4.74	23.52	20.36	27.11	86.68
2	162.69	18.57	7.74	22.08	17.68	28.67	97.66

Notes:
1. The water content of the alcohol 50% by volume.
2. Average ambient air temperature 23° C.
3. Barometric pressure 28.62 in/hg.
4. M & W gear nozzle 1.1 mm.

Injecting Water in a Turbocharged Diesel Tractor with M & W Gear Kit

Engine Test Data

Run #	Force lbs	Diesel ml	fuel sec.	Water ml	fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Diesel flow Kg/hr	Water flow Kg/hr	Air flow Kg/hr
1	40	100	25.5	50	37.0	300	4.1	1.30	12.04	4.87	267.53
2	80	100	22.0	50	29.0	330	5.2	1.50	13.96	6.21	287.38
3	120	100	20.0	50	23.5	365	6.9	1.70	15.36	7.66	305.94
4	160	100	17.5	40	16.5	430	8.6	2.00	17.59	8.73	331.83
5	200	100	16.5	40	14.5	465	10.0	2.15	18.61	9.93	344.05
6	232	100	14.5	40	13.0	520	12.1	2.45	21.18	11.08	367.27

Notes:
1. Distilled water.
2. Average ambient temperature 23° C.
3. Barometric pressure 28.65 in/hg.
4. M & W Gear nozzle 0.64 mm.
5. Engine speed 2100.

Injecting Water in a Turbocharged Diesel Tractor with M & W Gear Kit

Computed Test Parameters

Run #	Brake Torque N.M.	Power kW	Brake thermal efficiency	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	81.34	9.28	6.14	22.21			86.68
2	162.69	18.55	10.60	20.59			93.11
3	244.03	27.83	14.45	19.92			99.13
4	325.38	37.11	16.86	18.91			107.52
5	406.72	46.39	19.87	18.49			111.48
6	471.80	53.81	20.25	17.34			119.00

Notes:
1. Distilled water.
2. Average ambient temperature 23° C.
3. Barometric pressure 28.65 in/hg.
4. M & W Gear nozzle 0.64 mm.

Injecting Water in a Turbocharged Diesel Tractor with M & W Gear Kit

Engine Test Data

Run #	Force lbs	Diesel ml	Fuel sec.	Water ml	fuel sec.	Exhaust temp. °C	Turbo pressure psi	ΔP in/H_2O	Diesel flow Kg/hr	Water flow Kg/hr	Air flow Kg/hr
1	40	100	25.5	50	59.5	280	4.1	1.30	12.04	3.03	267.53
2	80	100	22.5	50	45.5	330	5.5	1.50	13.65	3.96	287.38
3	120	100	19.5	50	37.5	380	7.0	1.75	15.75	4.86	310.40
4	160	100	17.0	50	32.0	430	8.8	2.00	18.07	5.63	331.83
5	200	100	15.9	50	27.0	470	10.6	2.25	19.81	6.67	351.96
6	232	100	14.5	40	19.5	520	12.4	2.50	21.18	7.38	371.00

Notes:
1. Distilled water.
2. Average ambient temperature 23° C.
3. Barometric pressure 28.65 in/hg.
4. M & W Gear nozzle 0.51 mm.
5. Engine speed 2100.

Injecting Water in a Turbocharged Diesel Tractor with M & W Gear Kit

Computed Test Parameters

Run #	Brake Torque	Power kW	Brake thermal efficiency	Air diesel ratio	Air alcohol ratio	% of energy from alcohol	Volumetric efficiency %
1	81.34	9.28	6.14	22.22			86.68
2	162.69	18.55	10.84	21.06			93.11
3	244.03	27.83	14.09	19.71			100.57
4	325.38	37.11	16.37	18.37			107.52
5	406.72	46.39	18.66	17.76			114.04
6	471.80	53.81	20.25	17.52			120.21

Notes:

1. Distilled water.
2. Average ambient air temperature 23° C.
3. Barometric pressure 28.65 in/hg.
4. M & W Gear nozzle 0.51 mm.
5. Engine speed 2060 rpm.
6. Date 11/7/80.

REFERENCES

1 "Balanco Energetico Nacional," Brasil, Ministerio de Minas e Energia, Brasilia D.F. 1978.

2 Barnes, K. D., D. B. Kittleson, and T. E. Murphy. "Effects of Alcohol as Supplemental Fuel for Turbocharged Diesel Engines." SAE Paper No. 750469, Society of Automotive Engineering Congress and Exposition, Detroit, Michigan. February, 1975.

3 Baumeister, Theodore. "Marks' Standard Handbook for Mechanical Engineers. Eighth Edition. McGraw-Hill Book Company, New York. 1978.

4 Cruz, J. Marcio. "Dual-fueling a Diesel Engine with Carbureted Alcohol." M.S. Thesis. University of California at Davis, August 1979.

5 Cruz, J. Marcio, A. S. Ogunlowo, W. J. Chancellor, J. R. Goss. "Biomass-based Fuels for Diesel Engines." ASAE Paper No. 80-027, American Society of Agricultural Engineers Pacific Region Annual Meeting. Hilo, Hawaii. March 18-20, 1980.

6 Ford personal communication, Ford Tractor Department, Dearborn, Michigan. March 1980.

7 Harrington, J. A. and R. M. Pilot. "Combustion and Emission Characteristics of Methanol." Paper 750450, Automotive Engineering Congress and Exposition, Society of Automotive Engineers, Detroit, Michigan. February 1975.

8 Holmer, E. "Methanol as a Substitute Fuel in the Diesel Engines." Paper 2-4 presented at the International Symposium of Alcohol Fuel Technology, Wolfsburg, Germany. November 1977.

9 Hunt, Donnell, R. "Converting a Diesel Engine to Alcohol." Agricultural Engineering. December 1980.

10 Imgamells, J. C. and R. H. Lindquist. "Methanol as a Motor Fuel or Gasoline Blending Component." Chevron Research Company, Paper 750123, Society of Automotive Engineers.

11 Lawson, A., A. Last. "Development of an On-Board Mechanical Fuel Emulsifier for Utilization of Diesel/Methanol and Methanol/Gasoline Fuel Emulsions in Transportation." Ontario Research Foundation. Alcohol Fuel Technology Third International Symposium. Asilomar, California, Vol. I. May 28-31, 1979.

12 Lilgedahl, J. B., Carleton, W. M., Turnquist, P. K., and Smith, D. W. "Tractors and Their Power Units." Second Edition. John Wiley and Sons, New York, 524 pp. 1963.

13 M & W Gear Company. "Aquahol Injection for Diesel Engines." M & W Gear Company, Gibson City, IL. August 23, 1979.

14 Mercedes-Benz Pesquisa e Desenvolvimento: "Combustiveis alternativos em veiculos comerciais." Technical bulletin, Sao Paulo, Brazil. November 1977.

15 Nichols, Roberta. "Modification of a Ford Pinto for Operation on Methanol." Alcohols Fuels Technology Third International Symposium, Asilomar, California, Vol. I, May 28-31, 1979.

16 Obert, Edward F. "Internal Combustion Engines and Air Pollution." In text Educational Publishers, New York, 740 pp. 1973.

17 Panchapakesan, N. R., K. V. Gopalakrishnan and B. S. Murthy (1977). "Factors that Improve the Performance of an Ethanol-Diesel Oil Dual-fuel Engine." Paper 2-2 presented at the International Symposium on Alcohol Fuel Technology, Wolfsburg, Germany. November.

18 Pischinger, G., F. Clymans, and F. Fernandes. "Consumption Data and Considerations on Three Approaches to Diesel-Oil Substitution." VolksWagen do Brasil S/A, Sao Paulo, Brasil.

19 Pischinger, G. and N. L. M. Pinto. "Experiences with the Utilization of Ethanol/Gasoline and Pure Ethanol in Brazilian Passenger Cars." VolksWagen do Brazil. Alcohols Fuels Technology Third International Symposium. Asilomar, California, Vol. I, May 28-31, 1979.

20 Quatro rodas, "Corcel II: Mais vantagem com alcool." Quatro rodas, Braxil. Novembro 1980.

21 Roberson, J. A., Ocaowe, C. T. "Engineering Fluid Mechanics." Second Edition. Houghton Mifflin Company, Boston, MA, 520 pp. 1975.

22 Smith, James. "Breakthrough for Alcohol in Diesel Engines." Department of Agricultural Engineering, Colorado State University. 1980.

23 Solly, Richard K. "Coconut Oil and Coconut Oil-Ethanol Derivatives as Fuel for Diesel Engines." Institute of Natural Resources, University of the South Pacific, P.O. Box 1168, SUVA, FIJI.

24 Stout, B. A. et al. "Energy for World Agriculture." Food and Agriculture Organization of the United Nations, Rome 286 pp. 1979.

25 Taylor, C. F., Taylor, E. S. "The Internal Combustion Engines." Second Edition. International Textbook Company, Scranton, PA. 668 pp. 1966.

26 The National Experiment Station. "The Brandt System." Bulletin. Bellevue, France. September 1941.